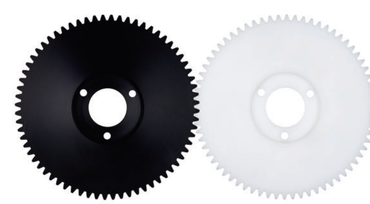
BOEDEKER
PLASTICS

BEARING

ROLLER/WHEEL

SHEAVE

GEAR



ENGINEERING DESIGN GUIDELINES

MITSUBISHI
CHEMICAL
GROUP

Courtesy of supplier partner Mitsubishi Chemical Group (Formerly Quadrant)

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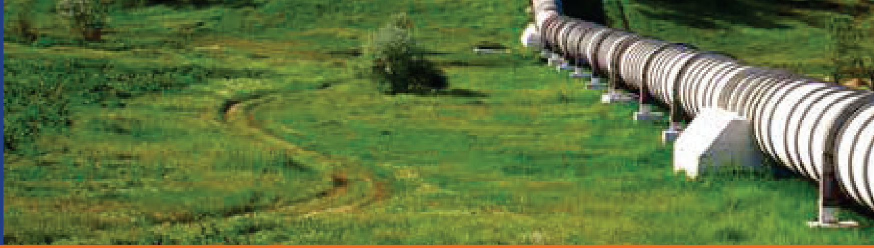
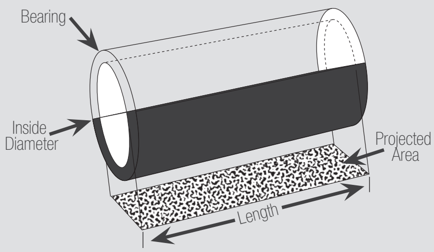
BEARING DESIGN



BEARING CONTACT AREA

FIG 21



Engineering thermoplastics are commonly used as bearings on newly designed and existing machinery, replacing:

- **Rolling element bearings**
- **Metallic plane bearings**
- **Slide pads**
- **Soft metals such as bronze and lead alloys**

PLASTICS > METALS

With plastics' inherently low friction properties, designers often eliminate the need for external lubrication while reducing potential damage to mating surfaces. Selection of an appropriate plastic bearing material requires consideration of an application's unit pressure, calculated linear velocity, ambient temperature and operation cycle time. Other special application requirements such as chemical resistance, dimensional stability and impact resistance must also be considered before final material selection. After choosing an appropriate material, design of the bearing (especially running clearance for any journal bearing) is required.

BEARING AND WEAR PROPERTIES COMPARISON

FIG 22

LESS EXTREME TEMP. ↔ MORE EXTREME TEMP.

Material	Service Temp.	Continuous Limiting PV	"k" Factor	Coefficient Friction of (Dynamic)	Compressive Strength	Cost Factor
TIVAR® 1000 UHMW-PE	180	3,000	111	0.12	3,000	0.5
Acetron® GP POM-C	180	2,700	200	0.25	15,000	1.2
Acetron® POM-H	180	2,700	200	0.25	16,000	1.2
Acetron® AF Blend POM-H	180	8,300	60	0.19	16,000	3.5
Semitron® ESd 225 POM-C	180	2,000	30	0.29	8,000	3.3
Nylatron® 703XL PA6	200	17,000	26	0.14	10,000	1.5
Nylatron® GSM Blue PA6	200	5,500	65	0.18	13,000	1.0
Quadrant® Nylon 101 PA66	200	2,700	80	0.25	12,500	1.0
Nylatron® MC 907 PA6	200	3,000	100	0.20	15,000	1.0
Nylatron® GSM PA6	200	3,000	90	0.20	14,000	1.0
Nylatron® GS PA66	200	3,000	90	0.20	16,000	1.0
Nylatron® NSM PA6	200	15,000**	12	0.18	14,000	1.4
Ertalyte® PET-P	210	2,800	60	0.20	15,000	1.6
Ertalyte® TX PET-P	210	6,000	35	0.19	15,250	1.8
Nylatron® LIG/LFG PA6	220	6,000	72	0.14	13,500	1.0
Nylatron® MC® 901 PA6	260	3,000	100	0.20	15,000	1.0
Techtron® HPV PPS	430	8,750	62	0.20	15,500	22
Techtron® PSBG PPS	450	25,000	800	0.20	15,000	17
Ketron® 1000 PEEK	480	8,500	375	0.32	20,000	19
Ketron® CA30 PEEK	482	25,000	150	0.20	29,000	55
Ketron® HPV PEEK	482	20,000	100	0.21	20,000	30
Duratron® T4301 PAI	500	40,000*	10	0.20	22,000	28
Duratron® T4501 PAI	500	22,500	150	0.20	16,000	28
Fluorosint® 500 PTFE	500	8,000	600	0.15	4,000	12
Fluorosint® 207 PTFE	500	8,000	85	0.10	3,800	12
Fluorosint® HPV PTFE	500	20,000	38	0.15	3,000	12
Duratron® D7015G PI	500	40,000	10	0.25	25,000	63
Duratron® CU60 PBI	600	37,500	60	0.24	50,000	76

* Value represents the LPV for a machined part with post curing after machining. Post curing parts machined from extruded or injection molded Duratron® PAI significantly increases the LPV.

** At surface speeds below 20 ft./min. the LPV (Basic Limiting PV) may be doubled.

STEP 1: DETERMINE BEARINGS' OPERATING PV

Application PV = Pressure (psi) x Velocity (FPM)

Determining Surface Velocity

For sleeve bearings, the formula **$V = 0.262 \times rpm \times D$** is used to determine the surface velocity "V" in fpm, from the shaft diameter, "D" (in.) and the shaft revolutions per minute, or rpm. For linear motion, the surface velocity is the speed at which the sliding surface is moving across the mating surface.

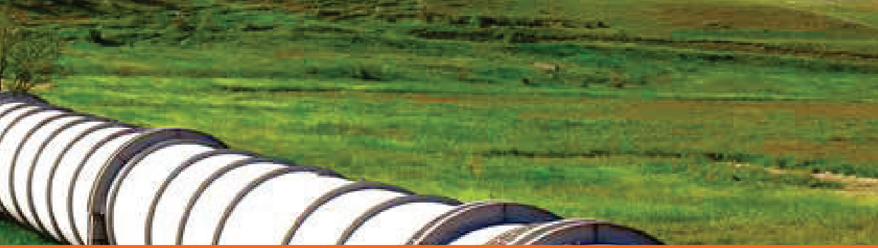
Determining Unit Pressure

For flat bearing surfaces, P is simply the total load (lbs.) divided by the total contact area expressed in square inches (in.^{2}). For sleeve bearings, P is calculated by dividing the total load on the bearing by the projected area of the bearing surface. The projected area of sleeve bearings is calculated by multiplying the bearing I.D. (inches) by the bearing length (inches), as seen in **Figure 21**.

A thermoplastic material must have enough structural and thermal capability to sustain operation at the given application PV. This capability is measured as a material's Limiting PV (LPV). This term is commonly reported as a single value although it may vary for extremes in velocity and load.

TIPS

The maximum unit pressure must always be less than the compressive strength of a selected material. A good design practice is to divide the compressive strength of a material by 4 and use this value as a maximum "working stress" or maximum unit pressure for a plastic bearing.



STEP 2: SELECT A MATERIAL & APPLY THE PV CORRECTION FACTORS

Figure 22 presents LPV values for various Quadrant plastic bearing materials. LPV is the maximum PV that a given material can withstand at 75°F, running continuously without lubrication. The basic LPV taken from this table must be adjusted to compensate for ambient temperatures other than 75°F, and for the cycle time, if continuous operation is not required. Adjustment of LPV is accomplished by multiplying by the correction factors ("H" and "C") obtained from **Figures 23 and 24**. When ambient temperature is approximately 75°F, use H=1 and when bearings are running continuously, C=1. To ensure success, the application PV must be lower than the PV adjusted.

$$PV_{ADJUSTED} = \text{Limiting PV of Quadrant Material Selected} \times H \times C$$

FIG 23

AMBIENT TEMPERATURE CORRECTION (H)

When ambient temperature (surrounding temperature, not heat generated in the bearing from operation) is higher or lower than 75°F, PV capabilities change. Since ambient temperatures above or below 75°F affect the allowable temperature rise and load capability of thermoplastic bearings, use formula below to compensate PV for variations in ambient temperature.

$$PV_{ADJUSTED} = PV \times H$$

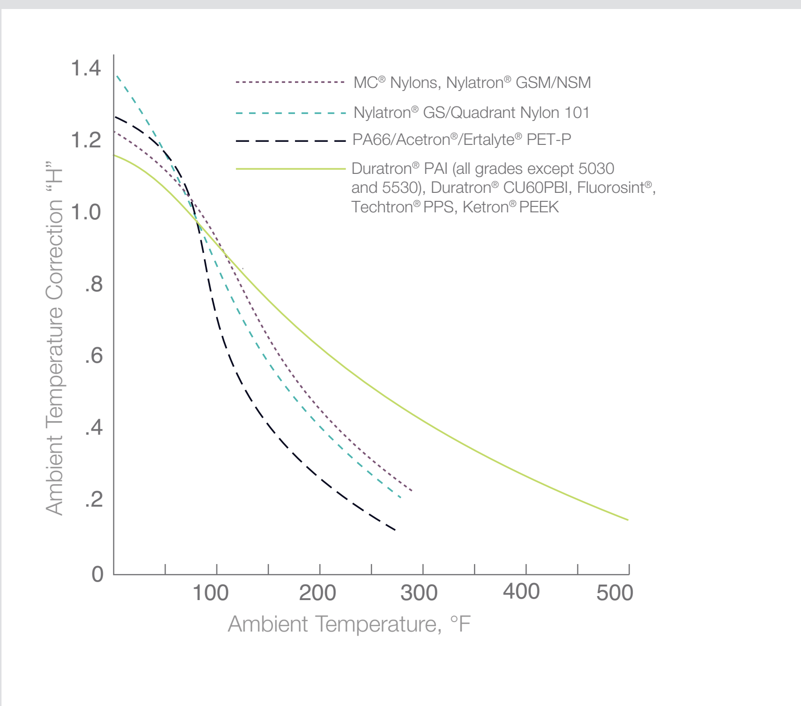


FIG 24

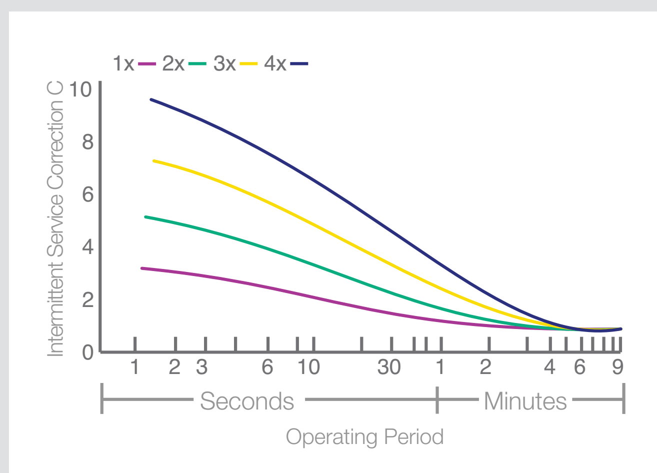
CYCLE TIME CORRECTION (C)

The rates of heat generation and heat dissipation greatly determine the performance of plastic bearings. If operation is intermittent rather than continuous, the rate of heat generation is reduced although the rate of heat dissipation remains constant.

INSTRUCTIONS FOR USE:

Locate operating period or "on" period on horizontal scale. Read upwards to intersect with the appropriate curve. If the off period is the same as the on period, use the (1X) curve. If the off period is two times the on period, use the (2X) curve. Interpolate conservatively. For example, if off period is three and one-half times the on period, use the (3X) curve.

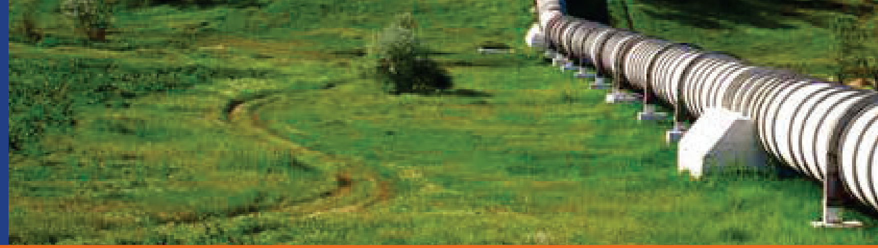
$$PV_{ADJUSTED} = PV \times C$$



TIPS

Continuous lubrication including oil, grease, and water greatly increase the service life of thermoplastic bearings. Lubrication is usually suggested for velocities greater than 400 FPM.

BEARING DESIGN



(a₁) BASIC SHAFT ALLOWANCE VERSUS SHAFT DIAMETER

FIG 25

Shaft Allowance, a_1 (inches)	Shaft Diameter (inches)
.005	1”
.009	2”
.012	3”
.015	4”
.017	5”
.020	6”
.022	7”
.024	8”
.026	9”
.028	10”
.030	11”
.032	12”

STEP 3: BEARING CLEARANCE

Clearance has been the least understood and most frequently encountered problem in the design of plastic bearings. Most plastic bearing failures are caused by insufficient clearance.

Plastic bearing clearances are much greater than those recommended for metal bearings. Metal bearings installed with excessive clearance often result in shaft vibrations and scoring (brinnelling) of the bearing and shaft. Plastics, on the other hand, are far more resilient, resist scoring and dampen shaft vibration. Total running clearance is obtained by adding three allowances. The total running clearance is then added to the nominal bearing I.D. (shaft diameter) to obtain the actual or design I.D. of the bearing.

TOTAL RUNNING CLEARANCE = $a_1 + a_2 + a_3$

a_1 = Basic shaft allowance.

The basic shaft allowance **a_1** is the same for all plastic bearing materials and depends only on the diameter of the shaft to be supported. **Figure 25** was developed from application data on plastic bearings.

FIG 26

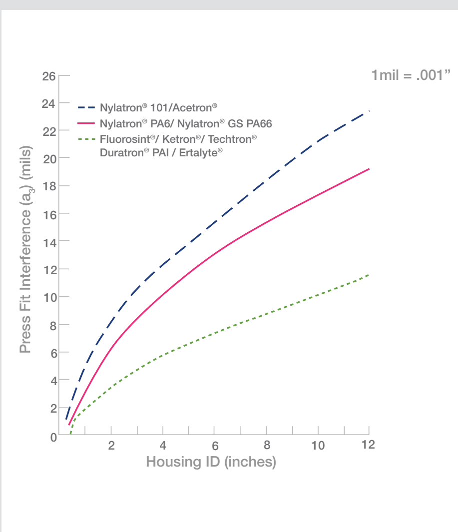
(a_2) Wall factor for plastic bearing materials at various ambient temperatures - for calculation of a_2 (inches)

	75°	100°	125°	150°	175°	200°	225°	250°	275°	300°	350°	400°	450°	500°
Quadrant® Nylon 101 PA66/Acetron® POM	.018	.021	.023	.026	.028	.031	.033	.036	.038					
Nylatron® PA6 grades	.015	.016	.018	.019	.021	.023	.024	.026	.026					
Nylatron® GS, Ertalyte® PET-P	.013	.015	.016	.018	.020	.022	.023	.025	.027					
Fluorosint® PTFE	.007	.007	.008	.008	.009	.009	.010	.010	.011	.011	.012	.013	.014	.015
Ketron® HPV PEEK, Techtron® HPV PPS	.007	.007	.008	.008	.009	.009	.010	.010	.011	.011	.012	.013	.014	.015
Bearing grade Duratron® PAI	.007	.007	.008	.008	.009	.009	.010	.010	.011	.011	.012	.013	.014	.015
Duratron® CU60 PBI	.007	.007	.008	.008	.009	.009	.010	.010	.011	.011	.012	.013	.014	.015

Note: For temperatures other than given use the next highest temperature that appears in the table.

(a_3) RECOMMENDED PRESS FIT INTERFERENCE VERSUS HOUSING INSIDE DIAMETER

FIG 27



a_2 = Wall thickness allowance (a function of the bearing material, bearing wall thickness, and the ambient operating temperature) **Obtain wall factor from Figure 26 and multiply by the nominal wall thickness to obtain a_2 .**

Wall thickness allowance (**a_2**) is derived from the coefficients of thermal expansion for the plastic bearing materials. Each plastic reacts to changing temperatures at a characteristic rate. The thicker the bearing wall, the more material there is available to expand with higher temperature. Hence, **Figure 26** demonstrates that the higher ambient temperatures and/or thicker bearing walls, the greater the required running clearance.

a_3 = **Press fit allowance**: Used only when the bearing is to be press fit. Note that **a_3** is the same as the recommended press fit interference **(obtain from Figure 27)**.

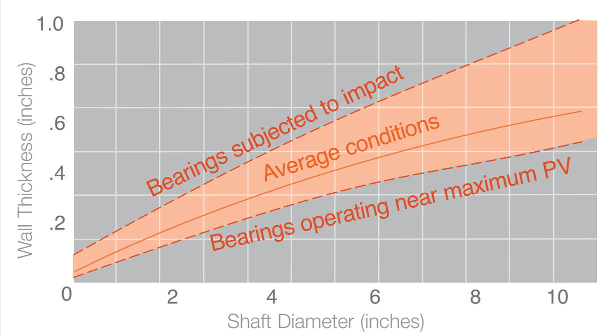
When plastic bearings are press fit into metallic housings or retainers, a recommended interference **(Figure 27)** should be used to ensure that the bushing is adequately secured to resist rotating with the shaft. During press fit, the plastic bearing conforms to the housing I.D. Therefore, the I.D. of the bearing closes-in. The I.D. close-in will approximately equal the press fit interference. Close-in is compensated with an additional I.D. clearance equal to the interference (**a_3**).

STEP 4: ADDITIONAL DESIGN CONSIDERATIONS

A) BEARING WALL THICKNESS

In many bearing applications, the nominal wall thickness is dictated by the geometry of existing equipment. The plastic bearing is designed from the dimensions of the shaft and the housing. When new equipment is being designed, the engineer is at greater liberty to establish nominal wall thickness. **Figure 28** suggests a range of nominal wall thicknesses for different shaft diameters. Maximum walls are recommended for bearings subjected to severe impact conditions, and minimum walls for bearings operating near the material's maximum recommended PV value.

FIG 28 BEARING WALL THICKNESS



B) BEARING LENGTH / DIAMETER RATIO

Bearing length to shaft diameter ratio has a noticeable effect on bearing friction. For a ratio of 1:1 (bearing length equal to the shaft diameter), friction is generally lowest. As the bearing length is increased to two or three times the shaft diameter, there is increased friction and an increased probability of local heating due to out-of-roundness and shaft vibration. On the other hand, very short bearings are often difficult to retain within the bearing housing.

C) SHAFTS AND MATING PARTS

Shafts and mating parts perform best if made from hardened and ground steel. Unhardened steel surfaces will wear quickly in many applications, particularly if unlubricated. Commercial shafting normally is supplied with a surface hardness of Rockwell C-55, although shafting with Rockwell hardnesses as low as C-35 will perform satisfactorily. Shafts and mating parts of stainless steel should be specified in a hardenable grade. In general, harder stainless grades such as 316 are suggested over 303/304 grades.

Mating metal parts should have a smooth surface obtained by grinding or hard plating. Commercial shafting normally is finished to 16 RMS although a 32 RMS is usually acceptable. The finish of the plastic bearing is not critical and can be as coarse as 125 RMS.

TIPS

TIVAR® UHMW-PE materials have a lower mechanical strength than other traditional thermoplastic bearings. As a result, please review TIVAR® Bushing/Bearing Design Specifications below.

D) TIVAR® UHMW-PE BUSHING/BEARINGS DESIGN SPECS

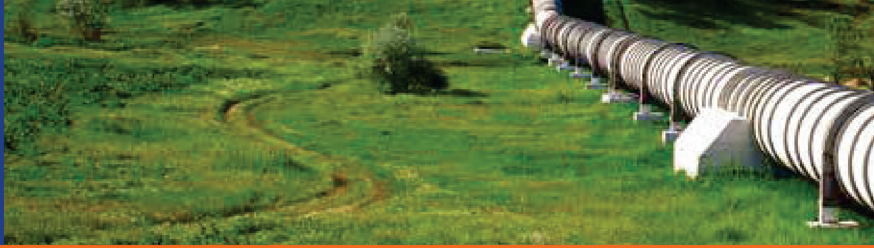
Press Fitting TIVAR® UHMW-PE Bearings:

- Add .8 to 1.0% to the nominal OD on bearing:
  $(OD_b - ID_h) / ID_h \times 100 = .8\%$ to 1.0%
  OD_b = Bearing OD
  ID_h = Mating Housing ID
- Bearing length to diameter ratio should be equal to or less than 1.5: $L/OD_b \leq 1.5$
- For each .004" or .10mm added to the nominal bushing O.D. for press fitting into a housing, the bushing I.D. will close in .001" or .03mm

Shaft Diameters/TIVAR® UHMW-PE Bearings:

- To produce a running fit, increase the nominal bearing I.D. by .001" or .03mm for shaft diameters less than 1" or 25mm in size.
- To produce a running fit on shafts 1" or 25mm and larger, increase the nominal bearing I.D. by .003" or .07mm for each 1" or 25mm in size.
- Recommended bearing wall thickness is one tenth of shaft diameter when designing a TIVAR® bearing.
- Increase the wall thickness for shock load conditions and decrease the wall thickness for applications near the limiting PV value.
- It is recommended that the length of a TIVAR® UHMW-PE bearing be equal to the shaft diameter unless under a high load, where more surface area is required to resist creep.

BEARING DESIGN



Duratron® CU60 PBI bushing next to the steel bearing it replaced due to shaft galling.

Techtron® HPV PPS

Nylatron® GSM Blue PA6

NYLATRON® BEARING FOR WET APPLICATIONS

If your bearing is to be water lubricated and made from a Quadrant Engineering Plastic Products' nylon, an additional clearance must be added for moisture expansion of the nylon. Use clearances below regardless of bearing diameters. Note that as wall thickness increases, moisture clearance increases in progressively smaller amounts. This is due to the increasing resistance of the thicker sections to moisture penetration. Add a moisture factor in your bearing design for water lubricated nylon bearings per table below:

Moisture Factor

1/8"	clearance in inches is	0.012"
3/16"	clearance in inches is	0.017"
1/4"	clearance in inches is	0.021"
3/8"	clearance in inches is	0.026"
1/2"	clearance in inches is	0.030"
3/4"	clearance in inches is	0.032"
1" +	clearance in inches is	0.033"

- Non-hygroscopic materials such as Ertalyte® PET-P and Acetron® GP POM-C may offer improved wear resistance in wet environments.

TIPS

Internally lubricated materials such as Nylatron® NSM PA6, Nylatron® GSM Blue PA6 nylon and Ertalyte® TX provide the lowest cost in use when application PV is less than Limiting PV.

BEARING DESIGN WORKSHEET

Note: This worksheet applies to sleeve bearings only. Contact Quadrant at TechServices@qplas.com or via our live chat feature at quadrantplastics.com with any questions.

INFORMATION REQUIRED

Housing bore ____________________ in.

Shaft diameter____________________ in.

Length ____________________ in.

Shaft rpm ____________________

Bearing load____________________ lbs.

How many bearings/shaft __________

Ambient temperature __________ °F

Cycle ❑ Continuous

❑ Intermittent

Time on______ Time off______

❑ Is bearing lubricated?

How? ____________________

BEARING DESIGN

STEPS 1 & 2: DETERMINE BEARINGS' OPERATING PV, SELECT MATERIAL & CORRECTION FACTORS

Projected area

Bearing ID ______________ x Length________________ = __________ sq. in.

Pressure

Bearing load _____________ ÷ Projected area __________ = __________ psi

Velocity

0.262 x ______________ rpm x Shaft diameter __________ = __________ fpm

(Note: do not exceed 400 fpm for velocity for unlubricated applications)

Application PV (imposed PV on bearing)

_____________ psi x ______________ fpm = __________ PV

See (Figure 22, page 34) for Limiting PV

Lubricated ❑ Unlubricated ❑

Material Selected ____

Corrections for Limiting PV - See **(Figures 23 and 24, page 35)**

Figure 23 – Temperature Correction H = __________

Figure 24 – Cycle Time Correction C = __________

$PV_{ADJUSTED}$ (Limiting PV for material selected; then adjusted by temperature and cycle corrections)

Limiting PV __________ x Temp. (H) _________ x Cycle (C) _________ =
of material

$PV_{ADJUSTED}$ _________

If the application PV is less than the $PV_{ADJUSTED}$ limit for the material selected, the bearing will work.

STEPS 3 & 4: TOTAL RUNNING CLEARANCE

$a_1 + a_2 + a_3$

a_1 = **(Figure 25, page 36)** = _________

a_2 = **(Figure 26, page 36)**

Bearing wall $\frac{(OD - ID)}{2}$ x Temp. factor for material

= _________

a_3 = **(Figure 27, page 36)** – used if bearing is press fit

= _________

STEP 5: DIMENSION OF THE BEARING

Housing dia.: ________ $+a_3$ ________ = OD of bearing ________

Shaft dia.: _____ $+ a_1$ _____ $+ a_2$ _____ $+ a_3$ = ID of bearing ________

If a nylon bearing is to be used in a water lubricated environment, add moisture factor per **page 38** *to the ID of the bearing to allow for moisture absorption:*

ID of bearing: ________ + Moisture absorption clearance ________ = ID of bearing

Length of housing _____________

STEP 6: BEARINGS, DIMENSIONS AND TOLERANCES

OD = _____________ ±0.004 in. or ± 0.001 in./in. of dia.

ID = _______________ +0.008 / -0.000 in. or +0.002/ -0.000 in./in. of dia.

Length = _______________ ±0.010 in. or ± 0.001 in./in. of length

**The greater of the tolerances will apply.*

ROLLER/WHEEL DESIGN

Rigid plastic rollers and wheels are commonly specified instead of metal. The non-abrasive and vibration dampening characteristics of the plastic rollers/wheels result in quieter operation. Typical rigid plastic roller/wheel material choices are:

- Acetron® POM Grades
- Nylatron® PA Grades
- Ertalyte® PET-P Grades

PLASTICS > METALS

Rigid plastics are also replacing traditional resilient elastomers such as polyurethane and vulcanized rubber. The rigid plastics are chosen for their lower coefficient of rolling resistance.

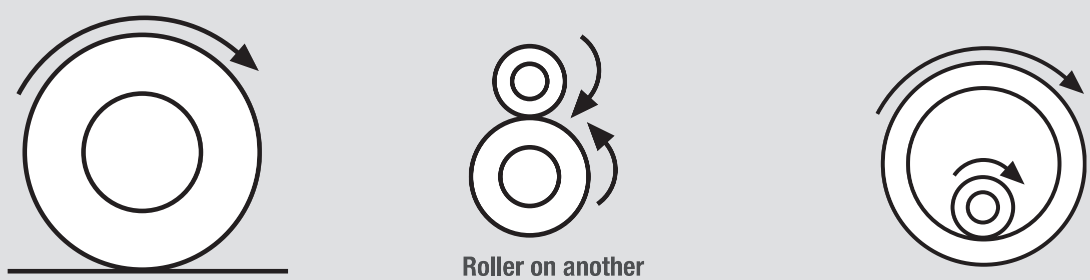
To determine the suitability of a rigid plastic roller/wheel, consider:

- Load upon the roller/wheel
- Speed of the roller/wheel
- Temperature around and on the roller/wheel
- Duty cycle of the roller/wheel – whether it is stationary or rotating
- Creep and fatigue properties of the roller/wheel material

The creep and fatigue properties play an important role in preventing flat spots, cracking and softening of the rollers/wheels in end-use. The first step in calculating suitability is to determine the load capacity of the proposed material. The load capacity equation is dependent upon the geometry and configuration of the wheels/rollers.

(1) roller on a flat surface (Figure 29 a.)
(2) roller on another rolling surface (Figure 29 b.)
(3) roller in another rolling surface (Figure 29 c.)

FIG 29



Roller on a flat surface (Figure 29 a.)

Roller on another rolling surface (Figure 29 b.)

Roller in another rolling surface (Figure 29 c.)

ROLLER/WHEEL DESIGN

DETERMINING THE LOAD CAPACITY OF A ROLLER/WHEEL

1: Select the roller configuration.

1. Roller on a flat surface
2. Roller on another rolling surface
3. Roller in another rolling surface **(See Figure 29)**

2: Select the potential roller/wheel material. For initial material selection, consider environmental temperature & load conditions for the application.

3: From **Figure 30**, obtain the material stress factor, K.
Note: Separate values are given for stationary vs. rotating situations.

4: Using the equation provided for the selected roller configuration, calculate the load capacity of the roller/wheel.

(1) Roller on a flat surface **(Figure 29 a.)**

$$W_{MAX} = K\,(L)\,(D_p)$$

(2) Roller on another rolling surface **(Figure 29 b.)**

$$W_{MAX} = K\,(L)\left(\frac{D_p \times D_m}{D_m + D_p}\right)$$

(3) Roller in another roller surface **(Figure 29 c.)**

$$W_{MAX} = K\,(L)\left(\frac{D_p \times D_m}{D_m - D_p}\right)$$

Where:
W_{MAX} = Maximum allowable contact load (lbs.)
D_p = Diameter of plastic roller (in.)
D_m = Diameter of metal roller (in.)
L = Contact length of roller (in.)

FIG 30 MATERIAL STRESS: FACTOR (K)*

LESS EXTREME ↔ MORE EXTREME

Material	Stationary	Rotating
TIVAR® 1000 UHMW - PE	5	12
Fluorosint® PTFE	5	17
Semitron® ESd 225 POM-C	23	76
Quadrant® Nylon 101 PA66	30	99
Nylatron® GSM Blue PA6	32	106
Nylatron® GSM PA6	39	130
Nylatron® NSM PA6	39	130
Techtron® PSBG PPS	42	75
Acetron® POM-H	45	150
Acetron® AF Blend POM-H	45	149
Acetron® GP POM-C	45	150
Nylatron® MC901 / 907 PA6	45	150
Ertalyte® PET–P	46	142
Nylatron® GS PA66	49	162
Techtron® HPV PPS	70	170
Duratron® T4503 PAI	89	157
Duratron® T4301 PAI	91	161
Duratron® T4501 PAI	96	170
Duratron® T4540 PAI	95	170
Ketron® CM CA30/HPV PEEK	96	171
Ketron® 1000 PEEK (Extruded)	120	213
Ketron® HPV PEEK	120	171
Ketron® CA30 PEEK	132	234
Duratron® T4203 PAI	168	298
Duratron® CU60 PBI	215	383

Based on maximum allowable contact stresses (psi).

TIPS

Load capacity calculations are purposefully conservative and are based on a 4x safety factor used to determine K. Designers are encouraged to test all rollers and wheels in conditions similar to those anticipated.

ROLLER/WHEEL DESIGN

ASSEMBLY/FABRICATION

The three commmon rigid plastic wheel/roller designs are:

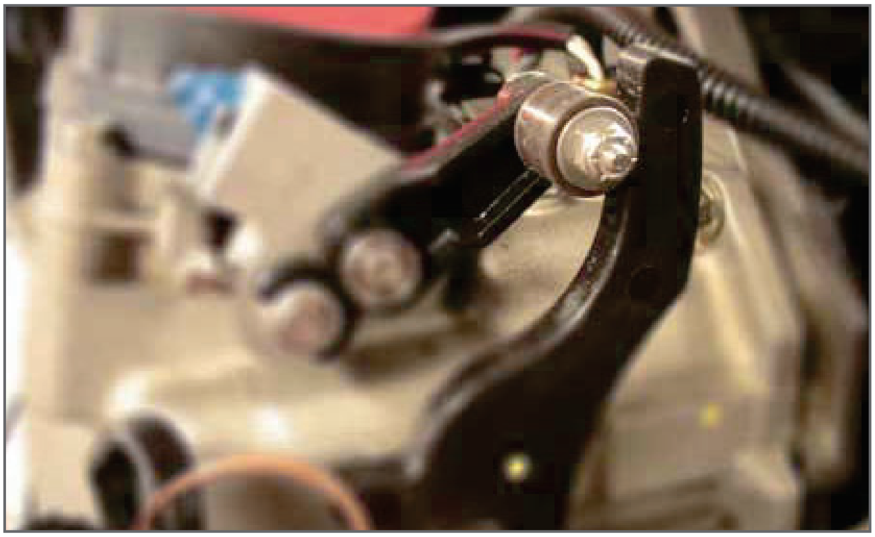
- **Solid rollers rotating directly on the shaft**
- **Solid rollers with ball or roller bearings**
- **Plastic sleeves on metal cores**

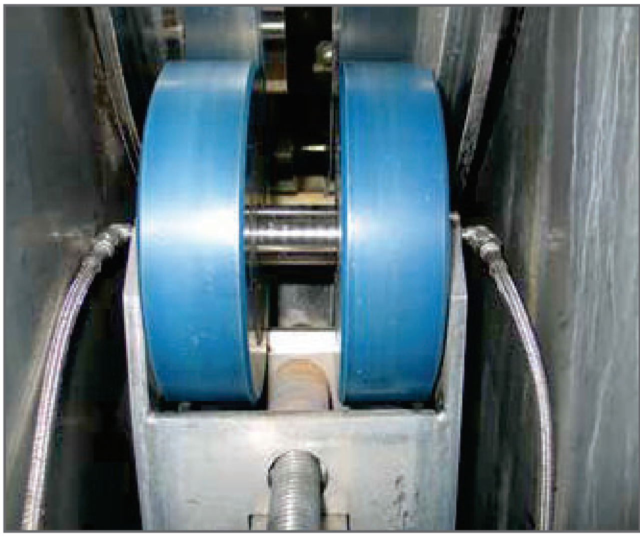
See **Figure 31** for details on the typical uses, advantages, limitations, and design/fabrication tips for these typical roller designs.

FIG 31

TYPICAL ROLLER/WHEEL DESIGNS

Roller/Wheel Design	Typical Use Conditions	Advantages	Limitations	Design/Fabrication Tips
Solid rollers rotating directly on the shaft	Intermittent service Low velocity Low load	Lowest cost	Design must account for moisture and temperature growth	Calculate Limiting PV and required running clearance with bearing design equations. Prevent lateral binding by considering the material's moisture and temperature growth when calculating the axial clearance.
Solid rollers with press-fit ball or roller bearings	For operating temperatures up to 120°F (49°C)	Quick and easy assembly	Not suitable for side-loaded wheels/ rollers	Press-fit made easier by heating-up the plastic roller.
Solid rollers with mechanically fastened snap rings or metal flanges	For operating temperatures above 120°F (49°C) For side loaded wheels/rollers	Mechanical fastening prevents axial movement		For rolling element bearings: Prevent axial and circumferential movement by securing the outer race. Press the bearing into the flanged sleeve. Then press into the wheel/ roller. Secure with a bolt through the flange to the roller.
Plastic sleeves on metal cores	High loads High temperatures High speeds	Balances the impact resistance of the plastic sleeve with the heat dissipation of the metal core		Make plastic wall thickness 10 to 15% of metal core OD. Contact Quadrant for design options.





SHRINK FITTING

Shrink fitting is the most common assembly method. Shrink fit interference and axial clearance depends upon the roller/wheel's operating temperature. **Figure 32** contains the interference and clearances for four elevated temperatures. To assemble, heat the plastic sleeve to 200°F.

Quadrant manufactures cast nylon roll covers for shrinking onto metal cores. Cast nylon roll covers are available in diameters up to 25" and in lengths up to 84". To shrink onto core, simply heat the plastic sleeve and metal core to 200°F and assemble with the aid of a hydraulic press.

CASTING PLASTIC SLEEVE ONTO METAL CORE

Directly casting the nylon plastic sleeve onto the metal core is the most efficient assembly method. It also eliminates slippage between the plastic sleeve and the metal core – the most common issue for shrink fits. Casting on the metal core is ideal for wheels/rollers with face widths less than 1".

FIG 32

INTERFERENCES & CLEARANCES AT ELEVATED TEMPERATURES

Average Operating Temperature of Sleeve	Shrink Fit Interference at 68°F (20°C). Value is in % of diameter.	Axial Clearance (b) at 68°F (20°C). Value is in % of sleeve width.
100°F (38°C)	0.25	0.05
140°F (60°C)	0.45	0.20
175°F (80°C)	0.65	0.40
200°F (93°C)	0.85	0.60

SHEAVE DESIGN

NYLATRON® GSM PA6 SHEAVES

SUPPORT THE SAME LOAD AS METAL

Stress on the wire rope – not the sheave – commonly limits the lifting capacity of a system. The point contact pressure for a steel sheave will be much higher than for a Nylatron® nylon sheave, and the resilience of nylon results in a larger point contact area and creates support for the wire rope. Lightweight Nylatron® nylon sheaves can support cyclical loads equal to steel sheave capabilities.

REDUCE WEIGHT

Because Nylatron® GSM PA6 nylon is approximately one seventh (1/7) the weight of conventionally used cast steel, Nylatron® nylon sheaves reduce dead weight at the end of the boom. This provides mobile cranes with greater stability and lifting capacity and lowers over-the-road weight.

The reduced weight of Nylatron® GSM PA6 sheaves makes handling, installation and replacement significantly easier and safer than with comparable metal sheaves.

EXTEND WIRE ROPE LIFE

Quadrant Engineering Plastic Products, in conjunction with a nationally recognized independent research institute, conducted wire rope endurance tests to obtain a comparison of the fatigue life of wire rope used with Nylatron® GSM PA6 sheaves and hardened steel sheaves under the same conditions.

Test results at stress levels of 10%, 20%, and 28.6% of ultimate wire rope strength indicate dramatic improvements in the endurance life of wire rope when used with cast Nylatron® sheaves. **Figure 33** summarizes results of the wire rope life testing. The tests prove Nylatron® nylon sheaves substantially increase rope cycle life.

RESISTS CORROSION

The corrosion resistant properties of nylon make these plastic parts ideal for marine use.

PLASTICS > METALS

For many years, manufacturers and operators of heavy-duty lifting equipment have sought ways to increase wire rope endurance life. Early attempts included lining the grooves of metal sheaves with resilient materials and mounting rims made of these materials on metal hubs. Growth in manufacturing of mobile lifting equipment now requires designers to consider reducing the dead weight of metal sheaves on the boom or mast, and improving lift and over-the-road performance. Expansion in offshore exploration has also generated a need for lifting equipment with corrosion resistant parts.

With the development of Nylatron® GSM PA6 cast nylon sheaves, the search for improved wire rope life, reduced weight, and corrosion resistance has been resolved. Nylatron® nylon sheaves are widely used on both mobile and offshore lifting equipment.

TIPS

- **Bronze bearings are not recommended for main load applications. Their use should be limited to moderate unit loads to avoid excessive frictional heat build-up and possible movement of the bearing in the bore.**
- **For lightly loaded applications where pressure-velocity (PV) values are not excessive, it may be possible to plain bore Nylatron® nylon sheaves for running directly on the shaft. Contact Quadrant at TechServices@qplas.com or via our live chat feature at quadrantplastics.com for appropriate running clearance information.**

FIG 33

WIRE ROPE LIFT TEST RESULTS*

Sheave Ratio	Rope Tension for Test	Approximate Design Factor (Fd)	Duration of Test	Increase in Rope Life Attained with Nylatron® GSM PA6 Sheaves*
24/1	10.0% of breaking strength	10.0	136,000 cycle	4.50 times
24/1	20.0% of breaking strength	5.0	68,000 cycles	2.20 times
24/1	28.6% of breaking strength	3.5	70,000 cycles	1.92 times
18/1	28.6% of breaking strength	3.5	39,000 cycles	1.33 times

Sheave Ratio = D_p / D_r = Sheave pitch diameter/rope diameter

*Conventional rope retirement criteria based only upon visible wire breaks may prove inadequate in predicting rope failure. Retirement criteria should be established based on the users' experience and demands of the specific applications for users of Nylatron® nylon sheaves.

SHEAVE DESIGN

DESIGN GUIDELINES

When designing with custom or standard sheaves, certain considerations should be observed by equipment engineers. Of special importance are groove configuration, bore configuration, bearing retention, and load capacity **(See Figure 35 - Page 50)**.The basic design of any sheave should conform to the appropriate minimum pitch diameter/rope diameter sheave ratios of 18/1 and 24/1 for the mobile crane industry. The 18/1 ratio conforms to the Power Crane and Shovel Associations and American National Standards Institute (ANSI) minimums for load hoisting cranes. The 24/1 ratio complies with most European standards and should be considered for export requirements.

RIM DIMENSIONS

The rim width (W_r), outside diameters (D_O), and tread diameters (D_T) are typically fixed design dimensions. The rim flat (F_r - shown in **Figure 35**) between the groove wall and rim edge should be a minimum of 1/8" to provide adequate side load stability.

GROOVE DIMENSIONS

The groove radius (R_g) for a Nylatron® nylon sheave should be a minimum of 5% greater than the nominal rope diameter divided by 2 to accommodate rope tolerances while giving adequate rope support.

$$R_g = 1.05\ (D_r\ /2)$$

Experience indicates that a groove angle Θ_g of 30° will generally provide optimum rope support for mobile crane sheaves.
Fleet angles $\geq$ 2° up to 4% generally require a 45° groove angle.
Typical American and European practice requires that the depth of the rope groove for mobile crane sheaves be made a minimum of 1.75 times the rope diameter.

WEB DIMENSIONS

Practical experience with crane sheaves has shown that the required design strength can be maintained with a minimum web width that is 10% greater than the rope diameter or:

$$W_w = 2.2\ (R_g)$$

Where: $W_w = 1.1 \bullet$ Groove Diameter
$R_g = 1.05 \bullet D_r / 2$

The benefit of reducing the web width is weight savings.
Additional strength can be obtained by adding ribs to the design.

HUB DIMENSIONS

The hub width (W_h) is generally a design requirement specified by the end user. In most cases it should be equal to or greater than the rim width for stability of the sheave in use. The minimum hub diameter (D_h) is 1.5 times the bearing outside diameter (D_b) for adequate wall support of the bearing. The wall thickness between the bearing and hub diameter should always be greater than 1".

$$D_h = 1.5(D_b)$$

The transitions from the hub diameter to the web and the web diameter to the rim must be tapered and radiused as appropriate based upon the design thicknesses and diameters.

BORE DIMENSIONS

Nylatron® nylon sheaves for heavy-duty applications should be installed with antifriction bearings. Needle roller bearings are generally recommended, as they provide a continuous contact area across the width of the bore. As the coefficient of thermal expansion of nylon is several times that of metal, the press fit allowance must be large enough for the bearing to maintain contact with the bore at temperatures up to 140°F.

$$d = .009\ \sqrt{D_b}$$

Where: d = Press fit allowance (in.)
D_b = Bearing outside diameter (in.)

The diameter of the sheave bore will be the O.D. of the bearing minus the press fit allowance. $D_B = D_b - d$
Sufficient press fit is critical to prevent buckling of a loaded sheave.

BEARING RETENTION

Circumferential bearing retention can be achieved using the press fit allowances (as calculated under bore dimensions) and pressing directly into the bore of the Nylatron® nylon sheave. A hydraulic press can be used, or the sheave can be heated to 180°-200°F and the bearing dropped into the expanded bore. Thrust washers or thrust plates should be placed on either side of the sheave hub to maintain sideways bearing retention. This is necessary to restrict bearing movement which may occur as the result of side forces encountered during operation.

There are two exceptions to bearing retention using the above procedure:

- **Two-row double-cup tapered roller bearings in heavy-duty sheave applications**
- **Bronze bearings in idler sheaves where the sheave is free to move from side-to-side on a shaft**

Since thrust washers or thrust plates cannot be used, other means of retention must be found to restrict sideways movement of the bearing.

A positive retention method for two-row double-cup tapered roller bearings is to place a steel sleeve insert in the bore of the Nylatron® sheaves into which the cup is pressed. The insert is held in the bore by external retaining rings on each side of the hub.

Positive retention of bronze bearings in Nylatron® idler sheaves can be accomplished by extending the length of the bushing beyond the hub on both sides, and placing external retaining rings on each side of the hub. Metal side plates bolted to the hub and overlapping the ends of the bearing can also be used for this purpose.

A steel sleeve insert, held in the bore by external retaining rings, is recommended with the use of two-row double-cup tapered roller bearings.

SHEAVE DESIGN

1: LOAD CAPABILITY OF NYLATRON® NYLON SHEAVES (WITH BEARINGS)

The following equations can be used to calculate the maximum groove and bore pressure acting on any sheave.

$$P_g = \frac{2\,(LP_{MAX})\,K_{\Theta}}{D_r \bullet D_t} \qquad (1)$$

$$P_b = \frac{2\,(LP_{MAX})\,K_{\Theta}}{D_b \bullet W_h} \qquad (2)$$

Where:

- P_g = Max groove pressure (psi)
- P_b = Max bore pressure (psi)
- LP_{MAX} = Max single line pull (lb.) or wire rope breaking strength divided by design safety factor
- D_r = Rope diameter (in.)
- D_t = Tread diameter (in.)
- D_b = Bore diameter (in.)
- W_h = Hub width (in.)
- K_{Θ} = Wrap factor = $\sin\left(\frac{\text{wrap angle}}{2}\right)$
- Θ = Wrap angle

FIG 34 **WRAP ANGLE FACTORS** K_{Θ}

Wrap Angle Θ*	K_{Θ}
180°	1.000
170°	0.996
160°	0.985
150°	0.966
140°	0.940
130°	0.906
120°	0.866
110°	0.819
100°	0.766
90°	0.707
80°	0.643
70°	0.573
60°	0.500

** Arc of groove contacted by rope.*

Maximum service pressure can safely reach 8,600 psi for short term loads (a few minutes). Maximum service pressure for static loads (>100 hours) should not exceed 3,500 psi. Equations (1) and (2) can be rewritten to calculate the maximum line pull for a Nylatron® sheave:

$$Lp_{MAX} = \frac{1750\,(D_r \bullet D_t)}{K_{\Theta}}$$

$$Lp_{MAX} = \frac{1750\,(Wh \bullet D_b)}{K_{\Theta}}$$

2.LOAD CAPABILITY OF PLAIN BORED SHEAVES

The load capacity for a plain bored Nylatron® nylon sheave is based upon the ability of the bore to act as a bearing. To determine the recommended load capacity, refer to the Bearing Design section of this manual, and make calculations as follows, assuming that the bore of the sheave is a Nylatron® GSM PA6 nylon bearing.

First, obtain the recommended limiting pressure velocity value ($PV_{ADJUSTED}$) for the given operating conditions. Next, calculate the maximum bore pressure from the equation:

$$P_b = \frac{PV_{ADJUSTED}}{V}$$

Where:

- P_b = Maximum bore pressure (psi)
- PV_a = Pressure velocity value (psi • fpm)
- V = Shaft surface speed (fpm)
- = 0.262 x shaft rpm x Ds (fpm)
- D_s = Shaft diameter (in.)

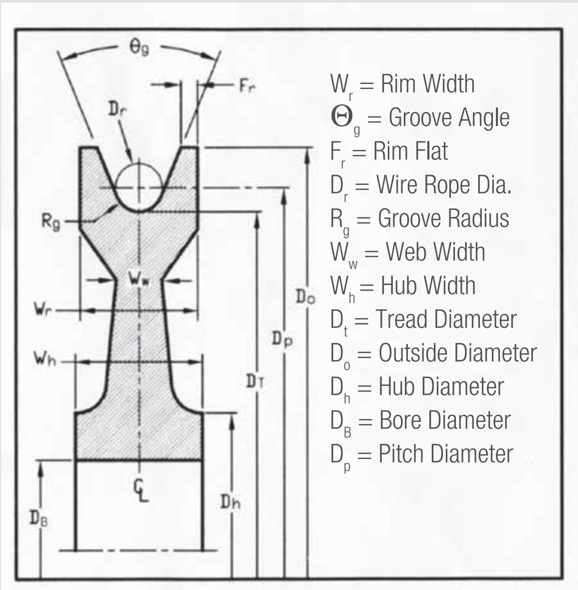
Bore pressure Pb should not exceed 1,000 psi. Take the calculated value for Pb or 1,000 psi, whichever is less, and substitute in the following equation to obtain the maximum load capacity for the conditions specified:

$$LC = P_b \bullet D_s \bullet W_h$$

Where:

- LC = Max load capacity (lbs.)
- W_h = Width of hub in contact with shaft (in.)

FIG 35 **SHEAVE NOMENCLATURE**



SHEAVE DESIGN

INFORMATION REQUIRED

Maximum single line pull (Load) _______________ lbs.
Line speed _______________ ft./min.
Fleet angle _______________ degrees
Temperature low __________°F high __________°F

Wrap Angle (Arc of sheave contacted by rope) __________°

SHEAVE DATA

Drawing Number? ________________
If no drawing is available...
W_r Rim width ________________ inches
D_o Outer diameter ________________ inches
D_t Tread diameter ________________ inches
D_h Center hub O.D. ________________ inches
W_h Hub width ________________ inches
D_b Center bore I.D. ________________ inches
Alignment or access holes required?

____

Number? ________________________
Pitch Circle? ______________________
Grease fittings? ____________________
Type? ___________________________
Location? ________________________

WIRE ROPE DATA

Rope O.D. __________ inches
Rated breaking strength _______
Brand of rope in use ___________________________

BEARING SPECIFICATIONS

Design _________________________
Mfr / Part Number ________________
O.D. of outer race __________inches
Bearing width __________inches
Method of attachment ___________________________

If you require any further assistance or a quote, contact Quadrant at TechServices@qplas.com or via our live chat feature at quadrantplastics.com for more information.

TIPS

- **Contact Quadrant at TechServices@qplas.com or via our live chat at quadrantplastics.com for special design requirements including underwater cable systems, V-belt applications, high temperature, sheave ratios below 18:1, fleet angles greater than 3°, or severe chemical environments. Industries that use sheaves for power transmission or load lifting applications typically have other bearing and wear requirements that could also benefit from the use of Quadrant's products.**
- **Nylatron's® wear and impact resistance, light weight, and corrosion resistance present unique advantages in a wide variety of wear and structural components (i.e. slide bearings, wire guides, bushings, rollers and roll covers).**
- **The pressure and load capacity limits recommended here are based on intermittent cyclical loading as in typical mobile hydraulic crane operation. If operation involves continuous cycling or loading, high speed and acceleration, or heavy impact forces, the limits should be reduced and the application thoroughly evaluated.**
- **Excessive loads and/or speeds may cause distortion of the bore and loss of press fit with the bearing. Accelerated groove wear may also result. For plain bored sheaves, excessive loads and/or speeds may cause accelerated wear and increased clearance in the bore.**

TIPS

- **The use of nylon thrust washers or plates where they will wear against the Nylatron® nylon sheave hub is not recommended.**
- **Calculation of tread pressure is not necessary if the ratio of groove diameter to rope diameter is 18:1 or larger.**

GEAR DESIGN

ENGINEERING PLASTIC GEARS OFFER:

PLASTICS > METALS

- Quiet operation
- Ability to run without lubrication
- Corrosion resistance
- Longer wear life and protection of mating gears
- Reduced inertia versus traditional all metal gears

Nylatron® has been the standard material of choice. It has been successfully used in a variety of industries for spur, worm, bevel, and helical gears for well over 40 years. All over the world, thermoplastic gears continue to replace traditional materials like steel, cast iron, bronze, phenolic, and even wood. Nylatron® balances strength, heat resistance, fatigue properties, impact resistance, and wear resistance; making it the most popular choice for gearing. Acetron® POM-C acetal, TIVAR® UHMW-PE, Techtron® HPV PPS, Ketron® PEEK, and new higher performance materials offer specific advantages for wet/high humidity conditions, chemically aggressive environments, light duty service, or high temperature applications.

DESIGNING NYLATRON® GEARS

Although nylon has significantly lower strength than a corresponding metal gear, reduced friction and inertia coupled with the resilience (bending) of thermoplastic gear teeth make direct substitution possible in many applications – especially gears made from nonferrous metals, cast iron and unhardened steel.

A step-by-step method for evaluating suitability of nylon spur gears is provided here. This method was developed using Quadrant's gear fatigue test data, and the maximum allowable bending stress of plastic gear teeth **(See Figure 36)**. Proper gear design will include calculation of a maximum allowable Torque (T_{MAX}) and/or a maximum allowable Horsepower (HP_{MAX}) for a given thermoplastic material. Applying a few critical correction factors are also essential to your design. Also provided are calculations for specific correction factors which can be accounted for in your design.

CORRECTION FACTORS

- (C_M) **Material Strength Factor:** To compare Nylatron® gears with other thermoplastic materials, one can multiply the calculated maximum torque (T_{MAX}) and horsepower (HP_{MAX}) values for Nylatron® spur gears by the Material Strength Factor **(See Figure 40)** of the material in question to determine appropriate torque and horsepower values.

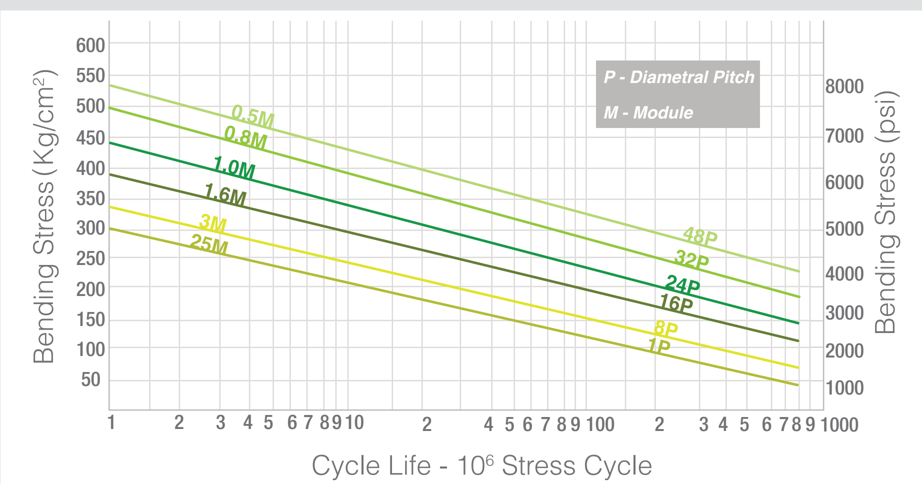
- (C_V) **Pitch-line Velocity:** Gear velocity can affect the performance capability of a thermoplastic gear. **Figure 41** provides correction factors for various gear speeds. Increased speeds will lower the maximum torque (T_{MAX}) and horsepower (HP_{MAX}) values. Nylatron gearing can operate up to pitch-line velocities of 4,000 to 6,000 fpm with continuous lubrication to reduce heat build-up.

- (C_S) **Service Life Factor:** Proper gear design is dependent on not only the application conditions, but how many rotation cycles the gear is expected to achieve. The number of expected gear cycles and the gear pitch will also affect the calculated maximum torque (T_{MAX}) and horsepower (HP_{MAX}) values. Utilize **Figure 42** for this factor.

- (C_T) **Temperature Correction Factor:** Increased service temperature of the gear application will equate to some amount of material softening, which also reduces expected maximum torque (T_{MAX}) and horsepower (HP_{MAX}) values. Apply the correction factor per **Figure 43** to account for this reduced load capability.

FIG 36

MAXIMUM TOOTH BENDING STRESSES VS. CYCLE LIFE FOR NYLON GEARS



Based on Pitch Line Velocity of 2,000 Ft./Min

P Diametral Pitch - Ratio of N (number of teeth) to P_d (pitch diameter)

M Module - is the metric equivalent to P

GEAR DESIGN

GEAR DESIGN METHOD

1: OBTAIN THE REQUIRED APPLICATION DATA:

(P) Diametral Pitch **$P = N/P_d$**
(N) Number of Teeth
(PA) Pressure Angle
(F) Face Width, inches
(RPM) Input RPMs
(T_I) Input Torque
(HP_I) Input Horsepower

2: CALCULATE DERIVED DATA AND CORRECTION FACTORS

(P_d) Pitch Diameter = **$P_d = N/P$**
(Y) Tooth Form Factor - **From Figure 38**
(S_B) Bending Stress - **From Figure 39**
(C_M) Material Strength Factor - **From Figure 40**
(C_V) Velocity Factor - **From Figure 41**
(C_S) Service Lifetime Factor - **From Figure 42**
(C_T) Temperature Factor - **From Figure 43**

FIG 37 GEAR DESIGN

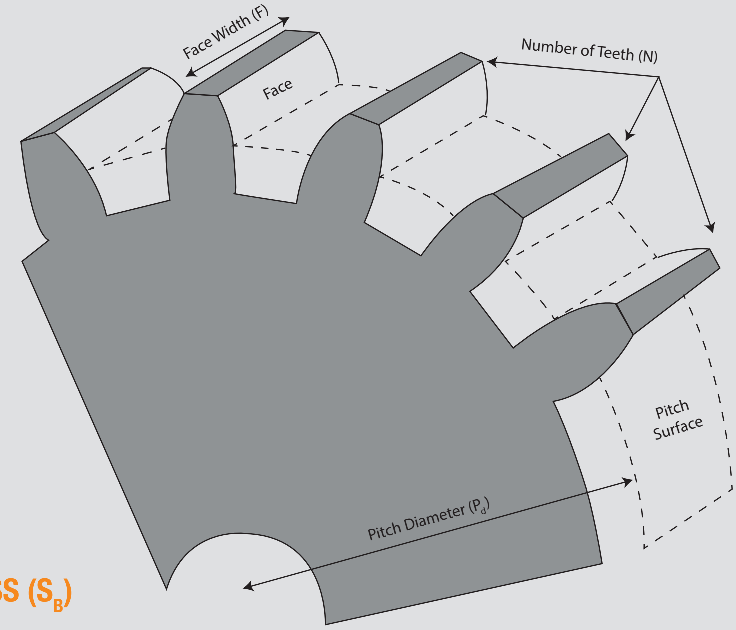


FIG 38 TOOTH FORM FACTOR (Y)

Number of Teeth	$14^{1/2°}$	20° Full Depth	20° Stub
	Pressure Angle		
14	–	–	0.540
15	–	–	0.566
16	–	–	0.578
17	–	0.512	0.587
18	–	0.521	0.603
19	–	0.534	0.616
20	–	0.544	0.628
22	–	0.559	0.648
24	0.509	0.572	0.664
26	0.522	0.588	0.678
28	0.535	0.597	0.688
30	0.540	0.606	0.698
34	0.553	0.628	0.714
38	0.566	0.651	0.729
43	0.575	0.672	0.739
50	0.588	0.694	0.758
60	0.604	0.713	0.774
75	0.613	0.735	0.792
100	0.622	0.757	0.808
150	0.635	0.779	0.830
300	0.650	0.801	0.855
Rack	0.660	0.823	0.881

FIG 39 NYLON BENDING STRESS (S_B)

Pitch	S_B
2	1994
3	2345
4	2410
5	2439
6	2675
8	2870
10	3490
12	3890
16	4630
20	5005

FIG 40 MATERIAL STRENGTH FACTOR (C_M)

Material	Operating Conditions: Non-Lubrication	Periodic Lubrication	Continuous Lubrication
Nylatron® NSM PA6	1.00	1.00	1.20
Nylatron® GS, GSM PA6	0.49	0.94	1.26
Nylatron® MC901/907 PA6	0.49	0.94	1.26
Acetron® GP POM-C	*	*	1.04
Phenolic	*	0.96	1.13
TIVAR® UHMW-PE	*	*	0.75

** Data not available*

FIG 41 VELOCITY FACTOR (C_V)

Velocity-fpm	Correction Factors
500	1.38
1000	1.18
2000	1.00
3000	0.93
4000	0.90
5000	0.88

FIG 42 SERVICE LIFE FACTOR (C_S)

Number of Cycles	16 pitch	10 pitch	8 pitch	5 pitch
1 million	1.26	1.24	1.30	1.22
10 million	1.00	1.00	1.00	1.00
30 million	0.87	0.88	0.89	0.89

FIG 43 TEMPERATURE FACTOR (C_T)

Materials	< 100°F C_T =	100°F to 200°F $C_T = 1 / [1 + \alpha(T - 100°F)]$
Nylatron® GSM, NSM, and MC Nylons	1.0	$\alpha = 0.022$
Nylatron® GS and Quadrant® Nylon 101 PA66	1.0	$\alpha = 0.004$
Acetron® GP POM-C	1.0	$\alpha = 0.010$

3: CALCULATE THE MAXIMUM TORQUE OR HORSEPOWER

Calculate the maximum allowable torque or horsepower, then multiply by the appropriate correction factors.

$$T_{MAX} = \frac{P_d \, S_B \, F \, Y}{2P} \times C_M \, C_V \, C_S \, C_T \quad \text{(Equation 1)}$$

$$HP_{MAX} = \frac{P_d \, S_B \, F \, Y \, RPM}{126{,}000 \, P} \times C_M \, C_V \, C_S \, C_T \quad \text{(Equation 2)}$$

4: COMPARE TO KNOWN INPUT TORQUE OR HORSEPOWER

Compare the maximum torque (T_{MAX}) and maximum horsepower (HP_{MAX}) above for plastic gears to the known input torque (T_I) and/or horsepower (HP_I).

- **T_I must be less than or equal to T_{MAX}**

or

- **HP_I must be less than or equal to HP_{MAX}**

If T_I and HP_I exceed the T_{MAX} and HP_{MAX} for the plastic gear, select another material or another pitch diameter and face width, then re-calculate using the new material correction factors.

DESIGN FOR OTHER GEAR STYLES:

The design formulas for spur gears may be modified when designing for other gear types which will have differing tooth contact forces. Detailed here are corrections for helical and bevel gears.

HELICAL GEARS

The Tooth Factor (Y) must be derived using a calculated Formative Number of Teeth (N_f) based on the following equation. Use this calculated number of teeth with Table 1 to determine Tooth Form Factor (Y).

$$N_f = \frac{N_H}{(\cos \psi)^3}$$

Where:

N_f = Formative number of teeth
N_H = Actual number of teeth (helical)
ψ = Helix angle (degrees)

In addition, a Normalized Diametral Pitch (P_N) is used which is calculated from the transverse diametral pitch (P_t) which is the pitch in the plane of rotation. Use P_N in place of P for Pitch Diameter (P_d) calculations. This is calculated from:

$$P_N = \frac{P_t}{\cos \psi}$$

P_N = Normalized Diametral Pitch
P_t = Transverse Diametral Pitch
ψ = Helix angle (degrees)

BEVEL GEARS

The Tooth Form Factor (Y) must be derived using a calculated Formative Number of Teeth (N_f) based on the following equation. Use this calculated number of teeth with **Figure 38** to determine Tooth Form Factor (Y).

$$N_f = \frac{N_B}{\cos \varnothing}$$

Where:

$\varnothing$ = Pitch angle (degrees)
N_B = Actual number of teeth (bevel)

It should be noted that Diametral Pitch (P) and Pitch Diameter (P_d) refer to the outside or larger tooth dimensions of bevel gears.

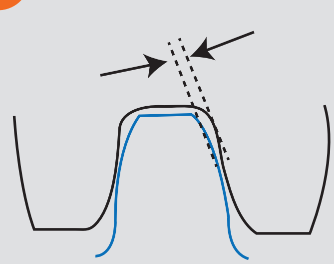
ADDITIONAL GEAR DESIGN CONCERNS:

Nylatron® Gears versus other materials – Nylatron® gears are generally superior to other engineering plastics, provided environmental factors such as temperature, humidity, and chemicals are within its useable limits. The choice of material depends on both environmental and operating running conditions.

Mating Gear Materials – For best operation, a Nylatron® gear should be mated with a metallic gear, as this arrangement promotes heat dissipation. Consider that the wear of a plastic gear is largely determined by the counterface, or opposing gear. A surface finish of 12 to 16 μ in. minimum is recommended on metal gears running against plastic gears. In general, it is best to avoid making both driven and driving gears from similar plastics. If an all plastic gear system is desired, a combination of dissimilar plastics is recommended (e.g. Nylatron® PA6 with Acetron® POM-C).

Backlash – The most frequent design error when converting metal to plastic gears is not allowing sufficient backlash. Plastics have a greater thermal expansion versus metals, and thus sufficient backlash must be designed in to compensate for frictional heat and changes in ambient conditions. The suggested backlash can be calculated using **Figure 44**:

FIG 44 BACKLASH



Backlash should be checked upon installation through a full rotation of the plastic gear.

Backlash = 0.100” / P
where P = Diametral Pitch

For a more stable material, like Ketron® PEEK:

$$\text{Backlash} = \frac{0.100''}{2P}$$

ADDITIONAL GEAR DESIGN CONCERNS (CONT.):

Moisture Absorption – Nylatron® does absorb some moisture, and will therefore increase slightly in size. However, most gears are of such a heavy cross-section that moisture pickup is extremely slow and does not require any special consideration when designing the gear. Again, increased backlash compensates for growth due to moisture.

Tooth Form – Field experience has shown that Nylatron® gearing can operate successfully utilizing any of the standard tooth forms in use today. However, when designing new equipment, it is suggested that consideration be given to the 20° pressure angle (PA) full depth tooth form (full root radius) to maximize bending strengths of the gear teeth. For Nylatron® spur gears, load carrying capacity is approximately 15% greater in a gear designed with a 20° PA versus a 14.5° PA. Also, service life increases by approximately 3.5 times will be seen under the same load.

Extended Performance – Where design permits, select the smallest tooth that will carry the load required. This will minimize heat build-up from higher teeth sliding velocities. Also, for higher torque capability, consider nylon gear blanks cast directly over machined steel inserts.

GEAR ASSEMBLY

Gears are commonly fastened to shafts using a variety of techniques including:

- Press fit over splined and/or knurled shafts for gears transmitting low torques
- Set screws for economical low torque gears
- Bolting a metal hub through the gear width is suitable for drive gears produced in small to intermediate quantities
- Machined keyways for gears carrying higher torques

KEYWAYS

When using a keyway to assemble a gear, radiused keyway corners are always preferred to reduce the stress concentrations and provide greater strength and toughness. The minimum keyway area is determined from the formula:

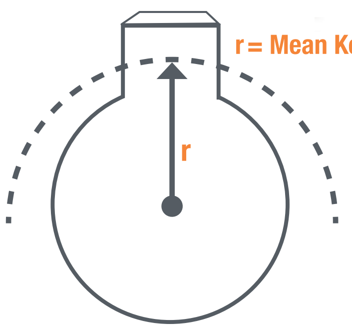
$$A = \frac{63{,}000\ HP}{RPM\ r\ S_k}$$

Where:
- A = Keyway area
- HP = Horsepower transmitted
- RPM = Gear speed (rpm's)
- r = Mean keyway radius
- S_k = Maximum permissible keyway stresses from Figure 45

FIG 45

MAX KEYWAY STRESS (S_K)

Materials	S_k (psi)
Nylatron® GS PA66	1,500
Nylatron® PA66	1,500
Nylatron® GSM/MC901 PA6	2,000
Acetron® POM	2,000
TIVAR® UHMW-PE	300



If the keyway size determined from the equation below is impractical and multiple keyways cannot be used, then a keyed flanged hub and check plate bolted through the gear should be used. The required number of bolts and their diameters at a particular pitch circle radius is calculated from a modified form of the equation:

$$\text{Minimum Number of Bolts} = \frac{63{,}000\ HP}{RPM\ r_1\ A_1\ S_k}$$

Where:

r_1 = Pitch circle radius of bolts

A_1 = Projected area of bolts (bolt diameter x gear width in contact with bolts)

Raise fractional values to the next highest number of bolts. Do not excessively tighten bolts during assembly to avoid the risk of gear distortion or bolt shearing due to material expansion during normal running. Consequently, the use of cup washers or similar are recommended where practical, although nylon washers provide a satisfactory alternative.

TIPS

Be sure to design in a .015" to .030" radius for keyway corners.

SPUR GEAR DESIGN WORKSHEET

STEP 1 – OBTAIN REQUIRED APPLICATION DATA

P	Diametral Pitch $P = N/P_d$	______________
N	Number of Teeth	______________
PA	Pressure Angle	______________
F	Face Width, inches	______________
RPM	Input RPMs	______________
T_I	Input Torque...or...	______________
HP_I	Input Horsepower	______________

STEP 2 – CALCULATE DERIVED DATA AND CORRECTION FACTORS

P_d	Pitch Diameter $P_d = N/P$	______________
Y	Tooth Form Factor **(From Figure 38)**	______________
S_B	Bending Stress **(From Figure 39)**	______________
Alternate Material		______________________________
C_M	Material Strength Factor **(From Figure 40)**	______________
C_V	Velocity Factor **(From Figure 41)**	______________
C_S	Service Life Factor **(From Figure 42)**	______________
C_T	Temperature Factor **(From Figure 43)**	______________

T_{MAX} Maximum Torque (in lbs) $= [P_d\, S_B\, F\, Y] / 2P \quad x \quad C_M\, C_V\, C_S\, C_T$ ________

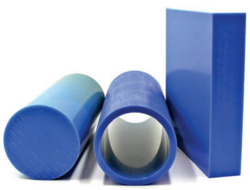
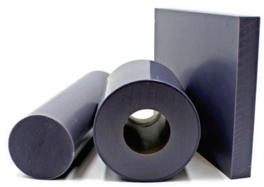
HP_{MAX} Maximum Horsepower $= [P_d\, S_B\, F\, Y\, RPM] / 126{,}000\, P \quad x \quad C_M\, C_V\, C_S\, C_T$ ________

FINAL STEP: Ensure $T_I < T_{MAX}$ or that $HP_I < HP_{MAX}$

ADDITIONAL GEAR
DESIGN TIPS

TIPS

- Heat dissipation and therefore performance is optimized by running plastic gears against metal gears. When running an all plastic gear system, dissimilar materials are suggested (e.g. nylon with acetal).
- Where design permits, select the smallest tooth that will carry the load required. This will minimize heat build-up from higher teeth sliding velocities.
- For higher torque capability, consider gear blanks cast directly over machined steel inserts.
- Nylatron® nylon gears are generally superior to other engineering plastics provided environmental factors such as temperature, humidity and chemicals are within its usable limits. The choice of material depends on both environmental and operational running conditions.
- The wear of a plastic gear is largely determined by the counterface, or opposing gear. In general, it is best to avoid making both driven and driving gears from similar plastics. Most plastic gears wear well against metal. A surface finish of 12-16 μ in. minimum is recommended on metal gears running against plastic gears.
- If the Nylatron® gear is to be completely immersed in water, it is suggested that you contact Quadrant at TechServices@qplas.com or via our live chat feature at quadrantplastics.com for design assistance.

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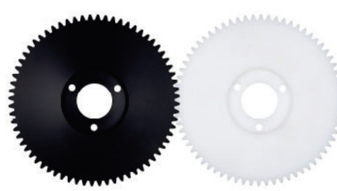


In addition, we offer CNC Machined and Injection Molded parts with our Plastics Only CNC Machine shop and Injection Molding Facility.

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We Stand Ready to Assist