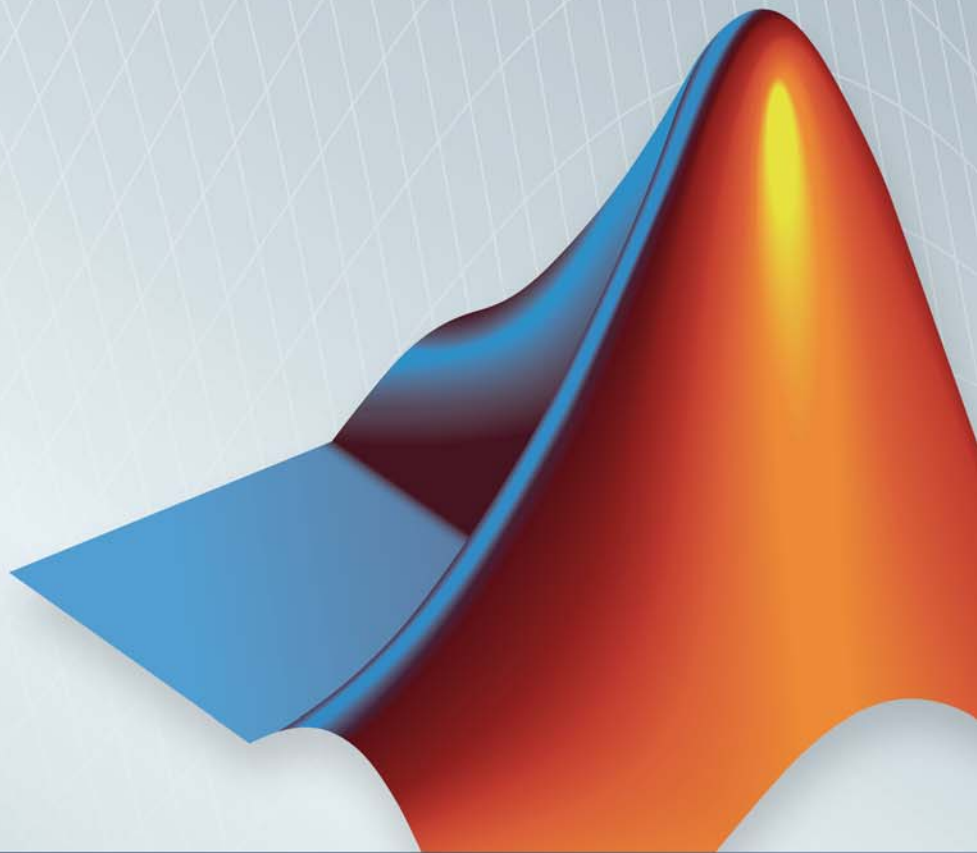


**SimDriveline™**

Reference

**R2013b**



**MATLAB® & SIMULINK®**



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*SimDriveline™ Reference*

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**Revision History**

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New for Version 2.0 Beta (Release 2010b+)  
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Revised for Version 2.1 (Release 2011b)  
Revised for Version 2.2 (Release 2012a)  
Revised for Version 2.3 (Release 2012b)  
Revised for Version 2.4 (Release 2013a)  
Revised for Version 2.5 (Release 2013b)



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# Blocks — Alphabetical List

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# Band Brake

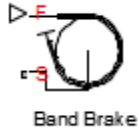
## Purpose

Frictional brake with flexible band wrapped around rotating drum

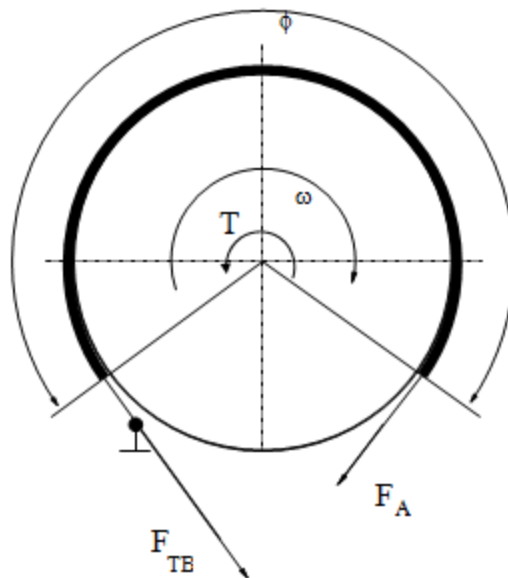
## Library

Brakes & Detents/Rotational

## Description



The block represents a frictional brake with a flexible band that wraps around the periphery of a rotating drum to produce a braking action. A positive actuating force causes the band to tighten around the rotating drum and it places the friction surfaces in contact. Viscous and contact friction between the drum and the flexible band surfaces cause the rotating drum to decelerate. Band brakes provide high braking torque at the cost of reduced braking precision in applications that include winch drums, chainsaws, go-karts and mini-bikes. The model employs a simple parameterization with readily accessible brake geometry and friction parameters.



The following formula provides the braking torque as a function of the external brake actuation force that tightens the belt.



$$T = (F_{TB} - F_A) r_D + \mu_{visc} \omega$$

In the formula, the parameters have the following meaning:

Parameter	Description
$T$	Braking torque
$F_{TB}$	Force acting on the tense branch of the band
$F_A$	External brake actuation force
$r_D$	Drum radius
$\mu_{visc}$	Viscous friction coefficient
$\mu$	Contact friction coefficient
$\phi$	Wrap angle

Forces  $F_{TB}$  and  $F_A$  satisfy the relationship:

$$\frac{F_{TB}}{F_A} = e^{\mu\phi}$$

Replacing the relationship in the braking torque formula eliminates force  $F_{TB}$ :

$$T = F_A (e^{\mu\phi} - 1) r_D + \mu_{visc} \omega$$

To avoid discontinuity at zero relative velocity, the model defines the actuation force  $F_S$  as a hyperbolic function:

$$F_A = F_{in} \tanh(4\omega / \omega_{threshold})$$

In the previous formula, the parameters represent the following quantities:

# Band Brake

Parameter	Description
$F_{in}$	Force input signal
$\omega_{threshold}$	Angular velocity threshold

Connection F is a physical signal port that represents the external tensioning force applied on the belt. Connection S is a conserving rotational port that represents the rotating drum shaft.

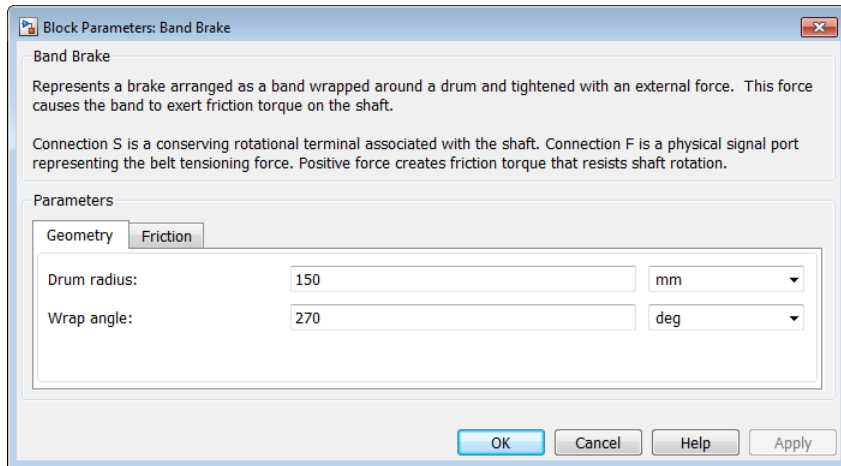
## Assumptions and Limitations

- The model does not account for actuator flow consumption

## Dialog Box and Parameters

The dialog box includes two tabs, **Geometry** and **Friction**.

### Geometry



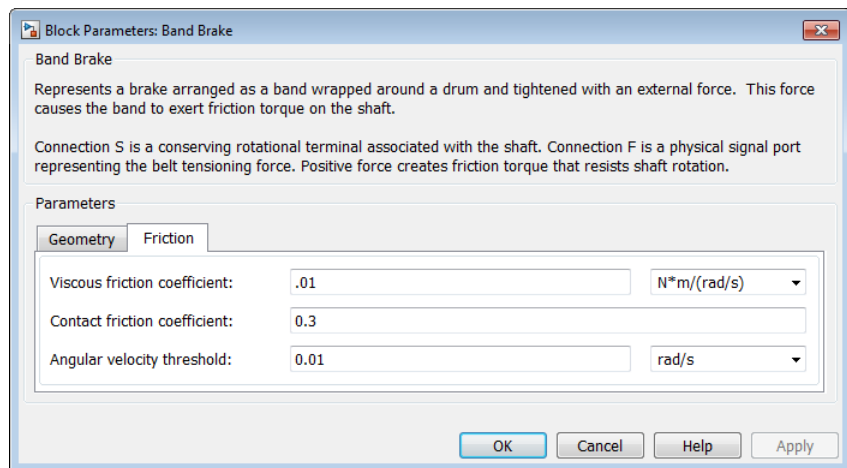
### Drum radius

Radius of the drum contact surface. The parameter must be greater than zero. The default value is 150 mm.

## Wrap angle

Contact angle between the flexible belt and the rotating drum. The parameter must be greater than zero. The default value is 130 mm.

## Friction



## Viscous friction coefficient

Value of the viscous friction coefficient at the belt-drum contact surface. The parameter must be greater than or equal to zero. The default value is .01 N\*m/(rad/s).

## Contact friction coefficient

Value of the Coulomb friction coefficient at the belt-drum contact surface. The parameter must be greater than zero. The default value is 0.3.

## Angular velocity threshold

Angular velocity at which the friction coefficient at the belt-drum contact surface practically reaches its steady-state value. The parameter must be greater than zero. The default value is 0.01 rad/s.

# Band Brake

---

## Ports

Port	Description
F	Physical signal port that represents the belt tensioning force
S	Conserving rotational port that represents the rotating drum shaft

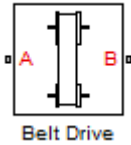
## See Also

Double-Shoe Brake, Loaded-Contact Rotational Friction, Rotational Detent

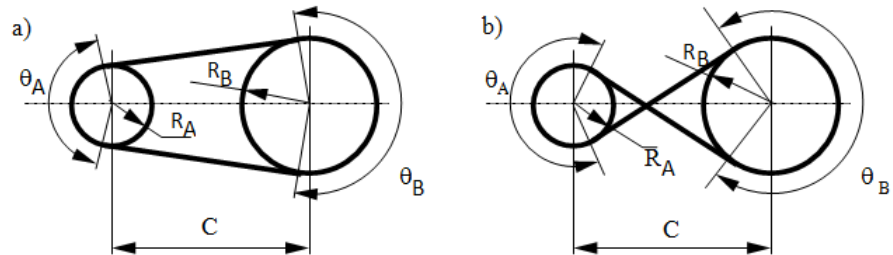
**Purpose** Flexible belt wrapped around two pulleys

**Library** Couplings & Drives

**Description**



The block represents a pair of pulleys connected with a flexible flat band or V-shaped belt. The model accounts for friction between the flexible belt and the pulley periphery. If the friction force is not sufficient to drive the load, the model allows slip. The relationship between the tensions in the tight and loose branches conforms to the Euler equation. The model accounts for centrifugal loading in the flexible belt, pulley inertia, and bearing friction. The schematic shows the belt drive.



In this schematic, a) represents an open belt configuration, and b) represents a crossed belt configuration. In the open belt configuration, both pulleys tend to rotate in the same direction, and the larger pulley tends to have a larger belt wrap angle. In the crossed belt configuration, the pulleys tend to rotate in opposite directions, and both pulleys have the same wrap angle.

For open belt configurations, the following equations compute the wrap angles:

$$\theta_A = \pi - 2 * \sin^{-1} \frac{R_A + R_B}{C},$$

$$\theta_B = \pi + 2 * \sin^{-1} \frac{R_A + R_B}{C}$$

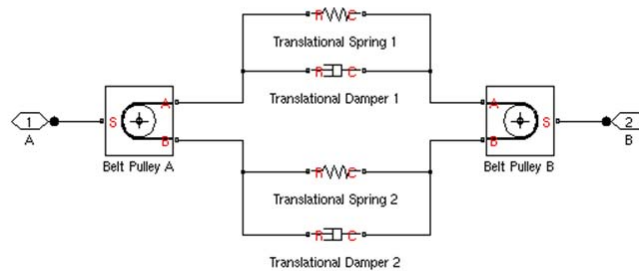
# Belt Drive

- $\theta_A$  is the wrap angle of pulley A.
- $\theta_B$  is the wrap angle of pulley B.
- $R_A$  is the effective radius of pulley A.
- $R_B$  is the effective radius of pulley B.
- $C$  is the distance between the centers of pulley A and pulley B.

The wrap angles for crossed belt configurations use this equation:

$$\theta_A = \theta_B = \pi + 2 * \sin^{-1} \frac{R_A + R_B}{C}$$

This schematic shows the open band configuration.



The spring and damper blocks account for compliance in the belt.

For more information about the dynamic equations used for the belt/pulley contact, see the Belt Pulley block reference page.

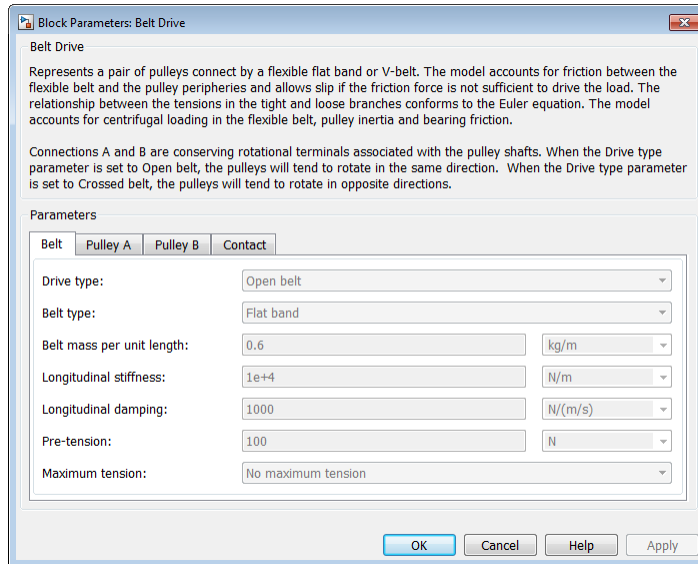
Connections A and B are the conserving mechanical rotational ports associated with the shafts of pulleys A and B, respectively.

## Assumptions and Limitations

- The pulleys do not translate.
- The friction coefficient and friction velocity threshold between the belt and each of the pulleys is the same.

## Dialog Box and Parameters

The dialog box includes three tabs. Use the **Belt** pane to specify belt characteristics.



### Drive type

Select the type of belt drive that you want to model.

- Open belt — (Default)
- Crossed belt

The schematic in this page shows the two drive types.

### Belt type

Type of flexible element modeled.

- Flat band — (Default)
- V-belt— Exposes the **V-belt sheave angle** and **Number of V-belts** parameters.

## **V-belt sheave angle**

Sheave angle of the V-belt. Default is 30 deg.

The **V-belt sheave angle** parameter is visible only when the **Belt type** parameter is V-belt.

## **Number of V-belts**

Number of parallel V-belts. Default is 1.

Non-integer values are rounded to the nearest integer. Increasing the number of belts increases the friction force, effective mass per unit length, and maximum allowable tension.

The **Number of V-belts** parameter is visible only when the **Belt type** parameter is V-belt.

## **Belt mass per unit length**

Linear density of each belt. Default is 0.6 kg/m.

The model uses this to calculate the centrifugal loading.

## **Longitudinal stiffness**

The effective stiffness of the belt. Default is  $1e+4$  N/m.

## **Longitudinal damping**

Effective damping of the belt. Default is 1000 N/(m/s).

## **Pre-tension**

Tension in the belt when the belt and pulleys are at rest. A positive value indicates that both pulley branches are initially in tension. A negative value indicates that the branches are initially slack. The default value is 100 N.

## **Maximum tension**

Specifies whether the block throws an assertion when the belt tension is too large.

- No maximum tension — (Default)



- Specify maximum tension— Throws an assertion when the belt tension is too high. Exposes the **Belt maximum tension** parameter.

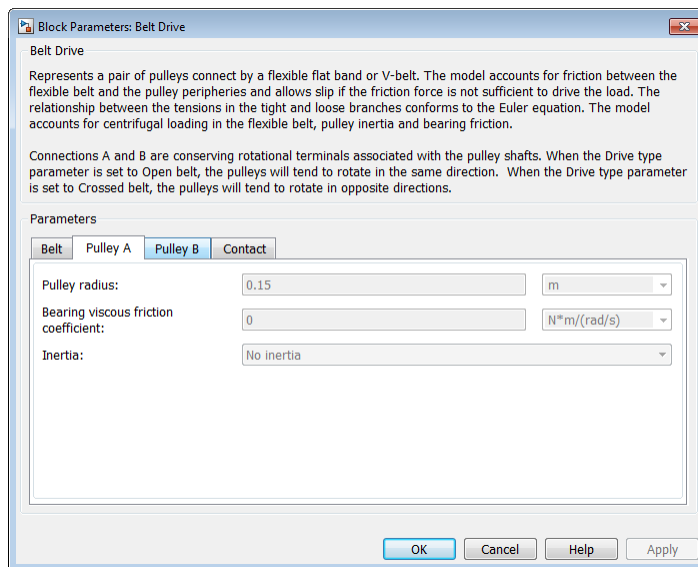
## Belt maximum tension

Maximum allowable tension for each belt. Default is  $1e+5$  N.

When the tension on either end of the belt meets or exceeds this value, the simulation ends with an error.

The **Belt maximum tension** parameter is visible only when the **Maximum tension** parameter is **Specify maximum tension**.

Select the **Pulley A** tab to specify the characteristics of pulley attached to port A, and the **Pulley B** tab to specify the characteristics of pulley attached to port B.



## Pulley radius

Radius of the pulley. Default is 0.15m for pulley A and 0.3 m for pulley B.

## **Bearing viscous friction coefficient**

Viscous friction associated with the bearings that hold the axis of the pulley. Default is 0 N\*m/(rad/s) for pulley A and 0.001 N\*m/(rad/s) for pulley B.

## **Inertia**

Specifies whether the block models rotational inertia of the pulley.

- No inertia — (Default)
- Specify inertia and initial velocity— Models rotational inertia. Exposes the **Pulley inertia** and **Pulley initial velocity** parameters.

## **Pulley inertia**

Rotational inertia of the pulley. Default is 0.01 kg\*m<sup>2</sup> for pulley A, and 0.03 kg\*m<sup>2</sup> for pulley B.

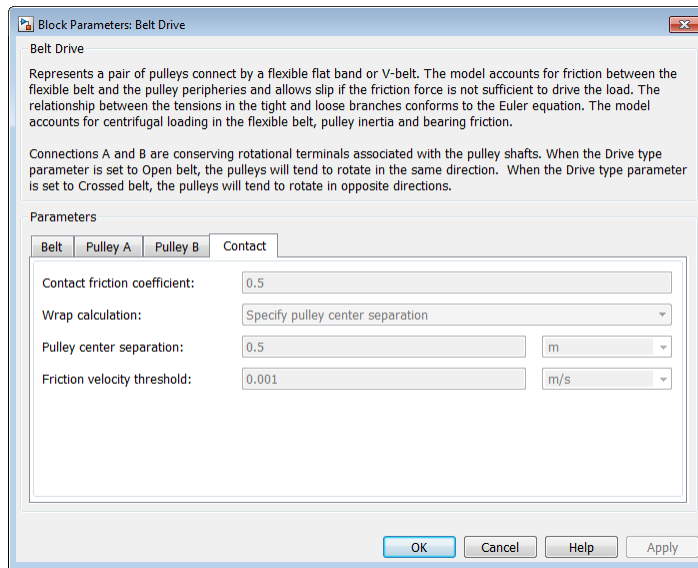
The **Pulley inertia** parameter is visible only when the **Inertia** parameter is Specify inertia and initial velocity.

## **Pulley initial velocity**

Initial rotational velocity of the pulley. Default is 0 rad/s.

The **Pulley initial velocity** parameter is visible only when the **Inertia** parameter is Specify inertia and initial velocity.

Select the **Contact** tab to specify contact characteristics.



## Contact friction coefficient

Coulomb friction coefficient between the belt and the pulley surface. Default is 0.5.

## Wrap calculations

Specifies whether the block calculates the wrap angle of the belt on the pulleys using the pulley radii, center separation, and drive type or if the wrap angles are specified directly.

- **Specify pulley center separation** — (Default) The block calculates the wrap angle of the belt on the pulleys using the pulley radii, center separation, and drive type.

Exposes the **Pulley center separation** parameter.

- **Specify wrap angles** — Uses the wrap angles that you specify.

Exposes the **Pulley A wrap angle** and **Pulley B wrap angle** parameters.

# Belt Drive

---

## **Pulley center separation**

Distance between the centers of the pulleys. Default is 0.5 m.

The **Pulley center separation** parameter is visible only when the **Wrap calculations** parameter is Specify pulley center separation.

## **Pulley A wrap angle**

Angle of contact between the belt and pulley attached to port A. Default is 145 deg.

The **Pulley A wrap angle** parameter is visible only when the **Wrap calculations** parameter is Specify wrap angles.

## **Pulley B Wrap angle**

Angle of contact between the belt and pulley attached to port B. Default is 215 deg.

The **Pulley B wrap angle** parameter is visible only when the **Wrap calculations** parameter is Specify wrap angles.

## **Friction velocity threshold**

Relative velocity required for peak kinetic friction in the contact. The friction velocity threshold improves the numerical stability of the simulation by ensuring that the force is continuous when the direction of the velocity changes. Default is 0.001 m/s.

## **Ports**

<b>Port</b>	<b>Description</b>
A	Conserving rotational port associated with the shaft of pulley A
B	Conserving rotational port associated with the shaft of pulley B

## **See Also**

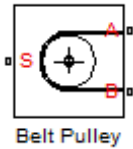
Chain Drive, Belt Pulley

“Rotational-Translational Couplings”.

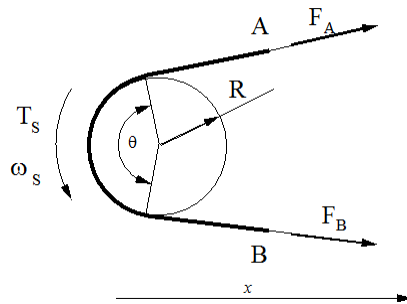
**Purpose** Flexible belt wrapped around a pulley

**Library** Couplings & Drives

**Description**



The block represents a pulley wrapped with a flexible flat band or V-shaped belt. The model accounts for friction between the flexible belt and the pulley periphery. If the friction force is not sufficient to drive the load, the model allows slip. The relationship between the tensions in the tight and loose branches conforms to the Euler equation. The model accounts for centrifugal loading in the flexible belt, pulley inertia, and bearing friction. The sign convention for the model is as shown.



Pulley rotation in the positive direction causes branch B to move in the positive direction and branch A to move in the negative direction. The model assumes that the belt is noncompliant along its length, and that the belt exhibits no resistance to motion when flexed around the pulley.

The Belt Pulley block uses the following equations to calculate friction force:

$$V_{rel} = V_A + \omega_S * R,$$

$$F_C = \rho * V_B^2,$$

$$F_{fr} = \begin{cases} F_B - F_C = (F_A - F_C) * \exp(f * \theta), & \text{if } V_{rel} \geq 0 \\ F_A - F_C = (F_B - F_C) * \exp(f * \theta), & \text{if } V_{rel} < 0 \end{cases}$$

$$V_A = -V_B$$

# Belt Pulley

---

- $V_{\text{rel}}$  is the relative velocity between the belt and pulley periphery.
- $V_A$  is the Branch A linear velocity.
- $V_B$  is the Branch B linear velocity.
- $\omega_S$  is the pulley angular velocity.
- $R$  is the pulley radius.
- $F_C$  is the belt centrifugal force.
- $\rho$  is the belt linear density.
- $F_{\text{fr}}$  is the friction force between the pulley and the belt.
- $F_A$  is the force acting along branch A.
- $F_B$  is the force acting along branch B.
- $f$  is the friction coefficient.
- $\theta$  is the contact wrap angle.

For a flat band, specify the value of  $f$  directly in the block parameters dialog box. For a V-belt, the model calculates the value using this equation:

$$f' = \frac{f}{\sin\left(\frac{\phi}{2}\right)}$$

- $f'$  is the effective friction coefficient for a V-belt.
- $\Phi$  is the V-belt sheave angle.

The idealization of the discontinuity at  $V_{\text{rel}} = 0$  is both difficult for the solver to resolve and not completely physically accurate. To alleviate this issue, the friction coefficient is assumed to change its value as a function of the relative velocity, with the following relationship:

$$\mu = -f * \tanh\left(4 * \frac{V_{rel}}{V_{thr}}\right)$$

- $\mu$  is the instantaneous value of the friction coefficient.
- $f$  is the steady-state value of the friction coefficient.
- $V_{thr}$  is the friction velocity threshold.

The friction velocity threshold controls the width of the region within which the friction coefficient changes its value from zero to a steady-state maximum. The friction velocity threshold specifies the velocity at which the hyperbolic tangent equals 0.999. The smaller the value, the steeper is the change of  $\mu$ .

This parameter uses the following equation to calculate the friction force:

$$F_{fr} = F_A - F_C = (F_B - F_C) * \exp(\mu * \theta)$$

The resulting torque on the pulley is given by the following equation:

$$T_S = (F_B - F_A) * R - \omega_S * b$$

- $T_S$  is the pulley torque.
- $b$  is the bearing viscous damping.

The equations that this model uses are effective for transmitting power to either branch of the belt or to and from the pulley. The tight and loose branch use the same calculation. Without sufficient tension, the frictional force is not enough to transmit power between the pulley and belt.

The model is valid when both ends of the belt are in tension. An optional warning can display at the MATLAB® command line when either belt end loses tension. When assembling a model, ensure that

# Belt Pulley

tension is maintained throughout the simulation. Interpret the results with this in mind.

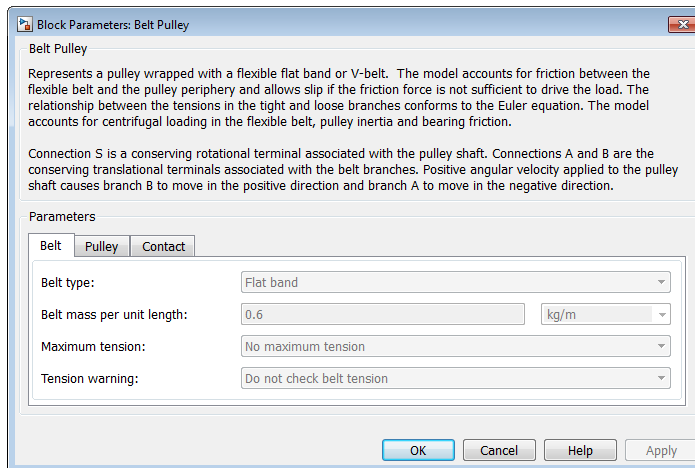
Connection S is a conserving rotational terminal associated with the pulley shaft. Connections A and B are the conserving mechanical translational ports associated with the belt ends. The sign convention is such that a positive rotation in port S tends to give a negative translation for port A and a positive translation for port B.

## Assumptions and Limitations

- The model does not account for compliance along the length of the belt.
- Both belt ends maintain adequate tension throughout the simulation.

## Dialog Box and Parameters

The dialog box includes three tabs. Use the **Belt** pane to specify belt characteristics.



## Belt type

Type of flexible element modeled.

- Flat band — (Default)



- **V-belt**— Exposes the **V-belt sheave angle** and **Number of V-belts** parameters.

## **V-belt sheave angle**

Sheave angle of the V-belt. Default is 30 deg.

The **V-belt sheave angle** parameter is visible only when the **Belt type** parameter is V-belt.

## **Number of V-belts**

Number of parallel V-belts. Default is 1.

Non-integer values are rounded to the nearest integer. Increasing the number of belts increases the friction force, effective mass per unit length, and maximum allowable tension. Non-integer values are rounded to the nearest integer.

The **Number of V-belts** parameter is visible only when the **Belt type** parameter is V-belt.

## **Belt mass per unit length**

Linear density of each belt. Default is 0.6 kg/m.

The model uses this parameter to calculate the centrifugal loading.

## **Maximum tension**

Specifies whether the block throws an assertion when the belt tension is too large.

- **No maximum tension** — (Default)
- **Specify maximum tension** — Throws an assertion when the belt tension is too high. Exposes the **Belt maximum tension** parameter.

## **Belt maximum tension**

Maximum allowable tension for each belt. Default is 1e+5 N.

When the tension on either end of the belt meets or exceeds this value, the simulation ends with an error.

# Belt Pulley

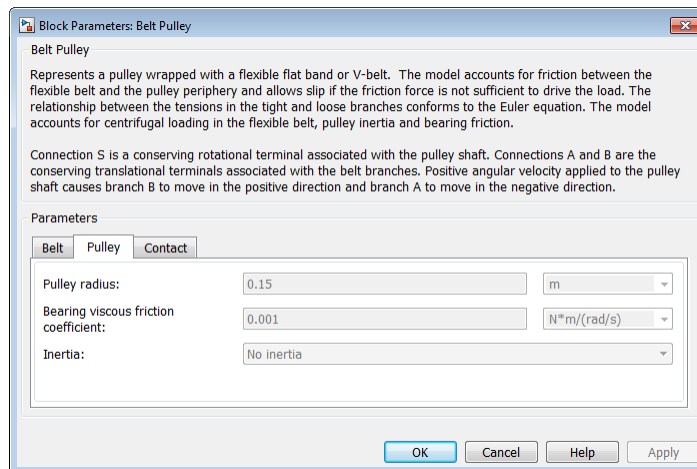
The **Belt maximum tension** parameter is visible only when the **Maximum tension** parameter is Specify maximum tension.

## Tension Warning

Specifies whether the block writes a warning at the MATLAB command line when the tension at either end of the belt falls below zero.

- Do not check belt tension — (Default)
- Warn if either belt end loses tension

Select the **Pulley** tab to specify the pulley characteristics.



## Pulley radius

Radius of the pulley. Default is 0.15 m.

## Bearing viscous friction coefficient

Viscous friction associated with the bearings that hold the axis of the pulley. Default is 0.001 N\*m/(rad/s).

## Inertia

Specifies whether the block models rotational inertia of the pulley.

- No inertia — (Default)
- Specify inertia and initial velocity — Models rotational inertia. Exposes the **Pulley inertia** and **Pulley initial velocity** parameters.

## Pulley inertia

Rotational inertia of the pulley. Default is  $0.01 \text{ kg}\cdot\text{m}^2$ .

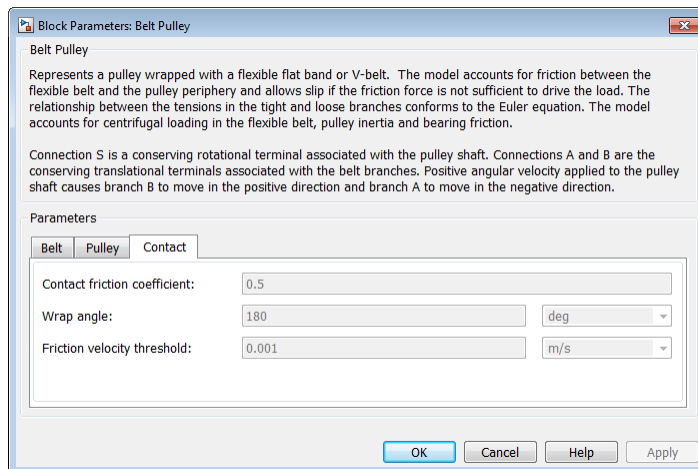
The **Pulley inertia** parameter is visible only when the **Inertia** parameter is Specify inertia and initial velocity.

## Pulley initial velocity

Initial rotational velocity of the pulley. Default is  $0 \text{ rad/s}$ .

The **Pulley initial velocity** parameter is visible only when the **Inertia** parameter is Specify inertia and initial velocity.

Use the **Contact** pane to specify contact characteristics.



## Contact friction coefficient

Coulomb friction coefficient between the belt and the pulley surface. Default is 0.5.

# Belt Pulley

---

## Wrap angle

Angle of contact between the belt and pulley. Default is 180 deg.

## Friction velocity threshold

Relative velocity required for peak kinetic friction in the contact. The friction velocity threshold improves the numerical stability of the simulation by ensuring that the force is continuous when the direction of the velocity changes. Default is 0.001 m/s.

## Ports

Port	Description
S	Conserving rotational port associated with the pulley shaft
A	Conserving translational port associated with belt end A
B	Conserving translational port associated with belt end B

## Example

The `sdl_power_window` SimDriveline™ example model shows the use of the Belt Pulley block.

## See Also

Rope Drum, Belt Drive

Wheel and Axle (in the Simscape™ documentation)

“Rotational-Translational Couplings”.

**Purpose** Roller chain wrapped around two sprockets

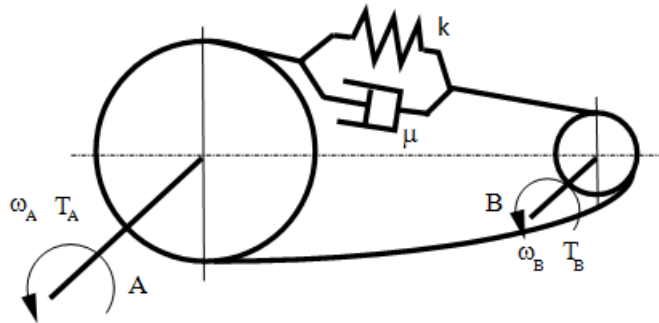
**Library** Couplings & Drives

**Description**



The block represents a drive built of two sprockets that are connected by a roller chain. The model accounts for chain slack and can be backdriven. The chain has damping relative to its linear velocity and stiffness that counteracts stretching when the chain is taut.

When one end of the chain is taut, the chain drive behaves as shown.



The damping effect in the chain always acts on the relative tangential velocity of the sprockets. The stiffness effect in the chain acts only when either the upper or lower leg of the chain is taut.

These equations describe the chain drive.

$$\dot{x} = \omega_A R_A - \omega_B R_B,$$

$$F = \begin{cases} -(x - S/2)k - \dot{x} * b, & x > S/2 \\ -\dot{x} * b, & S/2 \geq x \geq -S/2 \\ -(x + S/2)k - \dot{x} * b, & x < -S/2 \end{cases}$$

$$T_A = -F * R_A,$$

$$T_B = F * R_B$$

# Chain Drive

---

- $x$  is the chain linear displacement.
- $R_A$  is the radius of sprocket A.
- $R_B$  is the radius of sprocket B.
- $\omega_A$  is the angular velocity of sprocket A.
- $\omega_B$  is the angular velocity of sprocket B.
- $F$  is the chain tensile force.
- $S$  is the chain slack length.
- $k$  is the chain stiffness.
- $\mu$  is the chain damping.
- $T_A$  is the torque acting on sprocket A.
- $T_B$  is the torque acting on sprocket B.

You can add bearing viscous friction to either sprocket.

If you specify a maximum chain tension, then the simulation ends with an error when this value exceeds that maximum during runtime.

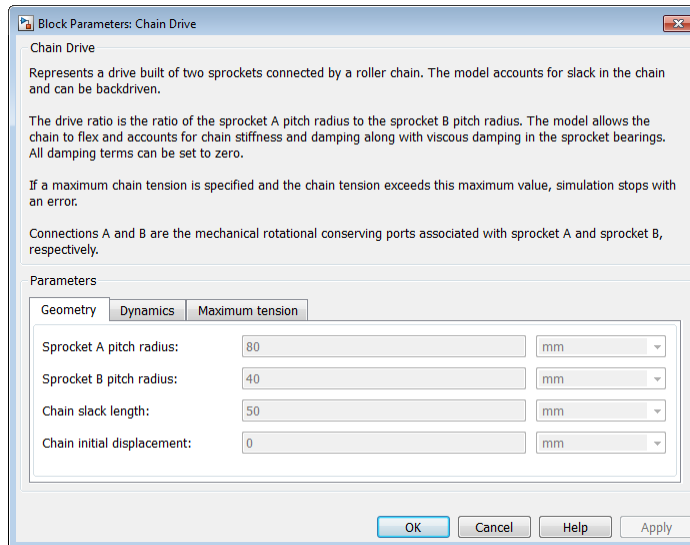
Connections A and B are conserving rotational ports that are associated with sprockets A and B, respectively.

## **Assumptions and Limitations**

- The sprocket tooth ratio is equal to the sprocket pitch radius ratio.
- Chain inertia is negligible.

## Dialog Box and Parameters

Use the **Geometry** pane to specify sprocket characteristics such as the pitch radius and chain slack length.



### Sprocket A pitch radius

The pitch radius of sprocket A. Default is 80 mm.

### Sprocket B pitch radius

The pitch radius of sprocket B. Default is 40 mm.

### Chain slack length

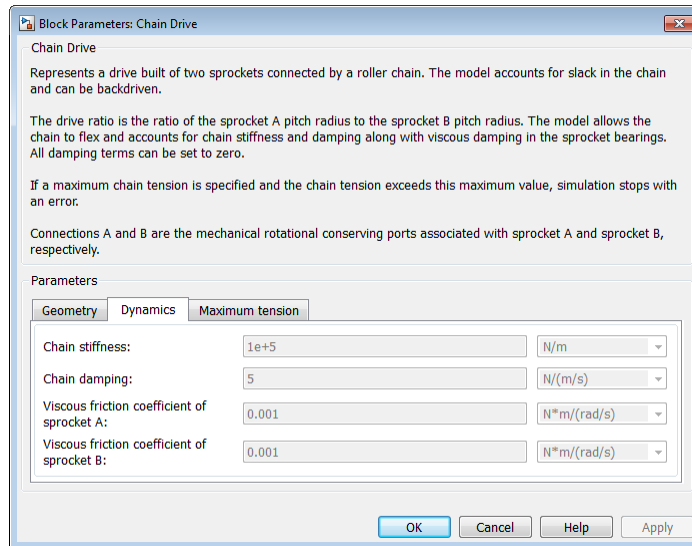
The slack length of the chain is the length that the chain can move before becoming taut. If one sprocket is held in place while the chain is taut, then the slack length is the tangential distance that the other sprocket must rotate before the other chain end is taut. Default is 50 mm.

# Chain Drive

## Chain initial displacement

The initial displacement of the chain. A value of 0 indicates that the chain can displace by half the slack length in either direction before it is taut. Default is 0 mm.

Select the **Dynamics** tab to specify stiffness, damping, and viscous coefficients.



## Chain stiffness

The longitudinal stiffness of the chain. This force is required to stretch the taut end of the chain. Default is 1e+5 N/m.

## Chain damping

The longitudinal damping of the chain. Default is 5 N/(m/s).

## Viscous friction coefficient of sprocket A

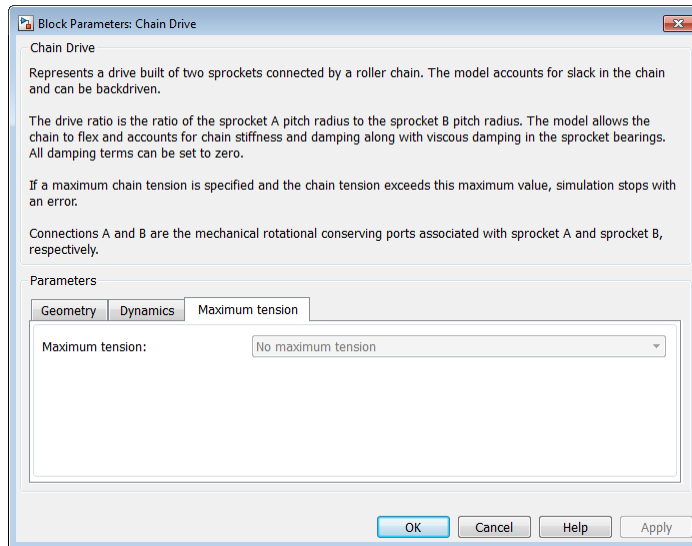
Viscous friction associated with the bearings that hold sprocket A. Default is 0.001 N\*m/(rad/s).



## Viscous friction coefficient of sprocket B

Viscous friction associated with the bearings that hold sprocket B. Default is  $0.001 \text{ N}\cdot\text{m}/(\text{rad}/\text{s})$ .

Select the **Maximum tension** tab to specify the maximum tension.



## Maximum tension

Specifies whether the block throws an assertion when the chain tension is too large.

- **No maximum tension** — (Default) Does not throw an assertion.
- **Specify maximum tension** — Throws an assertion when the chain tension meets or exceeds the value of the **Chain maximum tension** parameter.

## Chain maximum tension

Maximum allowable tension for the chain. If the chain tension meets or exceeds this value, the simulation ends with an error. Default is  $1 \text{e}+6 \text{ N}$ .

# Chain Drive

---

The **Chain maximum tension** parameter is visible only when the **Maximum tension** is Specify maximum tension.

## Ports

Port	Description
A	Conserving rotational port associated with sprocket A
B	Conserving rotational port associated with sprocket B

## Example

The `sdl_sheet_metal_feeder` SimDriveline example model shows the use of the Chain Drive block.

## See Also

Simple Gear

Variable Ratio Transmission

“Gear Coupling Control with Clutches”

# Compound Planetary Gear

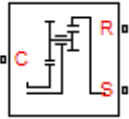
## Purpose

Compound planetary gear set of carrier, ring, planet, and sun wheels with adjustable gear ratios and friction losses

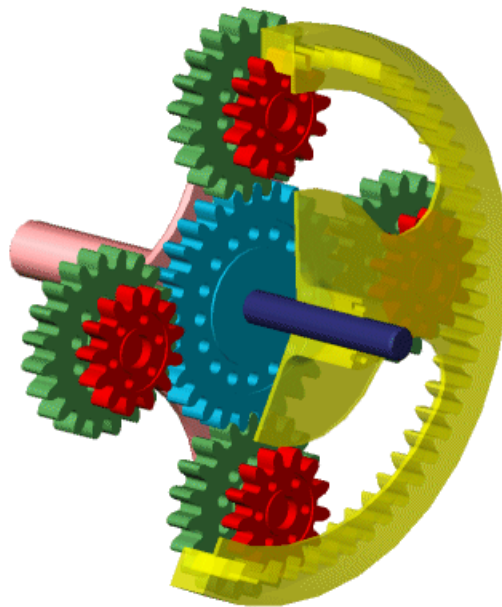
## Library

Gears

## Description



The Compound Planetary Gear block represents a set of carrier, sun, planet, and ring gear wheels. The planetary shaft has two gear wheels with different radii meshing with the ring and the sun, respectively. The ring and planet corotate with one fixed gear ratio. The planet and sun corotate with another fixed gear ratio. For this reason, a compound planetary gear is also called a *dual-ratio* planetary gear. For model details, see “Compound Planetary Gear Model” on page 1-33.



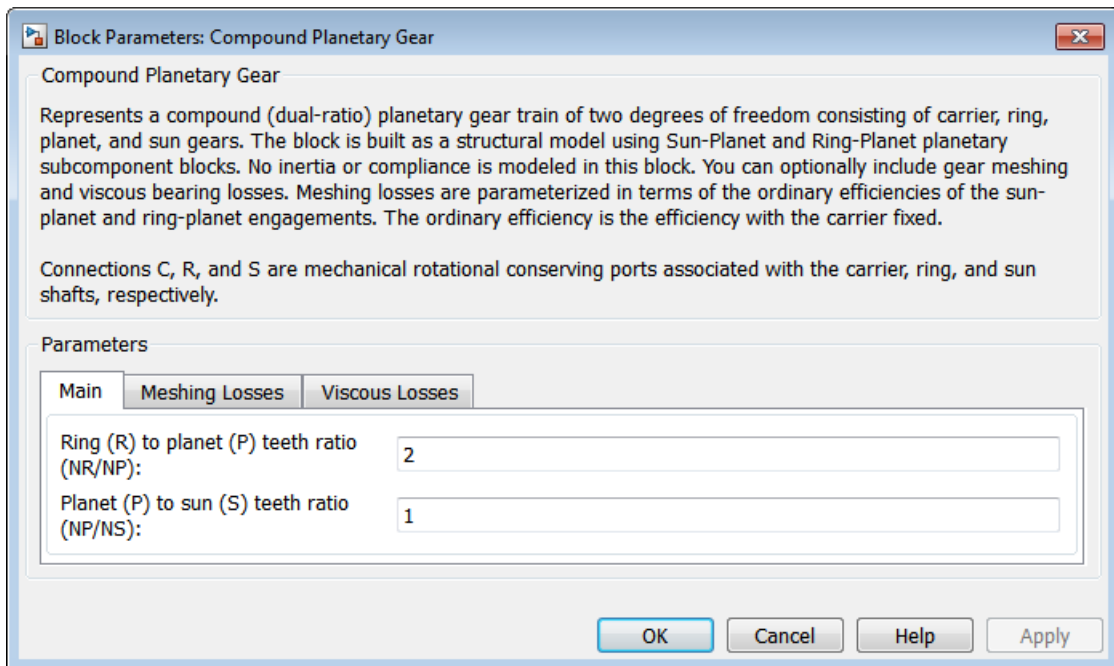
**Compound Planetary Gear Set**

# Compound Planetary Gear

## Ports

C, R, and S are rotational conserving ports representing, respectively, the carrier, ring, and sun gear wheels.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

## Main

Main	Meshing Losses	Viscous Losses
Ring (R) to planet (P) teeth ratio (NR/NP):	<input type="text" value="2"/>	
Planet (P) to sun (S) teeth ratio (NP/NS):	<input type="text" value="1"/>	

### Ring (R) to planet (P) teeth ratio (NR/NP)

Fixed ratio  $g_{RP}$  of the ring gear to the planet gear. The gear ratio must be strictly greater than 1. The default is 2.

### Planet (P) to sun (S) teeth ratio (NP/NS)

Fixed ratio  $g_{PS}$  of the planet gear to the sun gear. The gear ratio must be strictly positive. The default is 1.

## Meshing Losses

Main	Meshing Losses	Viscous Losses
Friction model:	<input type="text" value="No meshing losses - Suitable for HIL simulation"/>	

### Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses — Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

# Compound Planetary Gear

## Constant Efficiency

Main	Meshing Losses	Viscous Losses
Friction model:	Constant efficiency	
Sun-planet and ring-planet ordinary efficiencies:	[ 0.96 0.98 ]	
Sun-carrier and planet-carrier velocity thresholds:	[ 0.01 0.01 ]	rad/s

### Sun-planet and ring-planet ordinary efficiencies

Vector of torque transfer efficiencies  $[\eta_{SP} \ \eta_{RP}]$  for sun-planet and ring-carrier gear wheel pair meshings, respectively.

### Sun-carrier and planet-carrier velocity thresholds

Vector of absolute angular velocity thresholds  $\omega_{th}$  above which full efficiency loss is applied, for sun-carrier and planet-carrier velocities  $[\omega_s \ \omega_p]$ , respectively.

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses
Sun-carrier and planet-carrier viscous friction coefficients:	[ 0 0 ]	N*m/(rad/s)

### Sun-carrier and planet-carrier viscous friction coefficients

Vector of viscous friction coefficients  $[\mu_s \ \mu_p]$  for the sun-carrier and planet-carrier gear motions, respectively. The default is [0 0].

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Compound Planetary Gear Model

### Ideal Gear Constraints and Gear Ratios

Compound Planetary Gear imposes two kinematic and two geometric constraints on the three connected axes and the fourth, internal wheel (planet):

$$r_C \omega_C = r_S \omega_S + r_{P1} \omega_P, \quad r_C = r_S + r_{P1},$$

$$r_R \omega_R = r_C \omega_C + r_{P2} \omega_P, \quad r_R = r_C + r_{P2}.$$

The ring-planet gear ratio  $g_{RP} = r_R/r_{P2} = N_R/N_{P2}$  and the planet-sun gear ratio  $g_{PS} = r_{P1}/r_S = N_{P1}/N_S$ .  $N$  is the number of teeth on each gear. In terms of these ratios, the key kinematic constraint is:

$$(1 + g_{RP} g_{PS}) \omega_C = \omega_S + g_{RP} g_{PS} \omega_R.$$

The four degrees of freedom reduce to two independent degrees of freedom. The gear pairs are (1,2) = (P2,R) and (S,P1).

### Warning

**The gear ratio  $g_{RP}$  must be strictly greater than one.**

The torque transfers are:

$$g_{RP} \tau_{P2} + \tau_R - \tau_{\text{loss}}(\text{P2,R}) = 0, \quad g_{PS} \tau_S + \tau_{P1} - \tau_{\text{loss}}(\text{S,P1}) = 0,$$

with  $\tau_{\text{loss}} = 0$  in the ideal case.

### Nonideal Gear Constraints and Losses

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . See “Model Gears with Losses”.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

# Compound Planetary Gear

---

## **See Also**

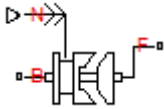
Planetary Gear, Ravigneaux Gear, Ring-Planet, Sun-Planet  
“Represent and Transfer Driveline Motion and Torque”



**Purpose** Cone clutch or brake with kinetic and static friction and controllable normal force

**Library** Clutches

## Description



A friction clutch transfers torque between two driveline axes by coupling them with friction. The Cone Clutch block models a friction clutch based on conical surfaces in contact, with kinetic friction and static (locking) friction acting on the two axes.

A cone clutch is typically used to implement a synchromesh gearbox, by allowing you to synchronize shaft speeds before engaging two gears. For model details, see “Cone Clutch Model” on page 1-40.

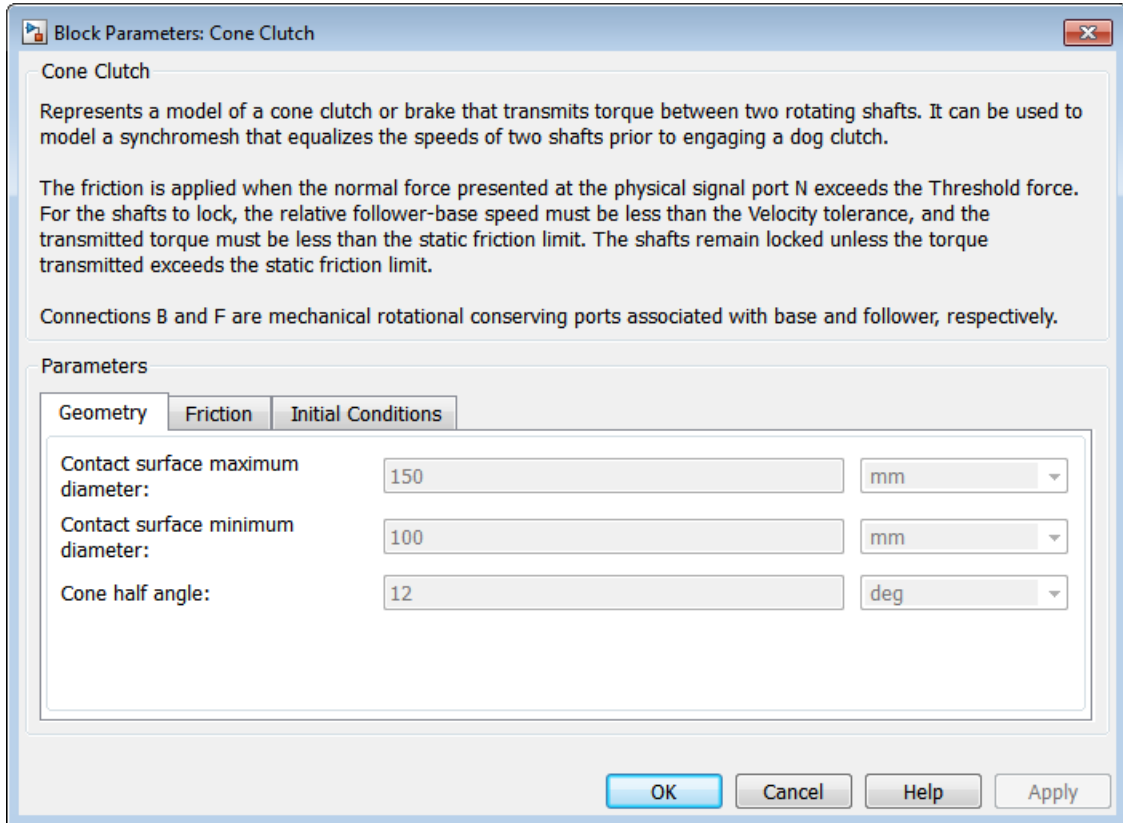
## Ports

B and F are rotational conserving ports representing, respectively, the clutch input (base) and output (follower) driveshaft axes. The clutch motion is measured as the *slip*  $\omega = \omega_F - \omega_B$ , the angular velocity of follower relative to base.

The clutch requires a physical signal input N that represents the normal force (in newtons) applied between the friction surfaces in contact. This signal should be positive or zero. A signal N less than zero is interpreted as zero.

# Cone Clutch

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

## Dimensions

Geometry	Friction	Initial Conditions
Contact surface maximum diameter:	<input type="text" value="150"/>	<input type="text" value="mm"/>
Contact surface minimum diameter:	<input type="text" value="100"/>	<input type="text" value="mm"/>
Cone half angle:	<input type="text" value="12"/>	<input type="text" value="deg"/>

### Contact surface maximum diameter

The outer conical diameter  $d_o$ . The default is 150.

From the drop-down list, choose units. The default is millimeters (mm).

### Contact surface minimum diameter

The outer conical diameter  $d_i$ . The default is 100.

From the drop-down list, choose units. The default is millimeters (mm).

### Cone half angle

The half opening angle  $\alpha$  of the cone geometry. The default is 12.

From the drop-down list, choose units. The default is degrees (deg).

# Cone Clutch

## Friction

Geometry	Friction	Initial Conditions
Friction model: Fixed kinetic friction coefficient		
Kinetic friction coefficient: 0.3		
Static friction coefficient: 0.35		
Velocity tolerance: 0.001 rad/s		
Threshold force: 1 N		

### Friction model

Select how to model the dimensionless Coulomb kinetic friction coefficient  $k_K$  across the clutch when the clutch is slipping. The default is Fixed kinetic friction coefficient.

- Fixed kinetic friction coefficient — Model Coulomb kinetic friction in terms of a constant kinetic friction coefficient.

### Fixed Kinetic Friction Coefficient

Friction model:	Fixed kinetic friction coefficient
Kinetic friction coefficient:	0.3

### Kinetic friction coefficient

The dimensionless kinetic friction coefficient  $k_K$  applied to the normal force across clutch. The default is 0.3.

- Table lookup kinetic friction coefficient — Model Coulomb kinetic friction in terms of a kinetic friction coefficient lookup function defined at discrete relative velocity values. If you select this option, the panel changes from its default.

## Table Lookup Kinetic Friction Coefficient

Friction model:	Table lookup kinetic friction coefficient	
Kinetic friction coefficient relative velocity vector:	0.1 0.2 0.3 0.4 0.5 1 2 5 10 100 1000 ]	rad/s
Kinetic friction coefficient vector:	[ 0.29 0.28 0.26 0.22 0.21 0.2 0.2 0.19 0.19 0.19 0.19 ]	
Friction coefficient interpolation method:	Linear	
Friction coefficient extrapolation method:	From last 2 points	

### Kinetic friction coefficient relative velocity vector

Vector of clutch slip values  $\omega$  at which to define kinetic friction coefficient.

From the drop-down list, choose units. The default is radians/second (rad/s).

### Kinetic friction coefficient vector

Vector of kinetic friction coefficient values  $k_K$ .

### Friction coefficient interpolation method

Method to interpolate kinetic friction coefficient function between discrete relative velocity values within the range of definition. The default is Linear.

### Friction coefficient extrapolation method

Method to extrapolate kinetic friction coefficient function outside the range of definition. The default is From last 2 points.

### Static friction coefficient

Dimensionless Coulomb static friction coefficient  $k_S$  applied to the normal force across the clutch when the clutch is locked. Must be larger than  $k_K$ . The default is 0.35.

# Cone Clutch

---

## Velocity tolerance

Sets the minimum angular speed  $\omega_{\text{Tol}}$  above which the clutch cannot lock. Below this speed, the clutch can lock. The default is  $1\text{e-}3$ .

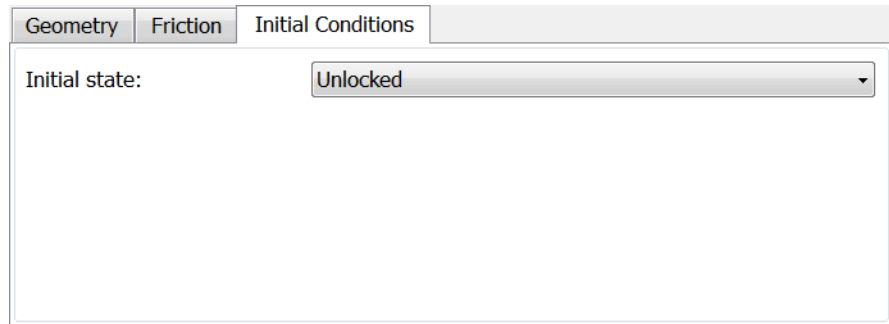
From the drop-down list, choose units. The default is radians/second (rad/s).

## Threshold force

The minimum normal force  $F_{\text{th}}$  needed to engage the clutch. This lower bound applies to the physical signal input normal force  $N$ . If  $N$  falls below this value, the clutch applies no normal force. The default is 1.

From the drop-down list, choose units. The default is newtons (N).

## Initial Conditions



The screenshot shows a software interface with three tabs: 'Geometry', 'Friction', and 'Initial Conditions'. The 'Initial Conditions' tab is active. Below the tabs, there is a label 'Initial state:' followed by a dropdown menu. The dropdown menu is currently set to 'Unlocked'.

## Initial state

From the drop-down list, choose whether the clutch starts simulation in a locked or unlocked state. The default is Unlocked.

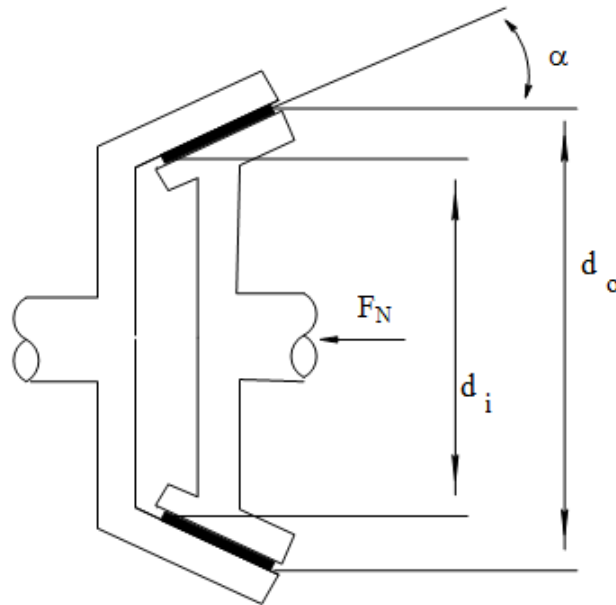
## Cone Clutch Model

The Cone Clutch is based on the Fundamental Friction Clutch. For the complete friction clutch model, consult the Fundamental Friction Clutch block reference page. This section discusses the specialized model implemented in the Cone Clutch.

When you apply a normal force  $F_N$ , the Cone Clutch block can apply two kinds of friction to the driveline motion, kinetic and static. The clutch applies kinetic friction torque only when one driveline axis is spinning relative to the other driveline axis. The clutch applies static friction torque when the two driveline axes lock and spin together. The block iterates through multistep testing to determine when to lock and unlock the clutch.

## Clutch Geometry and Variable Summary

The figure displays the geometry of the conical friction surfaces and normal force. The table summarizes the cone clutch variables.



**Clutch Geometry**

# Cone Clutch

## Clutch Variables

Symbol	Definition	Significance
$\omega$	Relative angular velocity (slip)	$\omega_F - \omega_B$
$\omega_{Tol}$	Slip tolerance for clutch locking	<i>See the following model</i>
$F_N$	Normal force applied to conical surfaces	Normal force applied, if greater than threshold: $F_N > F_{th}$
$a$	Cone half-angle	<i>See the preceding figure</i>
$r_{eff}$	Effective torque radius	Effective moment arm of clutch friction force
$k_K$	Kinetic friction coefficient	Dimensionless coefficient of kinetic friction of conical friction surfaces. Function of $\omega$ .
$k_S$	Static friction coefficient	Dimensionless coefficient of static friction of conical friction surfaces.
$\tau_K$	Kinetic friction torque	<i>See the following model</i>
$\tau_S$	Static friction torque limit	(static friction peak factor) · (kinetic friction torque for $\omega \rightarrow 0$ ) ( <i>See the following model</i> )

## Relation to Fundamental Friction Clutch

The Cone Clutch is based on the Fundamental Friction Clutch. Instead of requiring the kinetic and static friction limit torques as input signals, the Cone Clutch calculates the kinetic and static friction from the clutch parameters and the input normal force signal  $F_N$ . See the Fundamental Friction Clutch reference page for more information about the friction clutch.



## Kinetic Friction

The kinetic friction torque is the product of four factors:

$$\tau_K = k_K \cdot F_N \cdot r_{\text{eff}} \cdot \text{sgn}(\omega) .$$

The kinetic friction torque opposes the relative slip and is applied with an overall minus sign. It changes sign when  $\omega$  changes sign.

You specify the *kinetic friction coefficient*  $k_K$  as either a constant or a tabulated discrete function of relative angular velocity  $\omega$ . The tabulated function is assumed to be symmetric for positive and negative values of the relative angular velocity, so that you need to specify  $k_K$  for positive values of  $\omega$  only.

The *effective torque radius*  $r_{\text{eff}}$  is the effective radius, measured from the driveline axis, at which the kinetic friction forces are applied at the frictional surfaces. It is related to the geometry of the conical friction surface geometry by:

$$r_{\text{eff}} = \frac{1}{3 \sin \alpha} \frac{d_o^3 - d_i^3}{d_o^2 - d_i^2}$$

$d_o$  and  $d_i$  are the contact surface maximum and minimum diameters, respectively.

## Static Friction

The static friction limit is related to the kinetic friction, setting  $\omega$  to zero and replacing the kinetic with the static friction coefficient:

$$\tau_S = k_S \cdot F_N \cdot r_{\text{eff}} \geq 0 .$$

$k_S > k_K$ , so that the torque  $\tau$  needed across the clutch to unlock it by overcoming static friction is larger than the kinetic friction at the instant of unlocking, when  $\omega = 0$ .

The static friction limit defines symmetric *static friction torque limits* as:

$$\tau_S \equiv \tau_S^+ = -\tau_S^- .$$

# Cone Clutch

---

The range  $[\tau_s^-, \tau_s^+]$  is used by the Fundamental Friction Clutch.

## **Engagement and Locking Conditions**

The clutch engages (transmits torque) when the conical friction surfaces are subject to a positive normal force and generate kinetic friction:  $F_N > 0$  and  $\tau_K > 0$ .

The clutch locks if and only if it is engaged, and the slip is less than the velocity tolerance:  $|\omega| < \omega_{\text{tol}}$ .

## **Power Dissipated by the Clutch**

The power dissipated by the clutch is  $|\omega \tau_K|$ . The clutch dissipates power only if it is both slipping ( $\omega \neq 0$ ) and applying kinetic friction ( $\tau_K > 0$ ).

## **See Also**

Fundamental Friction Clutch

“Gear Coupling Control with Clutches”.

## **For More Information on Fundamental Friction Clutch**

The Disk Friction Clutch is based on the Fundamental Friction Clutch block. See the Fundamental Friction Clutch block reference page for more information about:

- “Clutch Friction” on page 1-130
- “Clutch Directionality” on page 1-129
- “Clutch Velocity Tolerance” on page 1-129
- “Clutch Initial State — Overriding Initial Default State by Manual Locking” on page 1-130
- “Clutch State, Transition, and Variable Summary” on page 1-131
- “Clutch States and Transitions” on page 1-134
- “Unlocked State: Kinetic Friction” on page 1-135
- “Locked State: Static Friction” on page 1-136
- “How the Friction Clutch Locks and Unlocks” on page 1-136

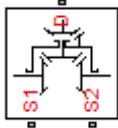
## Purpose

Differential gear set with adjustable driveshaft gear ratio and friction losses

## Library

Gears

## Description



The Differential block represents a differential gear that couples rotational motion about the longitudinal driveshaft axis to rotational motion about two lateral or side axes. The differential is composed of a simple gear and two symmetric sun-planet bevel gears. The simple gear comprises a differential crown gear attached to the carrier of one of the sun-planet bevel gears, plus a bevel gear attached to the driveshaft. For model details, see “Differential Gear Model” on page 1-49.

Any axis can be the input. In normal use, the longitudinal driveshaft is the input, and motion, torque, and power flow out through the side shafts. In general, the output axes have different angular velocities. The longitudinal motion is divided by the drive gear ratio that you specify and then split equally between the two lateral shafts.

Differential Crown Location Relative to Center-line	Side Axes Rotation Relative to Longitudinal Axis Rotation
Right	Same
Left	Opposite

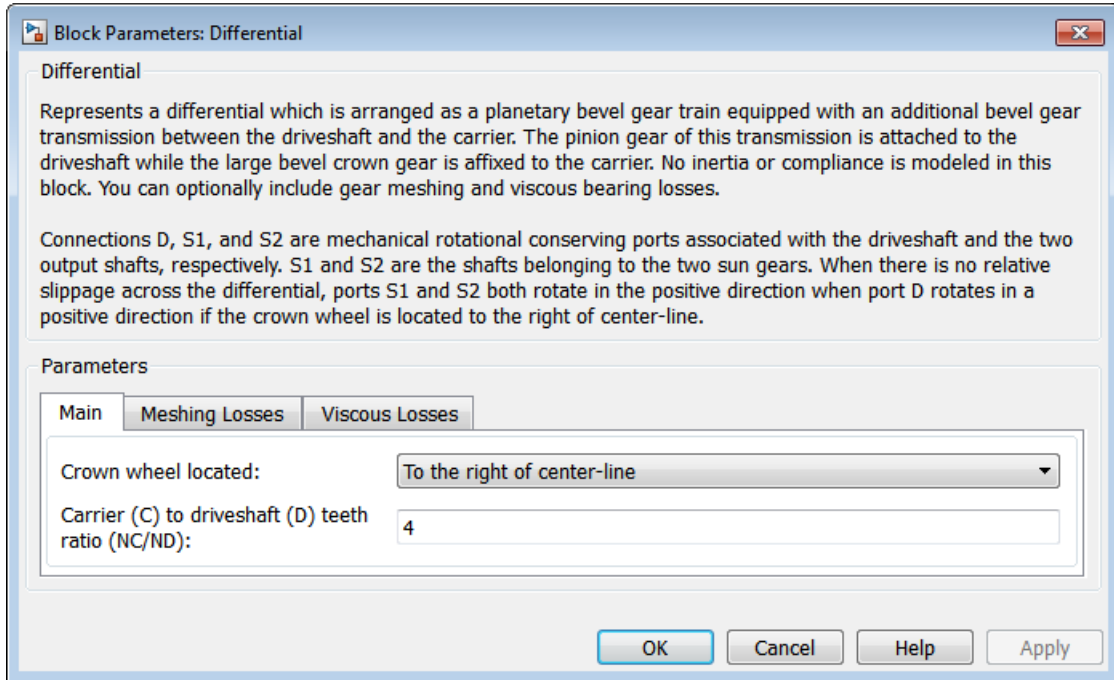
Differentials in drivelines often have a controllable clutch connecting the two output shafts. You can add this clutch control by connecting a clutch block to the Differential block.

## Ports

D, S1, and S2 are rotational conserving ports representing, respectively, the longitudinal driveshaft and the two side shafts.

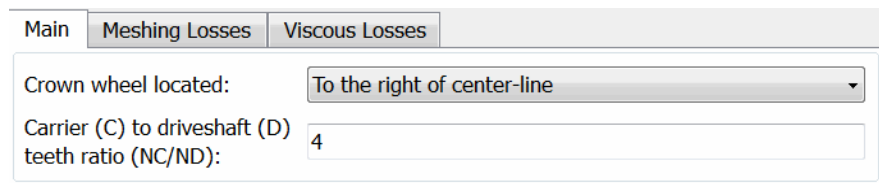
# Differential

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



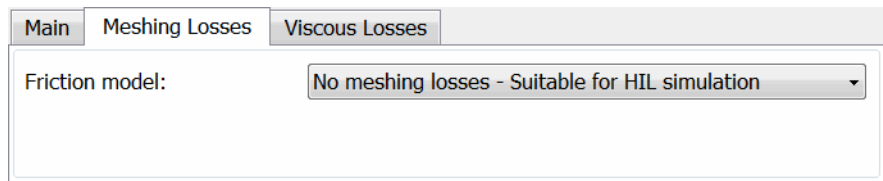
## Crown wheel located

Select the placement of the bevel crown gear with respect to the center-line of the gear assembly. The default is To the right of the center-line.

## Carrier (C) to driveshaft (D) teeth ratio (NC/ND)

Fixed ratio  $g_D$  of the carrier gear to the longitudinal driveshaft gear. The default is 4.

## Meshing Losses



Friction model: No meshing losses - Suitable for HIL simulation

## Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses — Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

## Constant Efficiency

Main	Meshing Losses	Viscous Losses
Friction model:	Constant efficiency	
Sun-sun and carrier-driveshaft ordinary efficiencies:	[ 0.85 0.92 ]	
Sun-carrier and driveshaft-casing velocity thresholds:	[ 0.01 0.01 ]	rad/s

### Sun-sun and carrier-driveshaft ordinary efficiencies

Vector of torque transfer efficiencies  $[\eta_{SS} \eta_D]$  for sun-sun and carrier-longitudinal driveshaft gear wheel pair meshings, respectively.

### Sun-carrier and driveshaft-casing velocity thresholds

Vector of absolute angular velocity thresholds  $\omega_{th}$  above which full efficiency loss is applied, for sun-carrier and longitudinal driveshaft-casing velocities  $[\omega_S \omega_D]$ , respectively.

Choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses
Sun-carrier and driveshaft-casing viscous friction coefficients:	[ 0 0 ]	N*m/(rad/s)

## Sun-carrier and driveshaft-casing viscous friction coefficients

Vector of viscous friction coefficients  $[\mu_S \mu_D]$  for the sun-carrier and longitudinal driveshaft-casing gear motions, respectively. The default is  $[0 \ 0]$ .

From the drop-down list, choose units. The default is newton-meters/(radians/second) ( $N \cdot m / (rad/s)$ ).

## Differential Gear Model

### Ideal Gear Constraints and Gear Ratios

Differential imposes one kinematic constraint on the three connected axes:

$$\omega_D = \pm(1/2)g_D(\omega_{S1} + \omega_{S2}),$$

with the upper (+) or lower (–) sign valid for the differential crown to the right or left, respectively, of the center-line. The three degrees of freedom reduce to two independent degrees of freedom. The gear pairs are (1,2) = (S,S) and (C,D). C is the carrier.

The *sum* of the lateral motions is the transformed longitudinal motion. The *difference* of side motions  $\omega_{S1} - \omega_{S2}$  is independent of the longitudinal motion. The general motion of the lateral shafts is a superposition of these two independent degrees of freedom, which have this physical significance:

- One degree of freedom (longitudinal) is equivalent to the two lateral shafts rotating at the same angular velocity ( $\omega_{S1} = \omega_{S2}$ ) and at a fixed ratio with respect to the longitudinal shaft.
- The other degree of freedom (differential) is equivalent to keeping the longitudinal shaft locked ( $\omega_D = 0$ ) while the lateral shafts rotate with respect to each other in opposite directions ( $\omega_{S1} = -\omega_{S2}$ ).

The torques along the lateral axes,  $\tau_{S1}$  and  $\tau_{S2}$ , are constrained to the longitudinal torque  $\tau_D$  in such a way that the power flows into and out of the gear, less any power loss  $P_{loss}$ , sum to zero:

# Differential

---

$$\omega_{S1}\tau_{S1} + \omega_{S2}\tau_{S2} + \omega_D\tau_D - P_{\text{loss}} = 0 .$$

When the kinematic and power constraints are combined, the ideal case yields:

$$g_D\tau_D = 2(\omega_{S1}\tau_{S1} + \omega_{S2}\tau_{S2}) / (\omega_{S1} + \omega_{S2}) .$$

## Fundamental Sun-Planet Bevel Gear Constraints

The effective differential constraint is composed of two subconstraints.

- The first comes from the coupling of the two sun-planet bevel gears to the carrier:

$$(\omega_{S1} - \omega_C) / (\omega_{S2} - \omega_C) = -g_{SP2} / g_{SP1} .$$

- The second comes from the coupling of the carrier to the longitudinal driveshaft:  $\omega_D = \pm g_D \omega_C$ .

$g_{SP1} = r_{S1} / r_{P1}$  and  $g_{SP2} = r_{S2} / r_{P2}$  are the sun-planet gear ratios of the underlying sun-planet bevel gears, in terms of the radii  $r$  of the sun and planet gears. The Differential block is implemented with  $g_{SP1} = g_{SP2} = 1$ , leaving  $g_D$  free to adjust.

## Nonideal Gear Constraints and Losses

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . See “Model Gears with Losses”.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## Examples

These SimDriveline example models contain working examples of differential gears:

- `sdl_4wd_dynamics`
- `sdl_limited_slip_differential`



- `sdl_torsen_differential`

## **See Also**

Disk Friction Clutch, Simple Gear, Sun-Planet Bevel  
“Represent and Transfer Driveline Motion and Torque”

# Disk Friction Clutch

---

**Purpose** Disk friction clutch or brake with kinetic and static friction and controllable pressure

**Library** Clutches

## Description



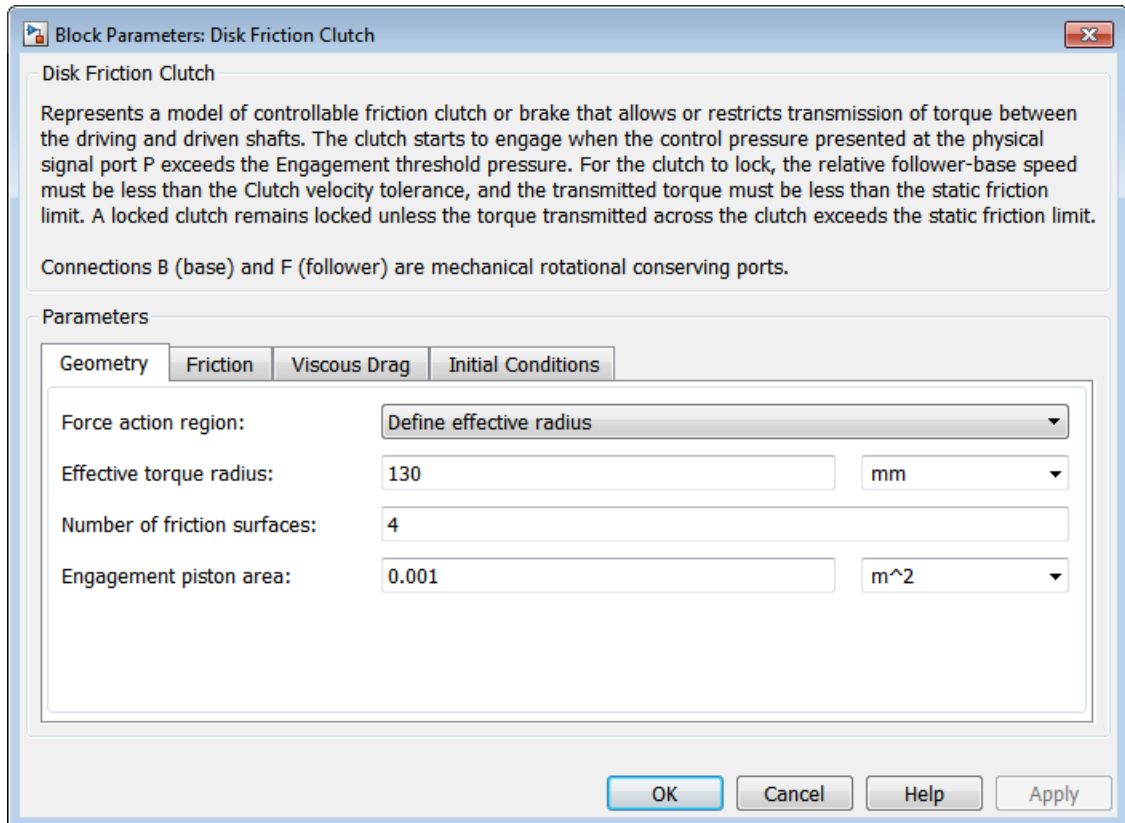
A friction clutch transfers torque between two driveline axes by coupling them with friction. The Disk Friction Clutch block models a friction clutch based on disks in contact, with kinetic friction and static (locking) friction acting on the two axes. For model details, see “Disk Friction Clutch Model” on page 1-60.

## Ports

B and F are rotational conserving ports representing, respectively, the clutch input (base) and output (follower) driveshaft axes. The clutch motion is measured as the *slip*  $\omega = \omega_F - \omega_B$ , the angular velocity of follower relative to base.

The clutch requires a physical signal input P that represents the hydraulic pressure (in pascals) modulating the applied kinetic friction. This signal should be positive or zero. A signal P less than zero is interpreted as zero.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with four tabs.

# Disk Friction Clutch

## Dimensions

Dimensions	Friction	Viscous Drag	Initial Conditions
Force action region:	Define effective radius		
Effective torque radius:	130	mm	
Number of friction surfaces:	4		
Engagement piston area:	0.001	m <sup>2</sup>	

### Force action region

Select how to model the clutch friction geometry. The default is Define effective radius.

- Define effective radius — Model friction geometry in terms of disk radius.

### Effective Radius

Force action region:	Define effective radius	
Effective torque radius:	130	mm

### Effective torque radius

The effective moment arm radius  $r_{\text{eff}}$  that determines the kinetic friction torque inside the clutch. The default is 130.

From the drop-down list, choose units. The default is millimeters (mm).

- **Define annular region** — Model friction geometry in terms of annulus dimensions. If you select this option, the panel changes from its default.

## Annular Region

Force action region:	Define annular region ▼	
Friction disk outside diameter:	150	mm ▼
Friction disk inside diameter:	100	mm ▼

### Friction disk outside diameter

The diameter  $2r_o$  across the outer edge of the friction disk annulus. The default is 150.

From the drop-down list, choose units. The default is millimeters (mm).

### Friction disk inside diameter

The diameter  $2r_i$  across the inner edge of the friction disk annulus. The default is 100.

From the drop-down list, choose units. The default is millimeters (mm).

### Number of friction surfaces

The number  $N$  of friction-generating contact surfaces inside the clutch. The default is 4.

### Engagement piston area

The effective area  $A$  of the clutch piston when the piston is applying pressure across the clutch. The default is 0.001.

From the drop-down list, choose units. The default is meters-squared ( $m^2$ ).

# Disk Friction Clutch

## Friction

Dimensions	Friction	Viscous Drag	Initial Conditions
Friction model:	Fixed kinetic friction coefficient		
Kinetic friction coefficient:	0.3		
Static friction coefficient:	0.35		
De-rating factor:	1		
Clutch velocity tolerance:	0.001	rad/s	
Engagement threshold pressure:	100	Pa	

### Friction model

Select how to model the dimensionless Coulomb kinetic friction coefficient  $k_K$  across the clutch when the clutch is slipping. The default is Fixed kinetic friction coefficient.

- Fixed kinetic friction coefficient — Model Coulomb kinetic friction in terms of a constant kinetic friction coefficient.

### Fixed Kinetic Friction Coefficient

Friction model:	Fixed kinetic friction coefficient
Kinetic friction coefficient:	0.3

### Kinetic friction coefficient

The dimensionless kinetic friction coefficient  $k_K$  applied to the normal force across clutch. The default is 0.3.

- Table lookup kinetic friction coefficient — Model Coulomb kinetic friction in terms of a kinetic friction coefficient

lookup function defined at discrete relative velocity values. If you select this option, the panel changes from its default.

## Table Lookup Kinetic Friction Coefficient

Friction model:	Table lookup kinetic friction coefficient	
Kinetic friction coefficient relative velocity vector:	0.1 0.2 0.3 0.4 0.5 1 2 5 10 100 1000 ]	rad/s
Kinetic friction coefficient vector:	[ 0.29 0.28 0.26 0.22 0.21 0.2 0.2 0.19 0.19 0.19 0.19 ]	
Friction coefficient interpolation method:	Linear	
Friction coefficient extrapolation method:	From last 2 points	

### Kinetic friction coefficient relative velocity vector

Vector of clutch slip values  $\omega$  at which to define kinetic friction coefficient.

From the drop-down list, choose units. The default is radians/second (rad/s).

### Kinetic friction coefficient vector

Vector of kinetic friction coefficient values  $k_K$ .

### Friction coefficient interpolation method

Method to interpolate kinetic friction coefficient function between discrete relative velocity values within the range of definition. The default is Linear.

### Friction coefficient extrapolation method

Method to extrapolate kinetic friction coefficient function outside the range of definition. The default is From last 2 points.

# Disk Friction Clutch

---

## **Static friction coefficient**

Dimensionless Coulomb static friction coefficient  $k_s$  applied to the normal force across the clutch when the clutch is locked. Must be larger than  $k_K$ . The default is 0.35.

## **De-rating factor**

Dimensionless de-rating factor  $D$  that accounts for clutch disk wear by proportionately reducing clutch friction. The default is 1.

## **Clutch velocity tolerance**

The minimum relative angular speed  $\omega_{Tol}$  above which the clutch cannot lock. Below this speed, the clutch can lock. The default is 1e-3.

From the drop-down list, choose units. The default is radians/second (rad/s).

## **Engagement threshold pressure**

The minimum pressure  $P_{th}$  needed to engage the clutch. This lower bound applies to the physical signal input pressure  $P$ . If  $P$  falls below this value, the clutch applies no pressure. The default is 100.

From the drop-down list, choose units. The default is pascals (Pa).



## Viscous Drag

Dimensions	Friction	Viscous Drag	Initial Conditions
Viscous drag torque coefficient:			
		<input type="text" value="0"/>	<input type="text" value="N*m/(rad/s)"/>

### Viscous drag torque coefficient

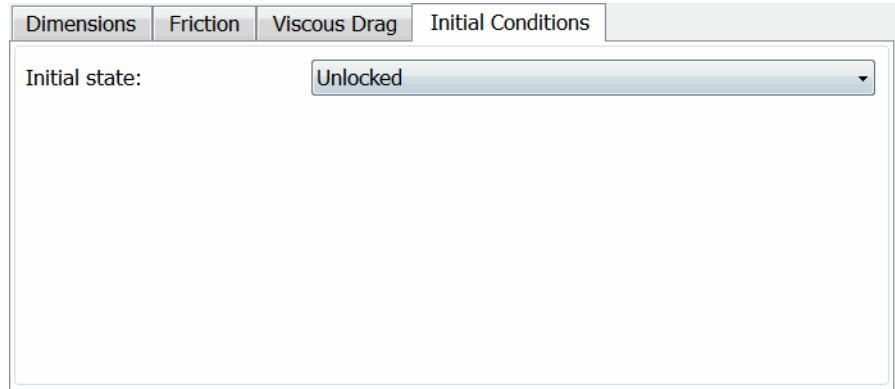
Viscous friction coefficient  $\mu$  applied to the relative slip  $\omega$  between the base and follower axes. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/ (rad/s)).

# Disk Friction Clutch

---

## Initial Conditions



The screenshot shows a software interface with four tabs: 'Dimensions', 'Friction', 'Viscous Drag', and 'Initial Conditions'. The 'Initial Conditions' tab is active. Below the tabs, there is a label 'Initial state:' followed by a dropdown menu. The dropdown menu is open, showing the option 'Unlocked' selected.

### Initial state

From the drop-down list, choose whether the clutch starts simulation in a locked or unlocked state. The default is Unlocked.

## Disk Friction Clutch Model

The Disk Friction Clutch is based on the Fundamental Friction Clutch. For the complete friction clutch model, consult the Fundamental Friction Clutch block reference page. This section discusses the simplified model implemented in the Disk Friction Clutch.

When you apply a pressure signal above threshold ( $P \geq P_{th}$ ), the Disk Friction Clutch block can apply two kinds of friction to the driveline motion, kinetic and static. The clutch applies kinetic friction torque only when one driveline axis is spinning relative to the other driveline axis. The clutch applies static friction torque when the two driveline axes lock and spin together. The block iterates through multistep testing to determine when to lock and unlock the clutch.

### Clutch Variable, State, and Mode Summary

This table summarizes the clutch variables.

## Clutch Variables

Symbol	Definition	Significance
$\omega$	Relative angular velocity (slip)	$\omega_F - \omega_B$
$\omega_{Tol}$	Slip tolerance for clutch locking	<i>See the following model</i>
$P, P_{th}$	Clutch pressure and threshold	Input pressure applied to clutch discs; threshold clutch pressure: $P_{th}, P > 0$ .
$P_{fric}$	Clutch friction capacity	$\max[(P - P_{th}), 0]$
$D$	Clutch de-rating factor	<i>See the following model</i>
$N$	Number of friction surfaces	<i>See the following model</i>
$A$	Engagement surface area	<i>See the following model</i>
$r_{eff}$	Effective torque radius	Effective moment arm of clutch friction force
$k_K$	Kinetic friction coefficient	Dimensionless coefficient of kinetic friction of clutch discs. Function of $\omega$ .
$k_S$	Static friction coefficient	Dimensionless coefficient of static friction of clutch discs.
$\mu$	Viscous drag coefficient	<i>See the following model</i>
$\tau_K$	Kinetic friction torque	<i>See the following model</i>
$\tau_S$	Static friction torque limit	(static friction peak factor) (kinetic friction torque for $\omega \rightarrow 0$ ) ( <i>See the following model</i> )

# Disk Friction Clutch

---

## Relation to Fundamental Friction Clutch

Instead of requiring the kinetic and static friction limit torques as input signals, the Disk Friction Clutch calculates the kinetic and static friction from the clutch parameters and the input pressure signal  $P$ .

### Kinetic Friction

The kinetic friction torque is the positive sum of viscous drag and surface contact friction torques:

$$\tau_K = \mu\omega + \tau_{\text{contact}} .$$

(The kinetic friction torque opposes the relative slip and is applied with an overall minus sign.) The contact friction is a product of six factors:

$$\tau_{\text{contact}} = k_K \cdot D \cdot N \cdot r_{\text{eff}} \cdot P_{\text{fric}} \cdot A \geq 0 .$$

You specify the *kinetic friction coefficient*  $k_K$  as either a constant or a tabulated discrete function of relative angular velocity  $\omega$ . The tabulated function is assumed to be symmetric for positive and negative values of the relative angular velocity, so that you need to specify  $k_K$  for positive values of  $\omega$  only.

The clutch applies a normal force from its piston as the product of the clutch friction capacity  $P_{\text{fric}}$  and engagement surface area  $A$ , on each of  $N$  friction surfaces. The pressure signal  $P$  should be nonnegative. If  $P$  is less than  $P_{\text{th}}$ , the clutch applies no friction at all.

The *effective torque radius*  $r_{\text{eff}}$  is the effective radius, measured from the driveline axis, at which the kinetic friction forces are applied at the frictional surfaces. It is related to the geometry of the friction surface by:

$$r_{\text{eff}} = \frac{2 r_o^3 - r_i^3}{3 r_o^2 - r_i^2}$$

$r_o$  and  $r_i$  are the outer and inner radii, respectively, of the friction surface, modeled as an annular disk.

The *clutch de-rating factor*  $D$  accounts for clutch wear. For a new clutch,  $D$  is one. For a clutch approaching a “uniform wear” state:

$$D \rightarrow \frac{3}{4} \frac{(r_o + r_i)^2}{r_o^2 + r_o r_i + r_i^2}$$

## Static Friction

The static friction limit is related to the kinetic friction, setting  $\omega$  to zero and replacing the kinetic with the static friction coefficient:

$$\tau_S = k_S \cdot D \cdot N \cdot r_{\text{eff}} \cdot P_{\text{fric}} \cdot A \geq 0 .$$

$k_S > k_K$ , so that the torque  $\tau$  needed across the clutch to unlock it by overcoming static friction is larger than the kinetic friction at the instant of unlocking, when  $\omega = 0$ .

The *static friction torque range* or limits are then defined symmetrically as:

$$\tau_S \equiv \tau_S^+ = -\tau_S^- .$$

## Wait State: Locking and Unlocking

The Wait state of the Disk Friction Clutch is identical to the Wait state of the Fundamental Friction Clutch, with the replacement of the positive kinetic friction condition ( $\tau_K > 0$ ) by the positive clutch friction capacity condition ( $P \geq P_{\text{th}}$ ).

## Power Dissipated by the Clutch

The power dissipated by the clutch is  $|\omega \tau_K|$ . The clutch dissipates power only if it is both slipping ( $\omega \neq 0$ ) and applying kinetic friction ( $\tau_K > 0$ ).

## Examples

These SimDriveline example models contain working examples of disk friction clutches that change gear couplings:

- `sdl_simple_clutch`
- `sdl_clutch_engage`

# Disk Friction Clutch

---

- `sdl_simple_transmission`
- `sdl_dual_clutch`
- `sdl_hydraulic_clutch`
- `sdl_crcr`
- `sdl_vehicle`

## See Also

Differential, Fundamental Friction Clutch, Torque Converter, Variable Ratio Transmission

“Gear Coupling Control with Clutches”

## For More Information on Fundamental Friction Clutch

The Disk Friction Clutch is based on the Fundamental Friction Clutch block. See the Fundamental Friction Clutch block reference page for more information about:

- “Clutch Friction” on page 1-130
- “Clutch Directionality” on page 1-129
- “Clutch Velocity Tolerance” on page 1-129
- “Clutch Initial State — Overriding Initial Default State by Manual Locking” on page 1-130
- “Clutch State, Transition, and Variable Summary” on page 1-131
- “Clutch States and Transitions” on page 1-134
- “Unlocked State: Kinetic Friction” on page 1-135
- “Locked State: Static Friction” on page 1-136
- “How the Friction Clutch Locks and Unlocks” on page 1-136

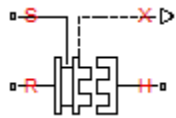
## Purpose

Dog clutch with ring, hub, and rectangular teeth, and optional dynamic backlash

## Library

Clutches

## Description



The Dog Clutch models a dog clutch of two halves; one of which (the ring) is fixed to the spinning driveshaft, while the other (the hub) is mounted freely on the same shaft. You can switch the Dog Clutch dynamics between two models:

- One model approximates the locking and unlocking as if the dog clutch were a friction clutch.
- The other model directly models the tooth-tooth dynamic interaction and captures internal compliance, including backlash.

For model details, see “Dog Clutch Model” on page 1-76.

## Ports

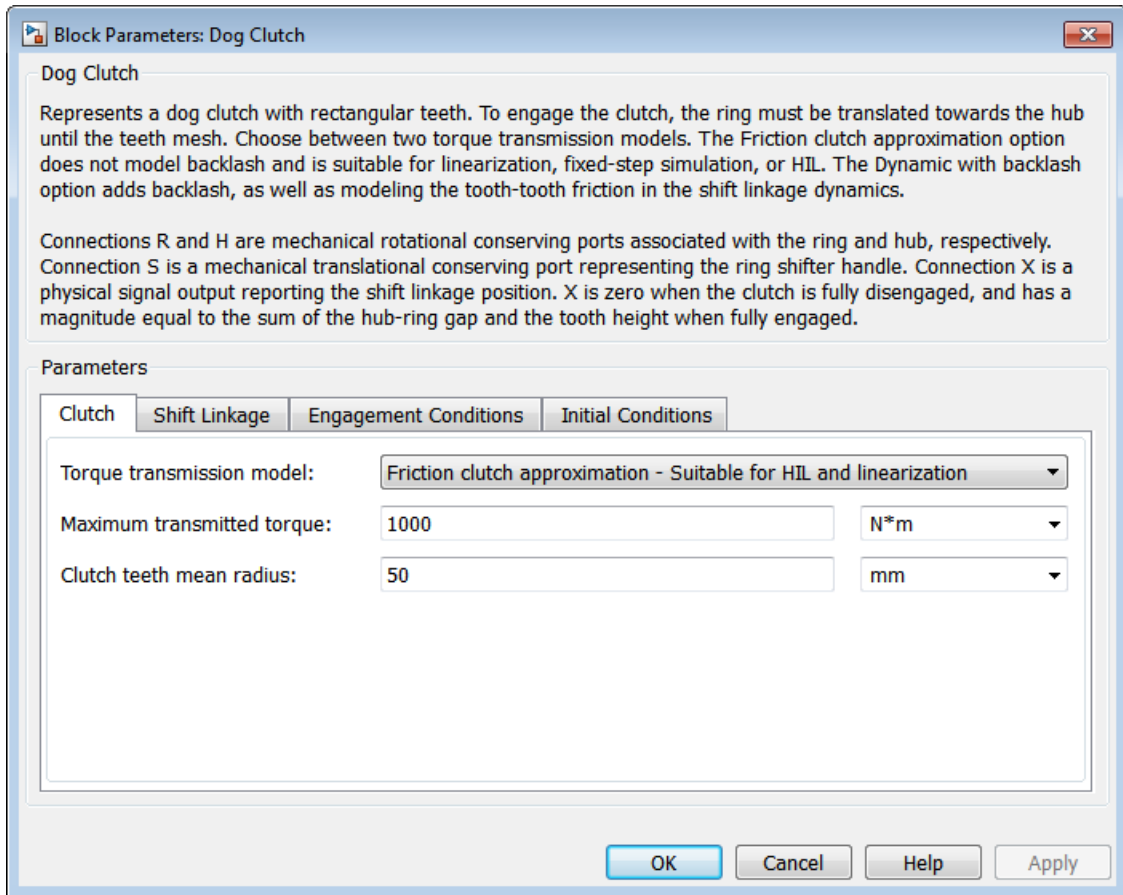
Port	Description
S	Translational conserving port that represents the ring shifter handle
X	Physical signal output port that represents the shift linkage position
R	Rotational conserving port that represents the clutch hub
H	Rotational conserving port that represents the clutch ring

# Dog Clutch

## Dialog Box and Parameters

The dialog box has one active area, **Parameters**, with four tabs.

### Clutch



### Torque transmission model

Choice of model used to determine torque transmission between the ring and hub shafts. The default value is Friction clutch approximation Suitable for HIL and linearization.



- Friction clutch approximation—Model torque transmission as a friction clutch.

## Friction Clutch Approximation

### Maximum transmitted torque

Maximum torque  $\tau_{\max}$  transmitted by the clutch if it is locked (no slip); used to set the static friction limit  $\tau_s$  for the friction clutch approximation. The default is 1000.

From the drop-down list, choose units. The default is newton-meters (N\*m).

### Clutch teeth mean radius

Mean radius of the clutch teeth. Must be greater than zero. The default is 50.

From the drop-down list, choose units. The default is millimeters (mm).

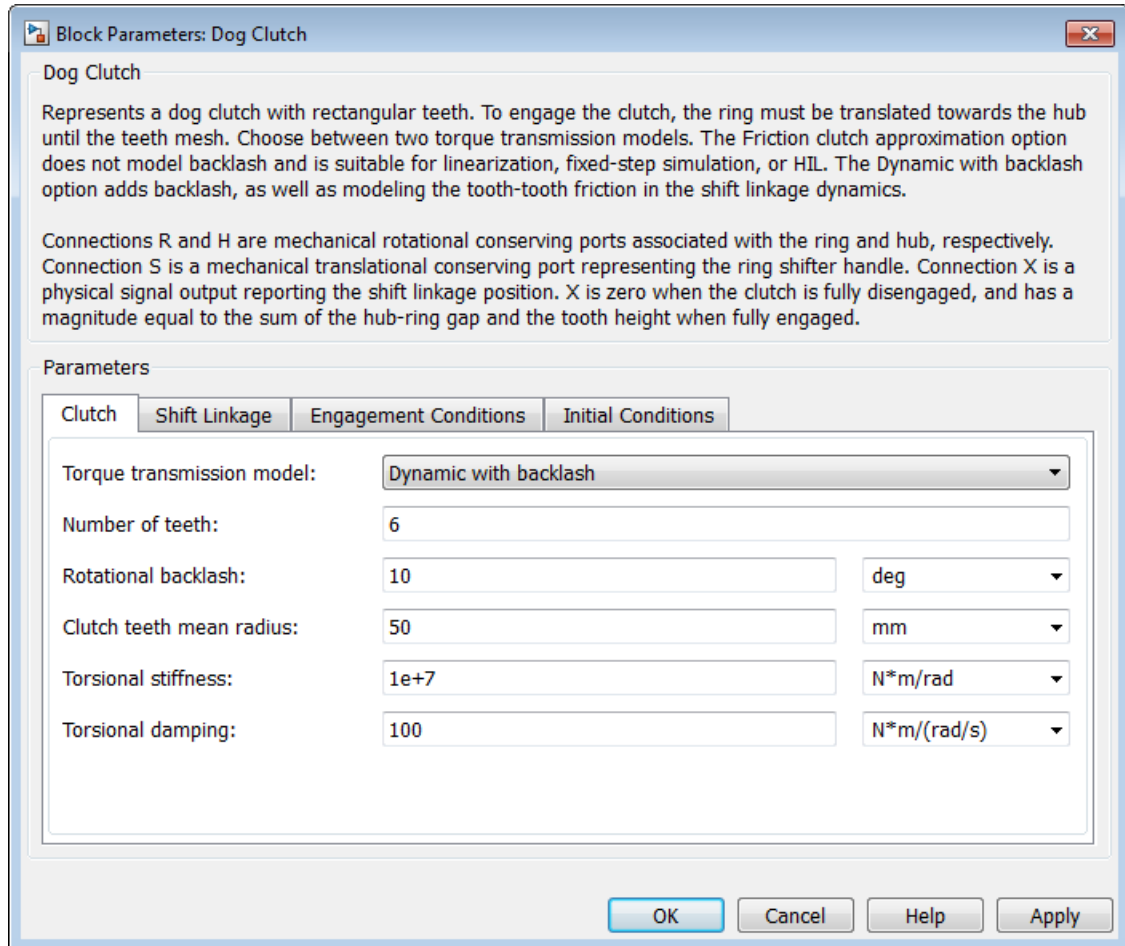
- Dynamic with backlash — Model torque transmission explicitly as meshing gear teeth with compliance and backlash. Selecting this option causes additional parameters to appear in the **Shift Linkage** and **Initial Conditions** tabs.

Exposed Parameter	Dialog Box Tab
Tooth-tooth friction coefficient	Shift Linkage
Initial ring-hub offset angle	Initial Conditions

**Tooth-tooth friction coefficient** to appear in the the **Shift Linkage** tab.

# Dog Clutch

## Dynamic with Backlash



## Number of teeth

Number of teeth  $N$  present in each half of the clutch (ring and hub). The parameter must be greater than or equal to one. The default value is 6.

## Rotational backlash

Angular play  $\delta$  of the ring with respect to the hub. The parameter must be greater than zero. The default value is 10.

From the drop-down list, choose units. The default is degrees (deg).

## Clutch teeth mean radius

Mean radius of the clutch teeth. The parameter must be greater than zero. The default is 50.

From the drop-down list, choose units. The default is millimeters (mm).

## Torsional stiffness

Rotational stiffness  $k$  of tooth-tooth contact. The parameter must be greater than zero. The default is  $1e+7$ .

From the drop-down list, choose units. The default is newton-meters/radian ( $N*m/rad$ ).

## Torsional damping

Rotational damping  $\mu_R$  of tooth-tooth contact. The parameter must be greater than zero. The default is 100.

From the drop-down list, choose units. The default is newton-meters/(radian/second) ( $N*m/(rad/s)$ ).

# Dog Clutch

## Shift Linkage

**Block Parameters: Dog Clutch**

**Dog Clutch**

Represents a dog clutch with rectangular teeth. To engage the clutch, the ring must be translated towards the hub until the teeth mesh. Choose between two torque transmission models. The Friction clutch approximation option does not model backlash and is suitable for linearization, fixed-step simulation, or HIL. The Dynamic with backlash option adds backlash, as well as modeling the tooth-tooth friction in the shift linkage dynamics.

Connections R and H are mechanical rotational conserving ports associated with the ring and hub, respectively. Connection S is a mechanical translational conserving port representing the ring shifter handle. Connection X is a physical signal output reporting the shift linkage position. X is zero when the clutch is fully disengaged, and has a magnitude equal to the sum of the hub-ring gap and the tooth height when fully engaged.

**Parameters**

Clutch	Shift Linkage	Engagement Conditions	Initial Conditions
Tooth height:	10	mm	
Hard stop at back of shift linkage:	Hard stop when fully disengaged		
Ring-hub clearance when disengaged:	3	mm	
Ring stop stiffness:	1e+6	N/m	
Ring stop damping:	1000	N/(m/s)	
Shift linkage viscous friction coefficient:	100	N/(m/s)	
Tooth-tooth friction coefficient:	0.05		

OK Cancel Help Apply

### Tooth height

Length  $h$  of the ring and hub teeth along the driveshaft axis. The parameter must be greater than zero. The default is 10.

From the drop-down list, choose units. The default is millimeters (mm).

### **Hard stop at back of shift linkage**

Hard stop that prevents the shift linkage from traveling beyond the fully disengaged position. Choices include:

- No hard stop when fully disengaged
- Hard stop when fully disengaged

The default value is Hard stop when fully disengaged.

### **Ring-hub clearance when disengaged**

Gap  $z_{\text{gap}}$  between ring and hub teeth when clutch is fully disengaged. The parameter must be greater than zero. The default is 3.

From the drop-down list, choose units. The default is millimeters (mm).

### **Ring stop stiffness**

Translational stiffness  $k_{\text{stop}}$  experienced by the ring if it moves beyond its end stops. The parameter must be greater than zero. The default is 1e+6.

From the drop-down list, choose units. The default is newtons/meter (N/m).

### **Ring stop damping**

Translational damping  $\mu_{\text{stop}}$  experienced by the ring if it moves beyond its end stops. The parameter must be greater than or equal zero. The default is 1000.

From the drop-down list, choose units. The default is newtons/(meter/second) (N/(m/s)).

### **Shift linkage viscous friction coefficient**

Translational damping  $\mu_{\text{T}}$  experienced by the shift linkage. The parameter must be great than zero. The default is 100.

# Dog Clutch

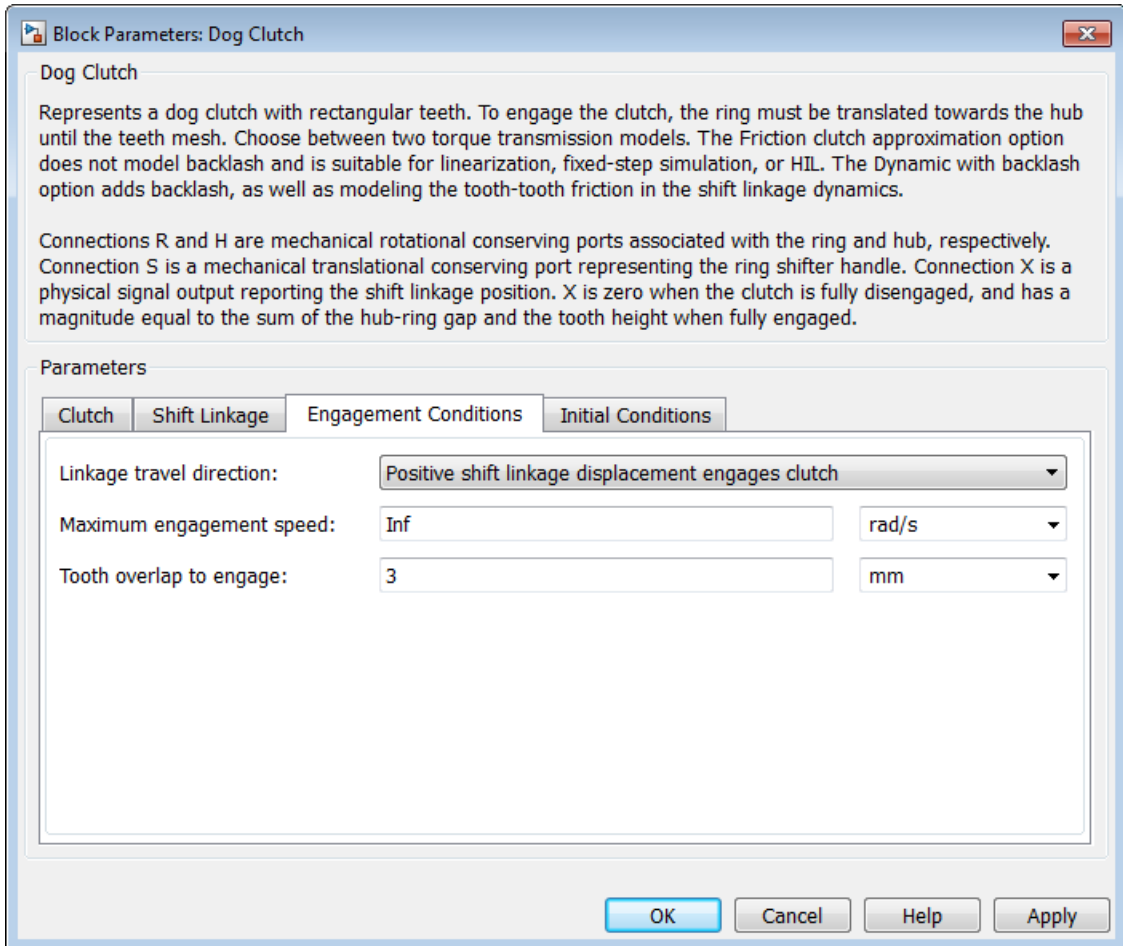
---

From the drop-down list, choose units. The default is newtons/(meter/second) (N/(m/s)).

## **Tooth-tooth friction coefficient**

Dimensionless coefficient  $k_K$  of kinetic friction between teeth in sliding contact. The parameter is visible only if **Torque transmission model** in the **Clutch** tab has the value **Dynamic with backlash**. The parameter must be greater than zero. The default is 0.05.

## Engagement Conditions



### Linkage travel direction

Direction of travel required to engage the clutch. The default setting is Positive shift linkage displacement engages clutch.

# Dog Clutch

---

## **Maximum engagement velocity**

Maximum slip  $\omega_{\max}$  allowed for clutch to engage. The parameter must be greater than zero. The default is infinity (Inf).

From the drop-down list, choose units. The default is radians/second (rad/s).

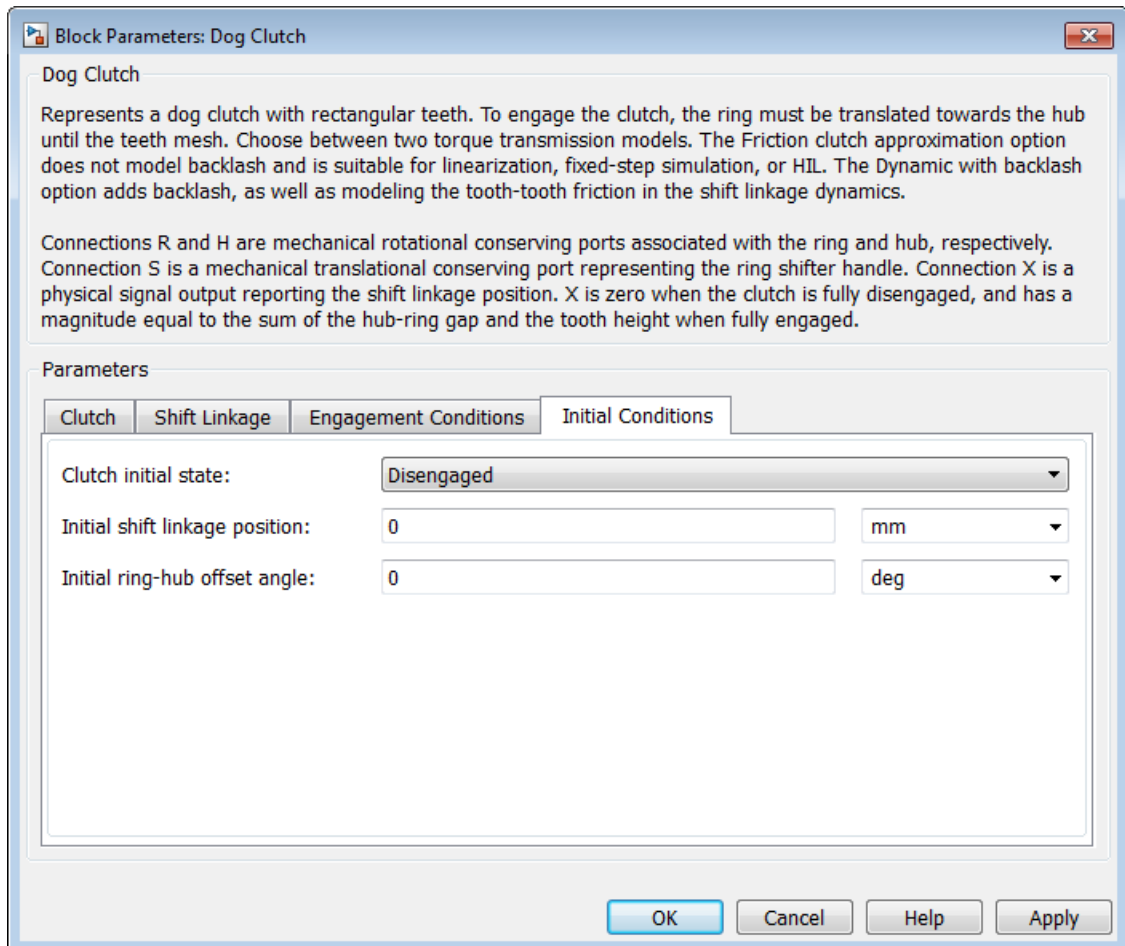
## **Tooth overlap to engage**

Minimum overlap  $h_o$  of ring and hub teeth along the shaft axis needed for the clutch to engage. The parameter must be greater than zero. The default is 3.

From the drop-down list, choose units. The default is millimeters (mm).



## Initial Conditions



### Clutch Initial state

Configuration in which the dog clutch begins simulation. Choices include:

# Dog Clutch

- **Disengaged**—Clutch transmits zero torque between the ring and hub shafts.
- **Engaged**—Clutch transmits torque between the ring and hub shafts.

The default value is **Disengaged**.

## Initial shift linkage position

Relative position between the shift linkage and the hub where the dog clutch begins simulation. The default value is 0 mm.

## Initial ring-hub offset angle

Relative angle  $\varphi_0$  between ring and hub assemblies where the dog clutch begins simulation. The default value is 0 deg. The parameter is visible only when **Torque transmission model** in the **Clutch** tab has the value **Dynamic with Backlash**.

The following restrictions apply to the value of the parameter.

Clutch initial state	Initial ring-hub offset angle
Disengaged	$-180^\circ/N \leq \varphi_0 \leq +180^\circ/N$ where $N$ is the number of teeth present in the clutch assembly
Engaged	$-\delta/2 \leq \varphi_0 \leq \delta/2$ where $\delta$ is the backlash angle of the dog clutch assembly

## Dog Clutch Model

The Dog Clutch is modeled depending on which torque transmission mechanism you choose:

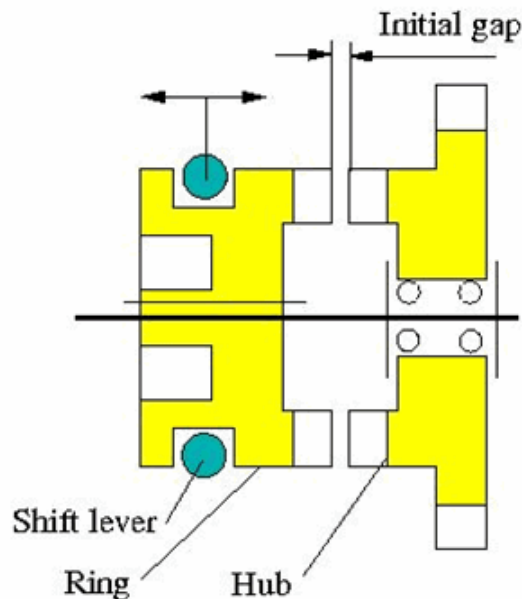
- If you use the friction clutch approximation, the Dog Clutch is based on the Fundamental Friction Clutch. For the complete friction clutch model, consult the Fundamental Friction Clutch block reference page.

- If you use the dynamic model, the Dog Clutch is modeled as a coupling that combines gear features (meshing teeth), torsional compliance (spring and damping torques), and backlash.

Positive force applied to the shift linkage at port S tends to move the ring toward the hub and engage the clutch. Negative applied force tends to move the ring away from the hub and disengage the clutch. When the clutch is fully disengaged, the shift linkage position  $z = 0$ . When the clutch is fully engaged,  $z$  is the sum of the hub-ring gap (clearance)  $z_{\text{gap}}$  and the tooth height  $h$ :  $z = z_{\text{gap}} + h$ .

## Clutch Geometry, Variable, and State Summary

The figure displays the geometry of the dog clutch ring and hub. The table summarizes the dog clutch variables.



**Clutch Geometry**

# Dog Clutch

## Clutch Variables

Symbol	Definition	Significance
$\omega$	Relative angular velocity (slip)	$\omega_R - \omega_H$
$\omega_{\max}$	Maximum slip for clutch engagement	<i>See the following model</i>
$z_{\text{gap}}$	Ring-hub clearance or gap	Initial distance between ring and hub along driveshaft axis (if clutch starts disengaged)
$h$	Tooth height	Length of ring and hub teeth, parallel to driveshaft axis
$h_o$	Minimum engagement tooth-tooth overlap	Smallest length of overlap between ring and hub teeth needed for clutch to engage
$\varphi$	Relative angle between ring and hub	Relative angular position of ring and hub
$\delta$	Backlash	Dynamical range of $\varphi$ without contact torque
$k_{RH}$	Torsional stiffness of ring-hub coupling	<i>See the following model</i>
$\mu_R$	Torsional viscous damping of ring-hub coupling	<i>See the following model</i>
$k_K$	Kinetic friction coefficient of tooth-tooth meshing	<i>See the following model</i>
$\mu_T$	Shift linkage translational viscous drag coefficient	<i>See the following model</i>
$\tau_C$	Kinetic contact torque	<i>See the following model</i>
$\tau_{\max}$	Maximum transmitted torque	<i>See the following model</i>

## Engagement and Locking Conditions

If not initially engaged, the dog clutch engages (transmits torque) only if:

- The ring slides far enough toward the hub that it closes the initial gap  $z_{\text{gap}}$ ; and translates further so that the ring and hub teeth overlap each other by more than minimum tooth engagement overlap  $h_o$ :  $z > z_{\text{gap}} + h_o$ , allowing the teeth to mesh.
- The slip  $\omega$  is less than the maximum engagement velocity:  $|\omega| < \omega_{\text{max}}$ .

If you use the friction clutch approximation, the dog clutch locks only if the torque across the clutch is smaller than the maximum transmitted torque, in addition the preceding conditions.

If you use the dynamical model, the dog clutch locks only if the relative angular position of the ring and hub also allow the teeth to mesh, in addition the preceding conditions.

---

**Tip** For simulation, you can adjust the minimum tooth-tooth overlap  $h_o$  to reduce engagement instability, the tendency of the clutch to switch rapidly between engaged and disengaged states.

---

## Power Dissipated by the Clutch

The power dissipated by the clutch is  $|\omega \tau_C|$ . The clutch dissipates power only if it is both slipping ( $\omega \neq 0$ ) and applying a torque ( $\tau_C > 0$ ).

## Friction Clutch Approximation

The friction clutch model is a simplification of the true clutch behavior and models the dog clutch as switching between disengaged, engaged (but unlocked), and locked. In its unlocked states (engaged and disengaged), this model of the dog clutch behaves like the alternative dynamical model.

# Dog Clutch

---

From the point where the teeth overlap by the necessary minimum  $h_o$ , the clutch normal force increases linearly with distance ( $z$ ). When the clutch is fully engaged, the normal force is such that the torque required to unlock the clutch is equal to the maximum transmitted torque  $\tau_{\max}$ . Set its value to slightly more than the maximum transmitted torque you expect to see in your model.

---

**Tip** If you plan to use the Dog Clutch for fixed-step, real-time, or hardware-in-the-loop (HIL) simulation, MathWorks recommends that you select the friction clutch approximation.

This approximation also helps to speed up simulation when you use variable-step solvers, if you are not interested in capturing backlash or shift lever friction effects.

---

## Dynamical Model

The dynamical model allows for realistic engagement dynamics: damped, elastic deformation of the meshing teeth; as well as backlash, translational friction, and end stops. There is no static friction or locking in this case.

### Contact Torque

If the clutch is engaged, it applies a contact torque  $\tau_c$  between the ring and hub, the sum of torsional spring-like compliance (with backlash) and rotational viscous drag.

<b>c</b>	<b>Range</b>
$-k_{RH}(\varphi - \delta/2) - \mu_R\omega$	$\varphi > \delta/2$
0	$ \varphi  < \delta/2$
$-k_{RH}(\varphi + \delta/2) - \mu_R\omega$	$\varphi < -\delta/2$

### End Stop Compliance and Translational Losses

The shift linkage translational motion (position  $z$  and velocity  $v = dz/dt$ ) is limited by linearly compliant end stops, with stiffness and damping.

- If the teeth do not align, or if  $|\omega| > \omega_{\max}$ , then the ring end stop is set to prevent the teeth from meshing. The clutch is disengaged.
- Otherwise, the ring end stop is set to the depth of the tooth plus the disengaged hub-ring gap ( $h + z_{\text{gap}}$ ). The clutch can engage.

The shift linkage motion is also opposed by translational friction  $F_z$ , the sum of Coulomb and viscous friction forces.

$$F_z = -k_K F_N \tanh(4v/v_{\text{th}}) - \mu_T v .$$

- The Coulomb term represents the friction of the relative sliding of the ring and hub teeth in mutual contact.

$F_N$  is the normal force between the teeth in contact and depends on transmitted torque.  $v_{\text{th}}$  is the linear velocity threshold.

- The viscous term represents the viscous drag experienced by the sliding shift linkage.

## See Also

Fundamental Friction Clutch, Loaded-Contact Rotational Friction, Torsional Spring-Damper

“Gear Coupling Control with Clutches”

### For More Information on Fundamental Friction Clutch

If modeled as a friction clutch, the Dog Clutch is based on the Fundamental Friction Clutch block. See the Fundamental Friction Clutch block reference page for more information about:

- “Clutch Friction” on page 1-130
- “Clutch Directionality” on page 1-129
- “Clutch Velocity Tolerance” on page 1-129
- “Clutch Initial State — Overriding Initial Default State by Manual Locking” on page 1-130
- “Clutch State, Transition, and Variable Summary” on page 1-131

# Dog Clutch

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- “Clutch States and Transitions” on page 1-134
- “Unlocked State: Kinetic Friction” on page 1-135
- “Locked State: Static Friction” on page 1-136
- “How the Friction Clutch Locks and Unlocks” on page 1-136



# Double-Pinion Planetary Gear

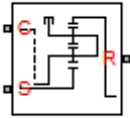
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<b>Purpose</b>	Planetary gear train with two meshed planet gear sets
<b>Library</b>	Gears

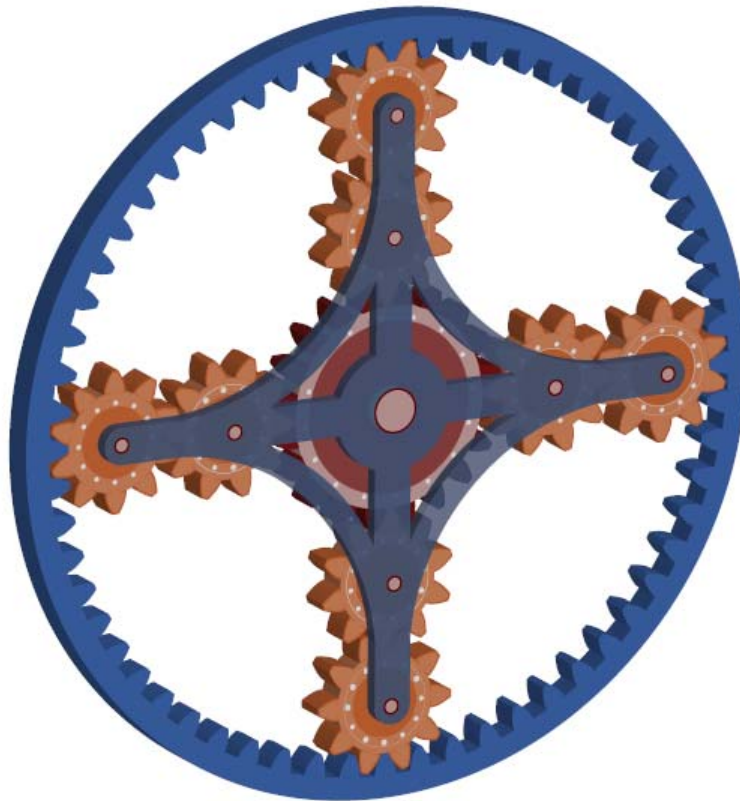
# Double-Pinion Planetary Gear

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## Description



This block represents a planetary gear train with two meshed planet gear sets that sit between the sun gear and the ring gear. A single carrier holds the two planet gear sets at different radii from the sun gear centerline, while allowing the individual gears to rotate with respect to each other. The gear model includes power losses due to friction between meshing gear teeth and viscous damping of the spinning gear shafts.



**Double-Pinion Planetary Gear Set**

# Double-Pinion Planetary Gear

Structurally, the double-pinion planetary gear resembles a Ravigneaux gear without its second, large, sun gear. The inner planet gears mesh with the sun gear and the outer planet gears mesh with the ring gear. Because it contains two planet gear sets, the double-pinion planetary gear reverses the relative rotation directions of the ring and sun gears.

The teeth ratio of a meshed gear pair fixes the relative angular velocities of the two gears in that pair. The dialog box provides two parameters to set the ring-sun and outer planet-inner planet gear teeth ratios. A geometric constraint fixes the remaining teeth ratios—ring-outer planet and inner planet-sun. This geometric constraint requires that the ring gear radius equal the sum of the sun gear radius with the inner and outer planet gear diameters.

$$r_r = r_s + 2 r_{pi} + 2 r_{po}$$

where:

- $r_r$  — Ring gear radius
- $r_s$  — Sun gear radius
- $r_{pi}$  — Inner planet gear radius
- $r_{po}$  — Outer planet gear radius

In terms of the ring-sun and outer planet-inner planet teeth ratios, the ring-outer planet teeth ratio is:

$$\frac{r_r}{r_{po}} = 2 \frac{\frac{r_r}{r_s} \frac{r_{po}}{r_{pi}} + 1}{\frac{r_r}{r_s} - 1 \frac{r_{po}}{r_{pi}}}$$

The inner planet-sun teeth ratio is:

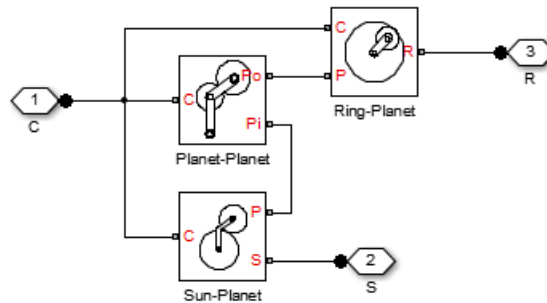
# Double-Pinion Planetary Gear

$$\frac{r_{pi}}{r_s} = \frac{\frac{r_r}{r_s} - 1}{2 \frac{r_{po}}{r_{pi}} + 1}$$

To represent a double-pinion planetary gear, this block uses an underlying Simscape block diagram with three SimDriveline Gear blocks:

- Ring-Planet
- Sun-Planet
- Planet-Planet

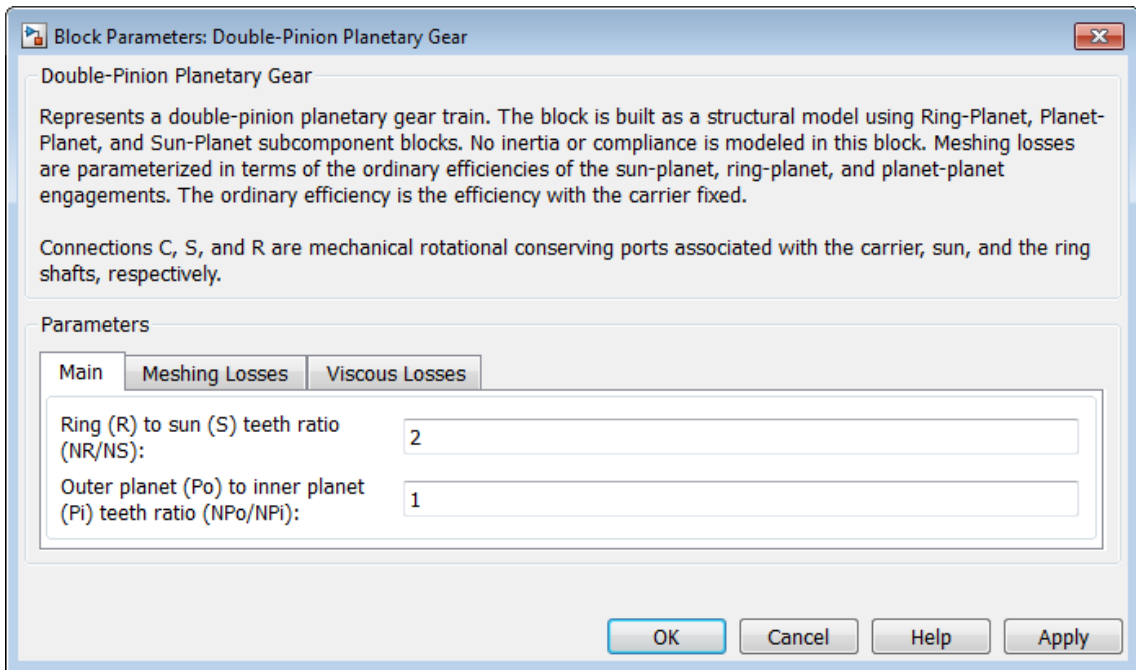
The figure shows the Simscape block diagram for this block.



## Dialog Box and Parameters

The block dialog box contains one active area, **Parameters**, with three tabs: **Main**, **Meshing Losses**, and **Viscous Losses**.

# Double-Pinion Planetary Gear



## Main

Specify the teeth ratios for ring-sun and outer planet-inner planet gears.

### Ring (R) to sun (S) teeth ratio (NR/NS)

Enter the teeth ratio between ring and sun gears. This ratio is the number of teeth in the ring gear divided by the number of teeth in the sun gear. The default value is 2.

### Outer planet (Po) to inner planet (Pi) teeth ratio (NPo/NPi)

Enter the teeth ratio between outer planet and inner planet gears. This ratio is the number of teeth in the outer planet divided by the number of gear teeth in the inner planet. The default value is 1.

## Meshing Losses

Specify the power losses due to friction between meshing gear teeth.

# Double-Pinion Planetary Gear

---

## Friction model

Select how to implement friction losses between meshing gears. The table describes the friction models.

Model	Description
No meshing losses Suitable for HIL simulation	Ignore friction losses. Treat gear meshing as ideal.
Constant efficiency	Include friction losses. Torque transfer between gears depends on a constant efficiency factor.

## Sun-planet, ring-planet, and planet-planet ordinary efficiencies

Enter a vector with the torque transfer efficiencies between sun-planet, ring-planet, and planet-planet gear pairs, in that order. The default vector is [0.98 0.98 0.98].

## Sun-carrier, ring-carrier, and planet-carrier velocity thresholds

Enter a vector with the sun-carrier, ring-carrier, and planet-carrier absolute angular velocities above which the full efficiency loss applies. From the drop-down list, select a physical unit. The default vector is [0.01 0.01 0.01] rad/s.

## Viscous Losses

Specify the power losses due to viscous damping of the gear carriers.

## Sun-carrier, ring-carrier, and planet-carrier viscous friction coefficients

Enter a vector with the viscous friction coefficients that dampen sun-carrier, ring-carrier, and planet-carrier motion, in that order. From the drop-down list, select a physical unit. The default vector is [0 0 0] N\*m/(rad/s).

# Double-Pinion Planetary Gear

---

## Ports

Port	Description
C	Conserving rotational port associated with the planet gear carrier
S	Conserving rotational port associated with the sun gear
R	Conserving rotational port associated with the ring gear

## See Also

[Planetary Gear](#) | [Compound Planetary Gear](#) | [Ravigneaux Gear](#) | [Planet-Planet](#) | [Ring-Planet](#) | [Sun-Planet](#)

# Double-Shoe Brake

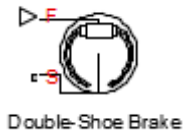
## Purpose

Frictional brake with two pivoted shoes diametrically positioned about rotating drum

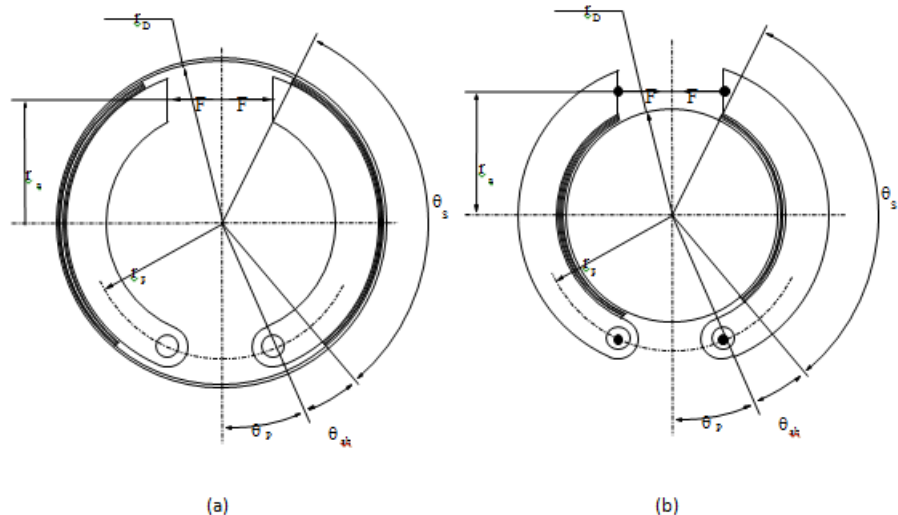
## Library

Brakes & Detents/Rotational

## Description



The block represents a frictional brake with two pivoted rigid shoes that press against a rotating drum to produce a braking action. The rigid shoes sit inside or outside the rotating drum in a diametrically opposed configuration. A positive actuating force causes the rigid shoes to press against the rotating drum. Viscous and contact friction between the drum and the rigid shoe surfaces cause the rotating drum to decelerate. Double-shoe brakes provide high braking torque with small actuator deflections in applications that include motor vehicles and some heavy machinery. The model employs a simple parameterization with readily accessible brake geometry and friction parameters.



In this schematic, a) represents an internal double-shoe brake, and b) represents an external double-shoe brake. In both configurations, a



positive actuation force  $F$  brings the shoe and drum friction surfaces into contact. The result is a friction torque that causes deceleration of the rotating drum. Zero and negative forces do not bring the shoe and drum friction surfaces into contact and produce zero braking torque.

The model uses the long-shoe approximation. Contact angles smaller than  $45^\circ$  produce less accurate results. The following formulas provide the friction torque the leading and trailing shoes develop, respectively:

$$T_{LS} = \frac{\mu p_a r_D^2 (\cos \theta_{sb} - \cos \theta_s)}{\sin \theta_a}$$

$$T_{TS} = \frac{\mu p_b r_D^2 (\cos \theta_{sb} - \cos \theta_s)}{\sin \theta_a}$$

In the formulas, the parameters have the following meaning:

Parameter	Description
$T_{LS}$	Brake torque the leading shoe develops
$T_{TS}$	Brake torque the trailing shoe develops
$\mu$	Effective contact friction coefficient
$p_a$	Maximum linear pressure in the leading shoe-drum contact
$p_b$	Maximum linear pressure in the trailing shoe-drum contact
$r_D$	Drum radius
$\theta_{sb}$	Shoe beginning angle

# Double-Shoe Brake

Parameter	Description
$\theta_s$	Shoe span angle
$\theta_a$	Angle from hinge pin to maximum pressure point  $\theta_a = \begin{cases} \theta_s & \text{if } 0 \leq \theta_s \leq \pi/2 \\ \pi/2 & \text{if } \theta_s > \pi/2 \end{cases}$

The model assumes that only Coulomb friction acts at the shoe-drum surface contact. Zero relative velocity between the drum and the shoes produces zero Coulomb friction. To avoid discontinuity at zero relative velocity, the friction coefficient formula employs the following hyperbolic function:

$$\mu = \mu_{Coulomb} \tanh \frac{4\omega_{shaft}}{\omega_{threshold}}$$

In the formula, the parameters have the following meaning:

Parameter	Description
$\mu$	Effective contact friction coefficient
$\mu_{Coulomb}$	Contact friction coefficient
$\omega_{shaft}$	Shaft velocity
$\omega_{threshold}$	Angular velocity threshold

Balancing the moments that act on each shoe with respect to the pin yields the pressure acting at the shoe-drum surface contact. The following formula provides the balance of moments for the leading shoe.

$$F = \frac{M_N - M_F}{c}$$

In the formula, the parameters have the following meaning:

Parameter	Description
$F$	Actuation Force
$M_N$	Moment acting on the leading shoe due to normal force
$M_F$	Moment acting on the leading shoe due to friction force
$c$	Arm length of the cylinder force with respect to the hinge pin

The following equations give  $M_N$ ,  $M_F$ , and  $c$ , respectively.

$$M_N = \frac{p_a r_p r_D}{\sin \theta_a} \frac{1}{2} [\theta_s - \theta_{sb}] - \frac{1}{4} [\sin 2\theta_s - \sin 2\theta_{sb}]$$

$$M_F = \frac{\mu p_a r_D}{\sin \theta_a} r_D [\cos \theta_{sb} - \cos \theta_s] + \frac{r_p}{4} [\cos 2\theta_s - \cos 2\theta_{sb}]$$

$$c = r_a + r_p \cos \theta_p$$

The parameters have the following meaning:

Parameter	Description
$p_a$	Maximum linear pressure at the shoe-drum contact surface
$r_p$	Pin location radius
$\theta_p$	Hinge pin location angle
$r_a$	Actuator location radius

# Double-Shoe Brake

---

The model does not simulate self-locking brakes. If brake geometry and friction parameters cause a self-locking condition, the model produces a simulation error. A brake self-locks if the friction moment exceeds the moment due to normal forces:

$$M_F > M_N$$

The following formula provides the balance of moments for the trailing shoe.

$$F = \frac{M_N + M_F}{c}$$

Formulas and parameters for  $M_N$ ,  $M_F$ , and  $c$  share the definition in the leading shoe section.

The net braking torque has the formula:

$$T = T_{LS} + T_{TS} + \mu_{visc}$$

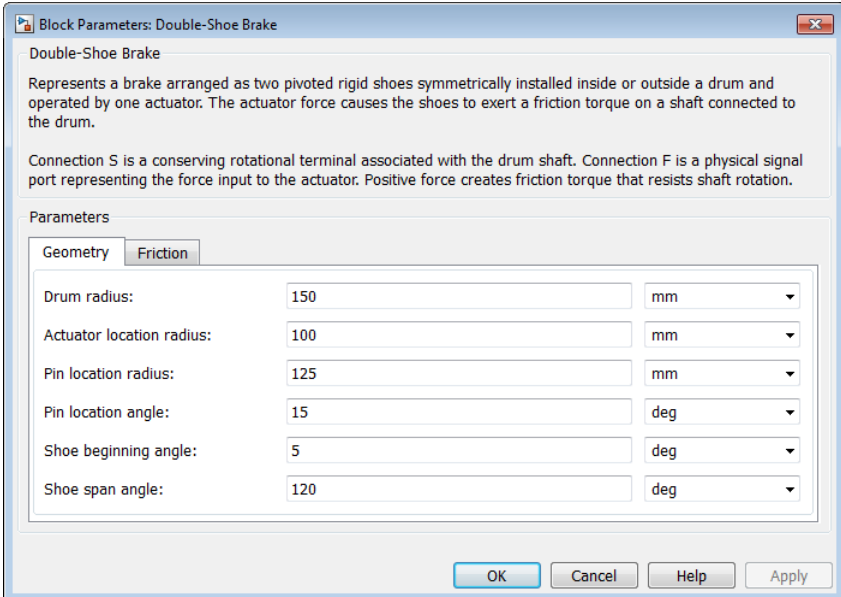
In the formula, parameter  $\mu_{visc}$  is the viscous friction coefficient.

## Assumptions and Limitations

- The brake uses the long-shoe approximation
- The brake geometry does not self-lock
- The model does not account for actuator flow consumption

## Dialog Box and Parameters

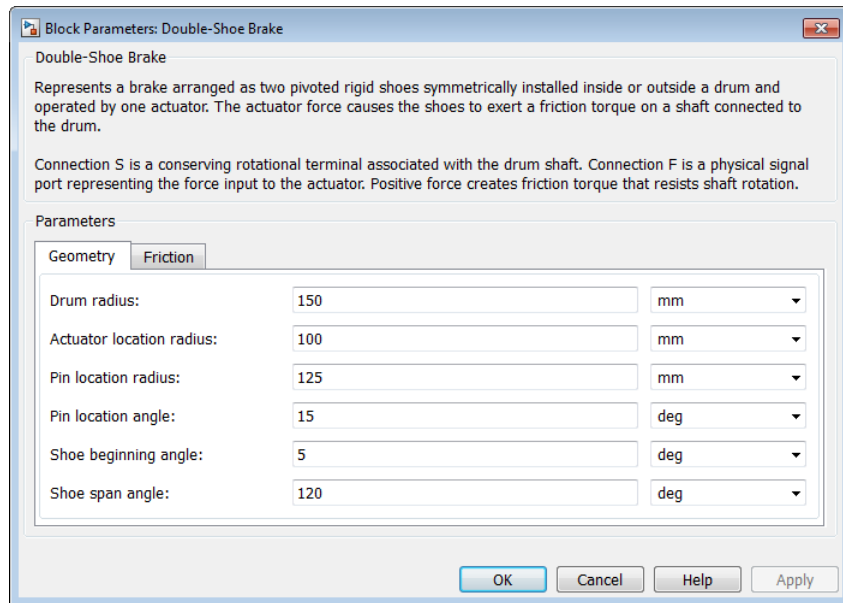
The dialog box includes two tabs, **Geometry** and **Friction**.



### Geometry

Brake geometry parameters.

# Double-Shoe Brake



## Drum radius

Radius of the drum contact surface. The parameter must be greater than zero. The default value is 150 mm.

## Actuator location radius

Distance between the drum center and the force line of action. The parameter must be greater than zero. The default value is 100 mm.

## Pin location radius

Distance between the hinge pin and drum centers. The parameter must be greater than zero. The default value is 125 mm.

## Pin location angle

Angular coordinate of the hinge pin location from the brake symmetry axis. The parameter must be greater than or equal to zero. The default value is 15 deg.

## Shoe beginning angle

Angle between the hinge pin and the beginning of the friction material linen of the shoe. The value of the parameter must be in the range  $0 \leq \theta_{sb} \leq (\pi - \text{pin location angle})$ . The default value is 5 deg.

## Shoe span angle

Angle between the beginning and the end of the friction material linen on the shoe. The value of the parameter must be in the range  $0 < \theta_{sb} \leq (\pi - \text{pin location angle} - \text{shoe beginning angle})$ . The default value is 130 deg.

## Friction

Brake friction parameters.

Block Parameters: Double-Shoe Brake

Double-Shoe Brake

Represents a brake arranged as two pivoted rigid shoes symmetrically installed inside or outside a drum and operated by one actuator. The actuator force causes the shoes to exert a friction torque on a shaft connected to the drum.

Connection S is a conserving rotational terminal associated with the drum shaft. Connection F is a physical signal port representing the force input to the actuator. Positive force creates friction torque that resists shaft rotation.

Parameters

Geometry Friction

Viscous friction coefficient: 0.01 N\*m/(rad/s)

Contact friction coefficient: 0.3

Angular velocity threshold: 0.01 rad/s

OK Cancel Help Apply

# Double-Shoe Brake

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## Viscous friction coefficient

Value of the viscous friction coefficient at the contact surface. The parameter must be greater than or equal to zero. The default value is  $0.01 \text{ n}\cdot\text{m}/(\text{rad}/\text{s})$ .

## Contact friction coefficient

Friction coefficient at the shoe-drum contact surface. The parameter must be greater than zero. The default value is **0.3**.

## Angular velocity threshold

Angular velocity at which the contact friction coefficient practically reaches its steady-state value. The parameter must be greater than zero. the default value is  $0.01 \text{ rad}/\text{s}$ .

## Ports

Port	Description
F	Physical signal port that represents the brake actuating force
S	Conserving rotational port that represents the rotating drum shaft

## See Also

Band Brake, Loaded-Contact Rotational Friction, Rotational Detent



# Double-Sided Synchronizer

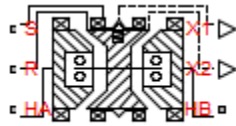
## Purpose

Set of back to back dog and cone clutch pairs assembled symmetrically about a translational detent to provide smooth gear engagement

## Library

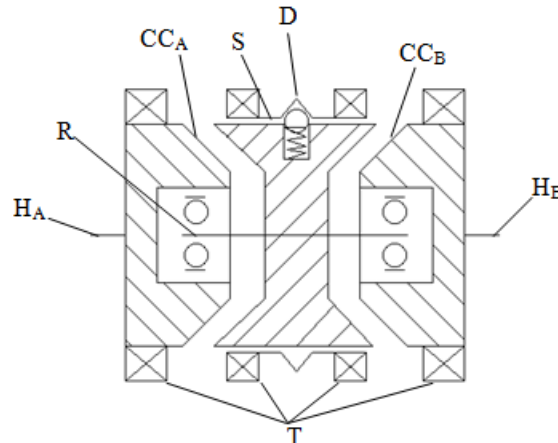
Clutches

## Description



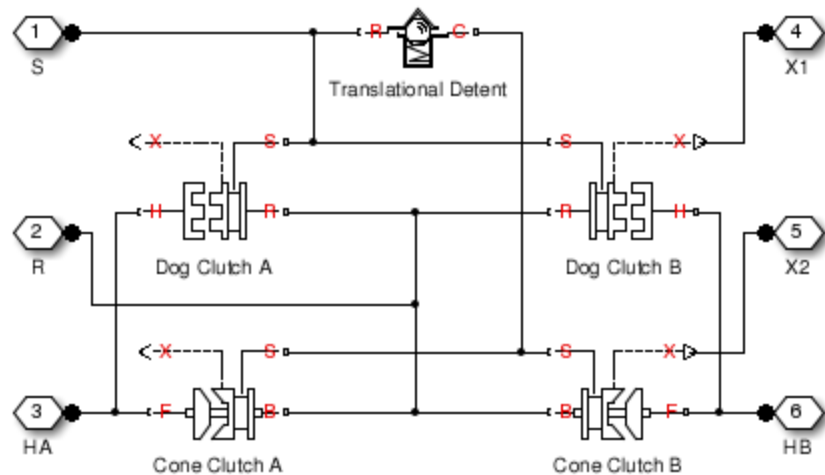
Double-Sided Synchronizer

The block represents a double-sided synchronizer that contains two back-to-back dog clutches, two back-to-back cone clutches, and one translational detent. Shift linkage translation along the negative direction causes the clutches to engage the ring with hub A. Shift linkage translation along the positive direction causes the clutches to engage the ring with hub B. When the magnitude of the shift linkage translation is smaller than the cone clutch ring-hub gap, the synchronizer is in neutral mode and does not transmit torque.



# Double-Sided Synchronizer

The schematic illustrates a double-sided synchronizer in the disengaged state. In this state, the ring (R) and hub ( $H_A$  and  $H_B$ ) shafts can spin independently at different speeds. As the shift linkage (S) translates in the negative direction, the faces of cone clutch A ( $CC_A$ ) come into contact. The friction in the cone clutch decreases the difference in rotational speed between the shafts. When the force on the shift linkage exceeds the peak force of detent (D), the dog clutch teeth (T) can engage. The detent peak force should be such that the cone clutch has enough time and normal force to bring the shafts to sufficiently similar speeds to allow engagement of the dog clutch. Similarly, translating the shift linkage along the positive direction allows the faces of cone clutch B ( $CC_B$ ) to come into contact, and can allow the shaft of the ring to engage with the shaft of the hub B ( $H_B$ ).



The model implements two Dog Clutch blocks, one Translational Detent, and two modified Dog Clutch blocks. Refer to each block reference page for more information on the corresponding block function.

Connections R,  $H_A$ , and  $H_B$  are mechanical rotational conserving ports that represent the ring (R), hub A ( $H_A$ ), and hub B ( $H_B$ ), respectively.

# Double-Sided Synchronizer

Connection S is a mechanical translational conserving port that represents the ring shifter handle.

Connections X1 and X2 are physical signal ports that output the shift linkage positions of the dog clutches and cone clutches, respectively. The following tables provide the values of X1 and X2 in common clutch engagement cases.

<b>Dog Clutch State</b>	<b>X1</b>
Disengaged	0
Fully engaged with hub A	Negative sum of ring-hub gap and tooth height
Fully engaged with hub B	Positive sum of ring-hub gap and tooth height

<b>Cone Clutch State</b>	<b>X2</b>
Disengaged	0
Fully engaged with hub A	Negative value of ring-hub gap
Fully engaged with hub B	Positive value of ring-hub gap

The values of X1 and X2 are zero when the synchronizer is fully disengaged. When the dog clutch is fully engaged with hub A, X1 is equal to the negative sum of its ring-hub gap and tooth height. When the dog clutch is fully engaged with hub B, X1 is equal to the positive sum of its ring-hub gap and tooth height. When the cone clutch is fully engaged with hub A, X2 is equal to the negative of its ring-hub gap. When the cone clutch is fully engaged with hub B, X2 is equal to its ring-hub gap.

## Assumptions and Limitations

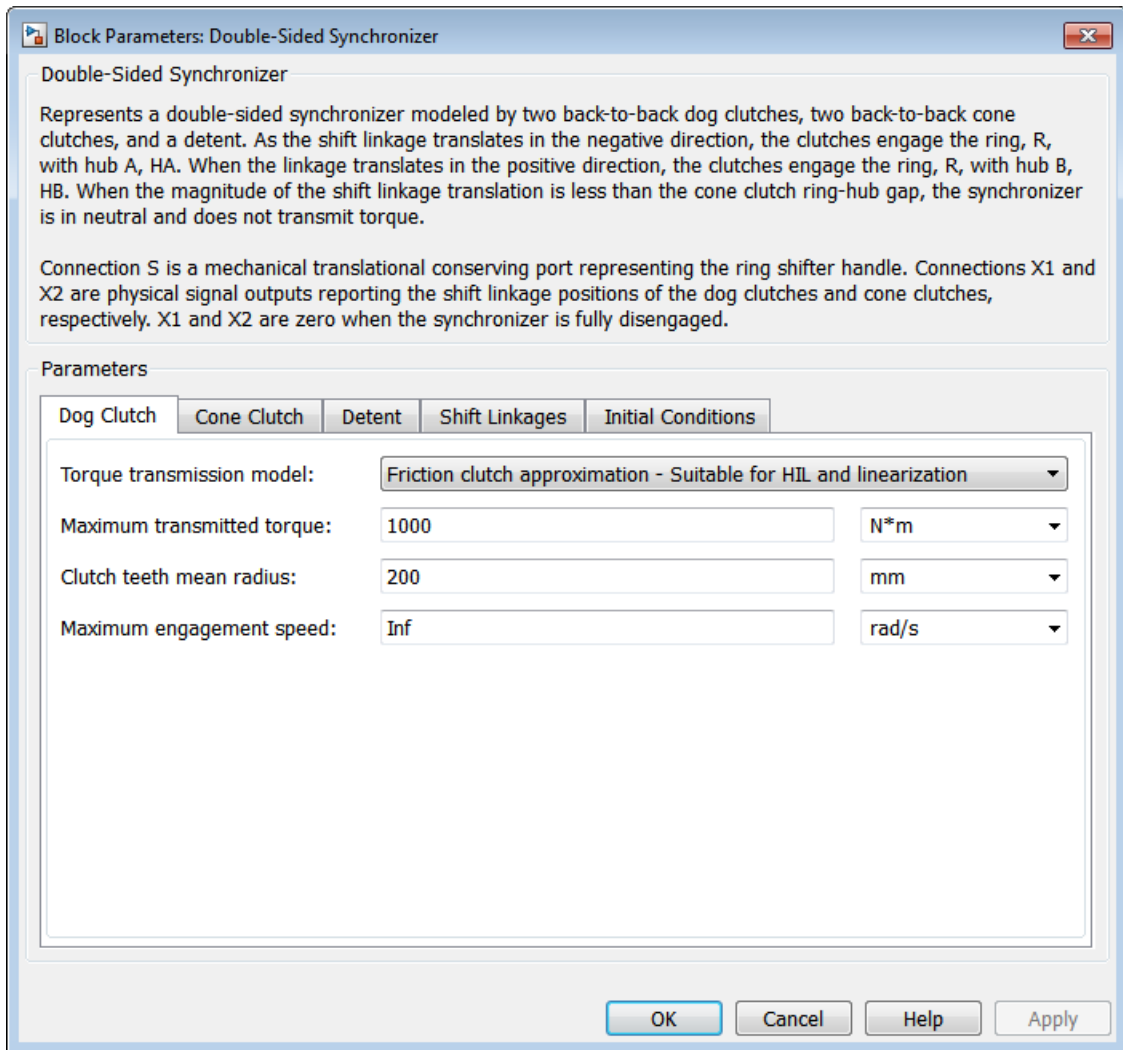
- The model does not account for inertia effects. You can add a Simscape Inertia block at each port to add inertia to the synchronizer model.

# Double-Sided Synchronizer

## Dialog Box and Parameters

The dialog box includes five tabs: **Dog Clutch**, **Cone Clutch**, **Detent**, **Shift Linkages**, and **Initial Conditions**.

### Dog Clutch



## **Torque Transmission Model**

Model used to represent torque transmission. The default setting is Friction clutch approximation - Suitable for HIL and linearization.

Each model provides the following common parameters:

### **Clutch teeth mean radius**

Mean radius of the clutch teeth. The parameter must be greater than zero. The default value is 200 mm.

### **Maximum engagement speed**

The relative angular speed between the ring and hub shafts above which the dog clutch cannot engage. The value is specific to the specific gearbox or transmission and must be minimized to avoid high dynamic impact during engagement. The default value is `inf` rad/s

## **Friction clutch approximation - Suitable for HIL and linearization**

### **Maximum transmitted torque**

Maximum expected torque across the clutch. The default value is 1000 Nm.

## **Dynamic with Backlash**

### **Number of teeth**

Number of teeth in the clutch. The parameter must be greater than or equal to one. The default value is 6.

### **Rotational backlash**

Angular play of the ring with respect to the hub. Greater backlash values correspond to wider tooth gaps that makes engagement easier. The parameter must be greater than zero. The default value is 10 deg.

# Double-Sided Synchronizer

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## **Torsional stiffness**

Stiffness of the tooth contact. The model assumes deformation due to tooth interaction is elastic. The angular deformation at the contact surface is proportional to the applied torque and the reciprocal of the contact stiffness. The parameter must be greater than zero. The default value is  $1e+7$  N\*m/rad.

## **Torsional damping**

Damping at the tooth contact surface. The value of the parameter is inversely proportional to the number of oscillations following impact. The parameter must be greater than zero. The default value is  $2$  N\*m/(rad/s).

## **Tooth-tooth friction coefficient**

Friction coefficient for the translational relative motion between the hub and ring teeth. The value of the parameter depends on the tooth material, lubrication, and quality of contacting surfaces. The parameter must be greater than zero. The default value is  $0.05$ .

## Cone Clutch

**Block Parameters: Double-Sided Synchronizer**

**Double-Sided Synchronizer**

Represents a double-sided synchronizer modeled by two back-to-back dog clutches, two back-to-back cone clutches, and a detent. As the shift linkage translates in the negative direction, the clutches engage the ring, R, with hub A, HA. When the linkage translates in the positive direction, the clutches engage the ring, R, with hub B, HB. When the magnitude of the shift linkage translation is less than the cone clutch ring-hub gap, the synchronizer is in neutral and does not transmit torque.

Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutches and cone clutches, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

**Parameters**

Dog Clutch	Cone Clutch	Detent	Shift Linkages	Initial Conditions
Contact surface maximum diameter:	<input type="text" value="150"/>	<input type="text"/>	<input type="text"/>	<input type="text" value="mm"/>
Contact surface minimum diameter:	<input type="text" value="100"/>	<input type="text"/>	<input type="text"/>	<input type="text" value="mm"/>
Cone half angle:	<input type="text" value="12"/>	<input type="text"/>	<input type="text"/>	<input type="text" value="deg"/>
Kinetic friction coefficient:	<input type="text" value="0.3"/>	<input type="text"/>	<input type="text"/>	<input type="text"/>
Static friction coefficient:	<input type="text" value="0.35"/>	<input type="text"/>	<input type="text"/>	<input type="text"/>
Velocity tolerance:	<input type="text" value="0.001"/>	<input type="text"/>	<input type="text"/>	<input type="text" value="rad/s"/>
Threshold force:	<input type="text" value="1"/>	<input type="text"/>	<input type="text"/>	<input type="text" value="N"/>

OK Cancel Help Apply

# Double-Sided Synchronizer

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## **Contact surface maximum diameter**

Outer diameter of the friction material linen that lines the cone clutch disks. The parameter must be greater than zero. The default value is 150 mm.

## **Contact surface minimum diameter**

Inner diameter of the friction material linen that lines the cone clutch disks. The parameter must be greater than zero, but smaller than the value of **Contact surface maximum diameter**.

## **Cone half angle**

Half angle of the clutch cone. The parameter must be greater than zero and less than or equal to 90°. The default value is 12 deg.

## **Kinetic friction coefficient**

Kinetic friction coefficient between the friction surfaces of the cone clutch disks. The parameter must be greater than zero. The default value is 0.3.

## **Static friction coefficient**

Static friction coefficient between the friction surfaces of the cone clutch disks. The parameter must be greater than the kinetic friction coefficient. The default value is 0.35.

## **Velocity tolerance**

Relative velocity between the cone clutch disks below which the disk friction surfaces lock. Surface locking requires that the torque across the R and H rotational ports be smaller than the product of the effective radius, the static friction coefficient, and the applied normal force. The default value is 0.001 rad/s.

## **Threshold force**

Minimum value of the applied normal force for frictional torque transmittal to occur. If the value of the parameter exceeds the value of the applied normal force, no frictional torque is transmitted.



## Detent

**Block Parameters: Double-Sided Synchronizer**

**Double-Sided Synchronizer**

Represents a double-sided synchronizer modeled by two back-to-back dog clutches, two back-to-back cone clutches, and a detent. As the shift linkage translates in the negative direction, the clutches engage the ring, R, with hub A, HA. When the linkage translates in the positive direction, the clutches engage the ring, R, with hub B, HB. When the magnitude of the shift linkage translation is less than the cone clutch ring-hub gap, the synchronizer is in neutral and does not transmit torque.

Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutches and cone clutches, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

**Parameters**

Dog Clutch	Cone Clutch	Detent	Shift Linkages	Initial Conditions
Peak force:	500	N		
Notch width:	3	mm		
Viscous friction coefficient:	0.1	N/(m/s)		
Friction to peak force ratio:	0.01			
Friction velocity threshold:	0.05	m/s		

OK Cancel Help Apply

# Double-Sided Synchronizer

---

## **Peak force**

Peak shear force of the detent. The default value is 500 N.

## **Notch width**

Width of the region where the detent exhibits shear force. The default value is 3 mm.

## **Viscous friction coefficient**

Viscous friction coefficient at the contact surface of the detent. The parameter must be greater than or equal to zero. The default value is 0.1 N/(m/s).

## **Friction to peak force ratio**

Ratio of the kinetic friction to the peak shear force of the detent. The parameter is used to set the value of the kinetic friction. The parameter must be greater than or equal to zero. The default value is 0.01.

## **Friction velocity threshold**

Velocity required for peak kinetic friction at the contact surface of the detent. The parameter ensures the force is continuous when the travel direction changes, increasing the numerical stability of the simulation. The parameter must be greater than zero. The default value is 0.05 m/s.

## Shift Linkages

**Block Parameters: Double-Sided Synchronizer**

**Double-Sided Synchronizer**

Represents a double-sided synchronizer modeled by two back-to-back dog clutches, two back-to-back cone clutches, and a detent. As the shift linkage translates in the negative direction, the clutches engage the ring, R, with hub A, HA. When the linkage translates in the positive direction, the clutches engage the ring, R, with hub B, HB. When the magnitude of the shift linkage translation is less than the cone clutch ring-hub gap, the synchronizer is in neutral and does not transmit torque.

Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutches and cone clutches, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

**Parameters**

Dog Clutch	Cone Clutch	Detent	Shift Linkages	Initial Conditions
Ring-hub clearance when dog clutch disengaged:	5	mm		
Tooth overlap to engage:	3	mm		
Tooth height:	10	mm		
Ring-hub clearance when cone clutch disengaged:	3	mm		
Dog clutch ring stop stiffness:	1e+6	N/m		
Cone clutch ring stop stiffness:	1e+6	N/m		
Dog clutch ring stop damping:	1000	N/(m/s)		
Cone clutch ring stop damping:	1000	N/(m/s)		
Viscous friction coefficient:	100	N/(m/s)		

OK Cancel Help Apply

# Double-Sided Synchronizer

---

## **Ring-hub clearance when dog clutch disengaged**

Distance between ring and dog clutch hub with the ring fully retracted. The parameter specifies the maximum travel distance if engagement requirements are not met. The parameter must be greater than the value of **Ring-hub clearance when cone clutch disengaged**. The default value is 5 mm.

## **Tooth overlap to engage**

Minimum value of the tooth overlap beyond which the dog clutch is considered engaged. The parameter must be greater than zero. The default value is 3 mm.

## **Tooth height**

Height of dog clutch teeth. The parameter must be greater than zero. The default value is 10 mm.

## **Ring-hub clearance when cone clutch disengaged**

Distance between the ring and cone clutch hub when the ring is fully retracted. The cone clutch can begin to engage once the slider has translated this distance. The ring can travel only this distance if the engagement requirements are not satisfied. The parameter must be greater than zero. The default value is 3 mm.

## **Dog clutch ring stop stiffness**

Stiffness of the hard stops on both sides of the dog clutch ring. The model assumes the ring and stops behave elastically. Contact deformation is proportional to the applied force and the reciprocal of the contact stiffness. The value of the stiffness must be assigned with reference to the parameter **Tooth overlap to engage**. Too low a stiffness could cause the deformation to exceed the required overlap and initiate a false engagement. The parameter must be greater than zero. The default value is  $1e+6$  N/m.

## **Cone clutch ring stop stiffness**

Stiffness of the hard stops on both sides of the cone clutch ring. The model assumes the ring and stops behave elastically. Contact deformation is proportional to the applied force and the reciprocal of the contact stiffness. The parameter must be greater than zero. The default value is  $1e+6$  N/m.

## **Dog clutch ring stop damping**

Translational contact damping between the dog clutch ring and the hub. The value of the damping is inversely proportional to the number of oscillations that occur after impact. The parameter must be greater than zero. The default value is  $1e+3$  N/(m/s).

## **Cone clutch ring stop damping**

Translational contact damping between the cone clutch ring and the hub. The value of damping is inversely proportional to the number of oscillations that occur after impact. The parameter must be greater than zero. The default value is  $1e+3$  N/(m/s).

## **Viscous friction coefficient**

Viscous friction coefficient for the relative translational motion between the hub and the ring. The value of the parameter depends on lubrication state and quality of contacting surfaces. The coefficient must be greater than or equal to zero. The default value is 100 N/(m/s).

# Double-Sided Synchronizer

## Initial Conditions

**Block Parameters: Double-Sided Synchronizer**

Double-Sided Synchronizer

Represents a double-sided synchronizer modeled by two back-to-back dog clutches, two back-to-back cone clutches, and a detent. As the shift linkage translates in the negative direction, the clutches engage the ring, R, with hub A, HA. When the linkage translates in the positive direction, the clutches engage the ring, R, with hub B, HB. When the magnitude of the shift linkage translation is less than the cone clutch ring-hub gap, the synchronizer is in neutral and does not transmit torque.

Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutches and cone clutches, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

Parameters

Dog Clutch   Cone Clutch   Detent   Shift Linkages   **Initial Conditions**

Initial state: All clutches unlocked

Dog clutch initial shift linkage position: 0 mm

Cone clutch initial shift linkage position: 0 mm

OK   Cancel   Help   Apply

## Initial state

Beginning configuration of cone and dog clutches. Choices include:

- Cone clutch A and dog clutch A locked
- Cone clutch A locked
- All clutches unlocked
- Cone clutch B locked
- Cone clutch B and dog clutch B locked

The default setting is All clutches unlocked.

## Dog clutch initial shift linkage position

Initial position of the shift linkage section that attaches to the dog clutch. The value of the parameter has the following restrictions:

Dog Clutch State	Parameter Restriction
Dog clutch A Initially engaged	Negative of the parameter value must be greater than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>
Dog clutch A Initially disengaged	Negative of the parameter value must be smaller than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>

# Double-Sided Synchronizer

<b>Dog Clutch State</b>	<b>Parameter Restriction</b>
Dog clutch B Initially engaged	Parameter value must be greater than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>
Dog clutch B Initially disengaged	Parameter value must be smaller than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>

The default value is 0 mm.

## **Cone clutch initial shift linkage position**

Initial position of the shift linkage section that attaches to the cone clutch. The value of the parameter has the following restrictions:

<b>Dog Clutch State</b>	<b>Parameter Restriction</b>
Cone clutch A initially engaged	Negative of parameter must be greater than the value of <b>Ring-hub clearance when cone clutch disengaged</b>
Cone clutch A initially disengaged	Negative of parameter must be smaller than the value of <b>Ring-hub clearance when cone clutch disengaged</b>



# Double-Sided Synchronizer

Dog Clutch State	Parameter Restriction
Cone clutch B initially engaged	Parameter must be greater than the value of <b>Ring-hub clearance when cone clutch disengaged</b>
Cone clutch B initially disengaged	Parameter must be smaller than the value of <b>Ring-hub clearance when cone clutch disengaged</b>

The default value is 0 mm.

## Initial dog clutch ring-hub offset angle

Initial angle between the ring and hub portions of the dog clutch. To edit the parameter, you must set **Dog Clutch > Torque transmission model** to **Dynamic with backlash**. If the clutch is disengaged, the initial angle must lie in the range  $-\pi/N \leq \theta \leq +\pi/N$ , where  $N$  denotes the number of teeth in the dog clutch. If the clutch is engaged, the initial angle must be in the range  $-\delta/2 \leq \theta \leq +\delta/2$ , where  $\delta$  is the backlash angle. The default value is 0 deg.

## Ports

Port	Description
S	Conserving translational port that represents the shift linkage
R	Conserving rotational port that represents the ring shaft
HA	Conserving rotational port that represents hub shaft A
HB	Conserving rotational port that represents hub shaft B
X1	Physical signal output port that measures the magnitude of the dog clutch translation
X2	Physical signal output port that measures the magnitude of the cone clutch translation

# Double-Sided Synchronizer

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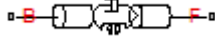
## **See Also**

SynchronizerCone Clutch, Dog Clutch, Translational Detent

**Purpose** Torsionally-compliant driveshaft

**Library** Couplings & Drives

## Description



The Flexible Shaft block models the torsional dynamics of a flexible shaft. The shaft is represented by an approximate lumped parameter circuit consisting of  $N$  segments.  $N$  is a finite integer greater than zero.

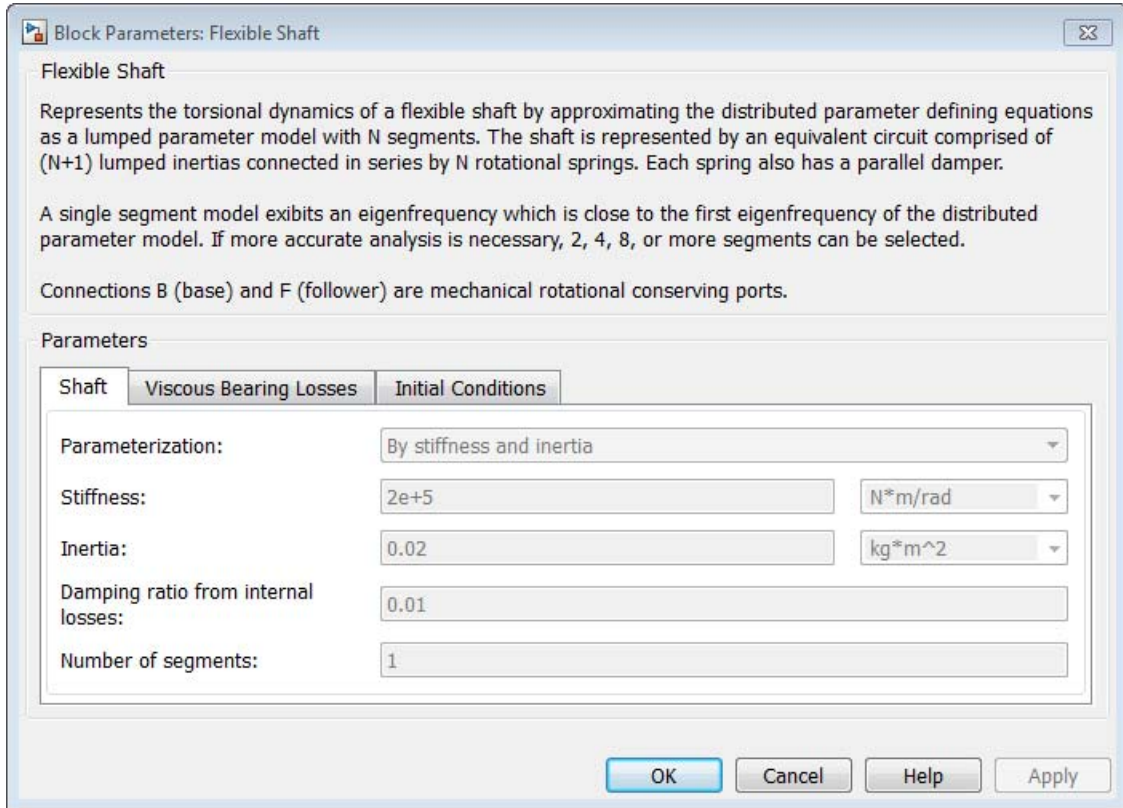
You can parameterize the block in either the overall stiffness and inertia or the dimensions and material properties of the shaft. You can also apply optional viscous friction at the base and follower ports to represent bearing losses at the shaft ends.

## Ports and Conventions

B and F are rotational conserving ports associated with the shaft input and output sections, respectively.

# Flexible Shaft

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

## Shaft

Shaft	Viscous Bearing Losses	Initial Conditions
Parameterization:	By stiffness and inertia	
Stiffness:	2e+5	N*m/rad
Inertia:	0.02	kg*m^2
Damping ratio from internal losses:	0.01	
Number of segments:	1	

### Parameterization

Select how to characterize the flexible shaft. The default is **By stiffness and inertia**.

- **By stiffness and inertia** — Specify shaft characteristics by its inertia and elastic stiffness.

### Stiffness and Inertia

#### Stiffness

Stiffness  $k$  of the shaft, the torque per unit angular deflection of the shaft. The default is  $2e+5$ .

From the drop-down list, choose units. The default is newton-meters/radian (N\*m/rad).

#### Inertia

Rotational inertia  $J$  of the shaft around its principal axis. The default is  $0.02$ .

From the drop-down list, choose units. The default is kilogram-meters<sup>2</sup> (kg·m<sup>2</sup>).

- **By material properties** — Specify shaft characteristics by its size and continuum properties. If you select this option, the panel changes from its default.

# Flexible Shaft

automate

The image shows a software dialog box for configuring shaft parameters. It has three tabs: 'Shaft', 'Viscous Bearing Losses', and 'Initial Conditions'. The 'Shaft' tab is active. The parameters are as follows:

Parameter	Value	Units
Parameterization:	By material properties	
Shaft geometry:	Solid	
Shaft length:	1	m
Shaft outside diameter:	0.075	m
Material density:	7.8e+3	kg/m <sup>3</sup>
Shear modulus:	7.93e+10	Pa
Damping ratio from internal losses:	0.01	
Number of segments:	1	

## Shaft Geometry

Select the geometry of the shaft. The default is **Solid**.

- **Solid** — Specify a solid shaft geometry.

### **Solid**

#### **Shaft outside diameter**

Diameter  $D$  of the solid shaft. Must be greater than 0. The default is **0.075**.

From the drop-down list, choose units. The default is meters (m).

**Annular**— Specify an annular, or hollow, shaft geometry. If you select this option, the panel changes from its default.

## Annular

Shaft	Viscous Bearing Losses	Initial Conditions
Parameterization:	By material properties	
Shaft geometry:	Annular	
Shaft length:	1	m
Shaft outside diameter:	0.075	m
Shaft inside diameter:	0.05	m
Material density:	7.8e+3	kg/m <sup>3</sup>
Shear modulus:	7.93e+10	Pa
Damping ratio from internal losses:	0.01	
Number of segments:	1	

### Shaft outside diameter

Outside diameter  $D$  of the hollow shaft. Must be greater than 0. The default is 0.075.

From the drop-down list, choose units. The default is meters (m).

### Shaft inside diameter

Inside diameter  $d$  of the hollow shaft. Must be greater than 0. The default is 0.05.

From the drop-down list, choose units. The default is meters (m).

### Shaft length

Length  $L$  of the shaft. Must be greater than 0. The default is 1.

From the drop-down list, choose units. The default is meters (m).

# Flexible Shaft

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## Material density

Mass density  $\rho$  of the shaft material. Must be greater than 0. The default is  $7.8e+3$ .

From the drop-down list, choose units. The default is kilograms/meter<sup>3</sup> (kg/m<sup>3</sup>).

## Shear modulus

Shear modulus  $G$  of the shaft material. Must be greater than 0. The default is  $7.93e+10$ .

From the drop-down list, choose units. The default is pascals (Pa).

## Damping ratio from internal losses

Damping ratio  $c$  for the first flexible torsional mode. The default value is 0.01.

## Number of segments

Number  $Nof$  rigid segments into which the shaft is divided. The default is 1.

## Viscous Bearing Losses

Shaft Viscous Bearing Losses Initial Conditions

Viscous friction coefficients at base (B) and follower (F): [ 0 0 ] N\*m/(rad/s)

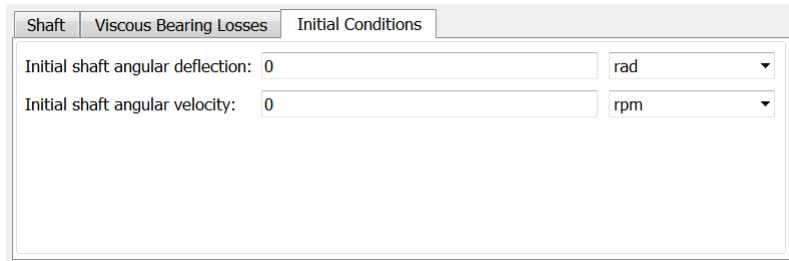
## Viscous friction coefficients at base and follower

Viscous friction coefficients applied at the base and follower, respectively. The default is [0 0].



From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Initial Conditions



Shaft	Viscous Bearing Losses	Initial Conditions
Initial shaft angular deflection: 0 <input type="text"/> rad <input type="button" value="v"/>		
Initial shaft angular velocity: 0 <input type="text"/> rpm <input type="button" value="v"/>		

### Initial shaft angular deflection

Initial torsional angular deflection of the shaft. The default is 0.

From the drop-down list, choose units. The default is radians (rad).

A positive initial deflection results in a positive torque action from the base (B) to the follower (F) port.

### Initial shaft angular velocity

Initial torsional angular velocity of the shaft. The default is 0.

From the drop-down list, choose units. The default is revolutions/minute (rpm).

At the start of simulation, the entire shaft rotates collectively at this angular velocity, with no relative motion between the segments.

## Flexible Shaft Model

The Flexible Shaft block approximates the distributed, continuous properties of a shaft by a lumped parameter model. The model contains a finite number,  $N$ , of lumped inertia-damped spring elements in series, plus a final inertia. The result is a series of  $N+1$  inertias connected by  $N$

# Flexible Shaft

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rotational springs and  $N$  rotational dampers. The block can also include viscous friction at the shaft ends (base and follower ports) to represent bearing losses at these points. Do not confuse this viscous friction at the shaft ends with the internal material damping which corresponds to losses arising in the shaft material itself.

The Flexible Shaft block model is parameterized in either the shaft stiffness  $k$  and inertia  $J$  or its dimensions and material properties.

## Shaft Characterized by Dimensions and Material Properties

The shaft stiffness and inertia are computed from the shaft dimensions and material properties by the following relationships:

$$J_p = (\pi/32)(D^4 - d^4) ,$$

$$m = (\pi/4)(D^2 - d^2)\rho L ,$$

$$J = (m/8)(D^2 + d^2) = \rho L \cdot J_p ,$$

$$k = J_p \cdot G/L ,$$

where:

$J_p$  Polar moment of inertia

$D$  Shaft outside diameter

$d$  Shaft inside diameter

For solid shafts:  $d = 0$

For annular (hollow) shafts:  $d > 0$

$L$  Shaft length

$m$  Shaft mass

$J$  Moment of inertia

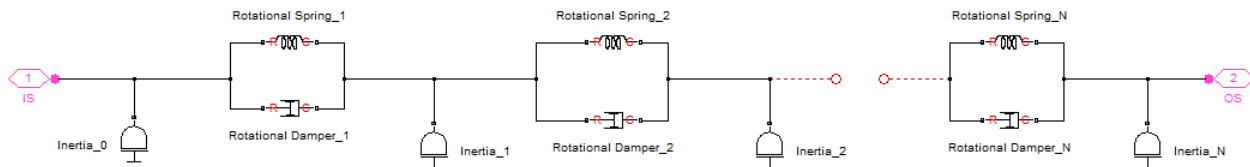
- $\rho$  Shaft material density
- $G$  Shear modulus of elasticity
- $k$  Shaft rotational stiffness

## Internal Material Damping

For either shaft parameterization, the internal material damping is defined by the damping ratio,  $c$ , for a single-segment model. In this case, the damping torque is  $2c/\omega_N$ .  $\omega_N$  is the undamped natural frequency =  $\sqrt{(2k/J)}$ . For an  $N$ -segment model, the damping applied across each of the  $N$  springs is  $2cN/\omega_N$ .

## Equivalent Physical Network

The following figure shows an equivalent physical network constructed from Simscape blocks only. There are  $N$  segments, each consisting for a spring, damper, and inertia. A segment represents a short section of the driveshaft, the spring representing torsional compliance and the damper representing material damping. The total shaft inertia is split into  $N+1$  parts, and partitioned as shown in the figure.



## Limitations

The distributed parameter model of a continuous torsional shaft is approximated by a finite number,  $N$ , of lumped parameters.

The flexible shaft is assumed to have a constant cross-section along its length.

## Tradeoff Between Accuracy and Speed

A larger number  $N$  of segments increases the accuracy of the model, but reduces its speed. The single-segmented model ( $N=1$ ) exhibits

# Flexible Shaft

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an eigenfrequency which is close to the first eigenfrequency of the continuous, distributed parameter model.

For greater accuracy, you can select 2, 4, 8, or more segments. For example, the four lowest eigenfrequencies are represented with an accuracy of 0.1, 1.9, 1.6, and 5.3 percent, respectively, by a 16-segmented model.

## References

[1] Bathe, K.-J., *Finite Element Procedures*, Prentice Hall, Inc, 1996.

[2] Chudnovsky, V., D. Kennedy, A. Mukherjee, and J. Wendlandt, "Modeling Flexible Bodies in SimMechanics and Simulink," *MATLAB Digest* 14(3), May 2006.

## See Also

Inertia, Rotational Spring, Rotational Damper, Torsional Spring-Damper

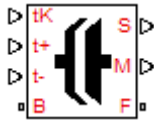
## Purpose

Friction clutch controlled by kinetic and upper and lower static locking friction signals

## Library

Clutches

## Description



A friction clutch transfers motion and torque between two driveline axes by coupling them with friction. The Fundamental Friction Clutch block models a standard friction clutch with kinetic friction and static (locking) friction acting on the two axes. For model details, see “Fundamental Friction Clutch Model” on page 1-129.

## Ports

B and F are rotational conserving ports representing, respectively, the clutch input (base) and output (follower) driveshaft axes. The clutch motion is measured as the *slip*  $\omega = \omega_F - \omega_B$ , the angular velocity of follower relative to base.

The clutch requires three physical signal inputs (all in newton-meters):

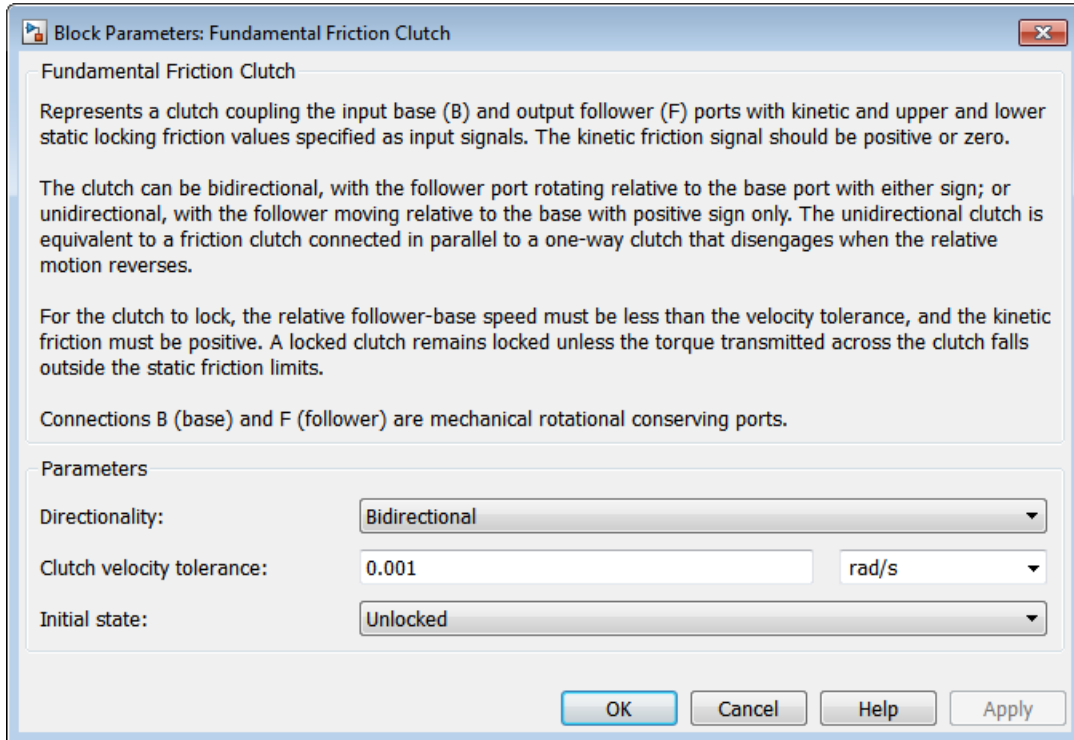
- Kinetic friction torque  $\tau_K \geq 0$  (port tK)
- Static friction upper limit torque  $\tau_S^+$  (port t+)
- Static friction lower limit torque  $\tau_S^-$  (port t-)

The clutch generates two physical signal outputs:

- Clutch slip  $\omega$  (port S)
- Clutch state or mode (port M)

# Fundamental Friction Clutch

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**.

### Directionality

Select **Bidirectional** or **Unidirectional** to determine how the follower axis can turn relative to the base, in both directions or only in the forward direction, respectively. The default is **Bidirectional**.

## Clutch velocity tolerance

Sets the minimum absolute slip  $\omega_{\text{Tol}}$  above which the clutch cannot lock. Below this speed, the clutch can lock. (See the diagram, Clutch States and Transitions on page 1-132.) The default is 0.001.

From the drop-down list, choose units. The default is radians/second (rad/s).

## Initial state

From the drop-down list, choose whether the clutch starts simulation in the Locked or the Initial Unlocked state. The default is Unlocked.

## Fundamental Friction Clutch Model

### Clutch Directionality

The friction clutch has two possible directionalities:

- Bidirectional ( $\omega \leq 0$  or  $\omega \geq 0$ ), allowing the follower to rotate relative to the base in either direction
- Unidirectional ( $\omega \geq 0$ ), allowing the follower to rotate relative to the base in the forward direction only.

A unidirectional clutch is equivalent to a friction clutch connected in parallel to a one-way clutch, which disengages when the relative motion reverses.

### Modeling Reverse Unidirectional Clutches

If you want a unidirectional clutch that allows the follower to rotate relative to the base in the reverse direction only, connect the Fundamental Friction Clutch block in your driveline with reversed orientation, follower (F) to base (B).

### Clutch Velocity Tolerance

You set the *clutch velocity tolerance* or threshold  $\omega_{\text{Tol}}$  for each clutch individually.

# Fundamental Friction Clutch

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## Clutch Friction

The Fundamental Friction Clutch block can apply two kinds of friction to driveline motion, kinetic and static, to oppose or prevent slipping of the two axes.

- The clutch applies kinetic friction torque, specified as a positive input signal, only when one driveline axis is spinning relative to the other driveline axis; that is, when the clutch is unlocked and the slip is nonzero.
- The clutch applies static friction torque when the two driveline axes lock and rotate together, without slip.

You specify *static friction limits* as input signals. These positive upper and negative lower limits define a locked range of static friction. If the torque across the clutch remains within this range, the clutch remains locked.

The block iterates through multistep testing to determine when to lock and unlock the clutch.

## Clutch Initial State — Overriding Initial Default State by Manual Locking

When simulation starts, the state of a clutch is either Locked or Initial-Unlocked. If you change the clutch initial state default and require it to be initially locked, the simulation starts with the clutch already in the Locked state, with no initial tests of clutch conditions.

Unlike the Unlocked state, the default Initial Unlocked state lacks a direction of motion. When simulation first starts, the clutch immediately tests its condition to see if it should be:

- Locked or unlocked
- If unlocked, rotating forward or in reverse

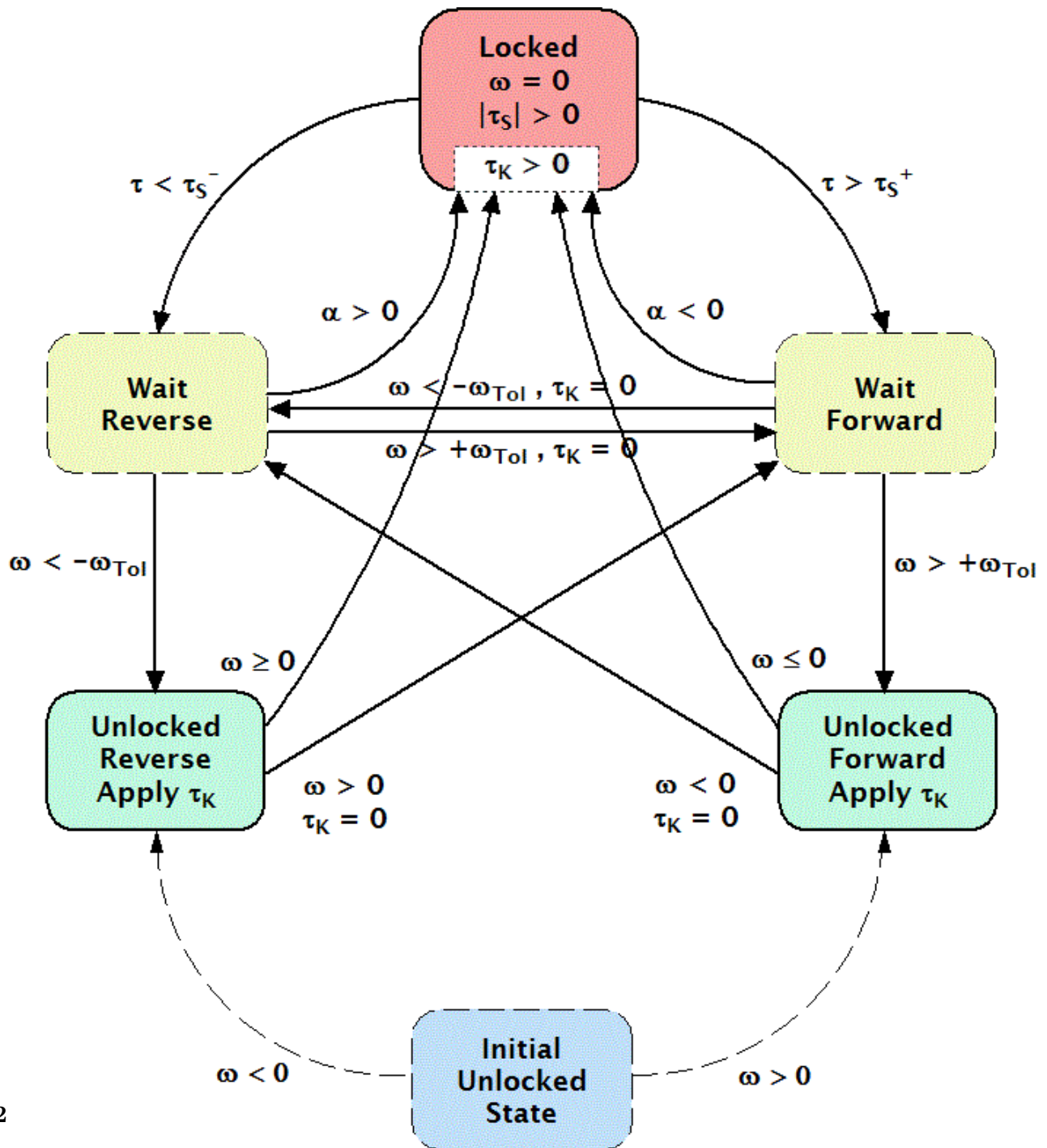
Then the clutch then moves itself to the appropriate state.



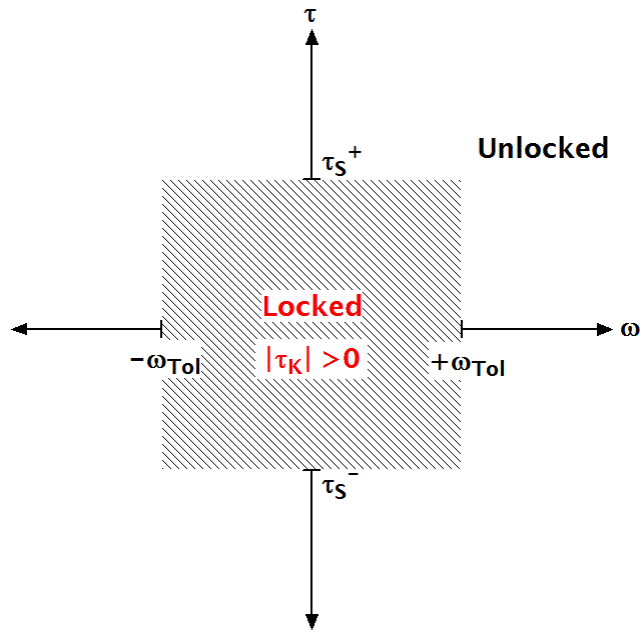
## **Clutch State, Transition, and Variable Summary**

The first chart summarizes the possible states and transitions of a bidirectional clutch. The states and transitions of a unidirectional clutch consist of just the right side of the chart. The second diagram summarizes the physical differences between the locked and unlocked states. The final tables summarize the clutch variables, states, and modes.

# Fundamental Friction Clutch



# Fundamental Friction Clutch



**Clutch Slip vs. Friction Torque**

## Clutch Variables

Symbol	Definition	Input Signal	Significance
$\omega$	Relative angular velocity (slip)		$\omega_F - \omega_B$
$a$	Relative angular acceleration		$d\omega/dt$
$\omega_{Tol}$	Slip tolerance for clutch locking		First locking condition: $ \omega  \leq \omega_{Tol}$
$\tau_K$	Kinetic friction torque	tK	Second locking condition: $\tau_K > 0$

# Fundamental Friction Clutch

## Clutch Variables (Continued)

Symbol	Definition	Input Signal	Significance
$\tau_S^\pm$	Static friction torque limits	$t^\pm$	Defines locked range
$\tau$	Total torque transferred across clutch		Clutch remains locked if $\tau_S^- < \tau < \tau_S^+$ .

## Clutch States and Modes

State	Mode Signal
Forward <i>or</i> Wait Forward	+1
Locked	0
Reverse <i>or</i> Wait Reverse	-1
Initial Unlocked State	0

## Clutch States and Transitions

A friction clutch can be in one of three physical states:

- *Unengaged* ( $\omega \neq 0$  and  $\tau_K = 0$ ), when the clutch applies no friction at all. The frictional surfaces are not in contact. The follower and base are independent, and no torque is transferred between them. No power is dissipated by the clutch in this state.
- *Engaged*, but not locked ( $\omega \neq 0$  and  $\tau_K > 0$ ), when the clutch applies kinetic friction as the frictional surfaces touch and slip past one another. The follower and base remain independent, but some torque is transferred between them.

The clutch dissipates power only in this state. The power dissipated is  $|\omega \tau_K|$ .

- *Locked* ( $\omega = 0$  and  $\tau_K > 0$ ), when the clutch applies static friction. The frictional surfaces lock together and do not slip. The follower and base effectively form a single axis. This state transfers the maximum torque possible. Because static constraints do no work, no power is dissipated by the clutch in this state.

There is also a fourth, virtual state called the *wait state*. See the diagram, Clutch States and Transitions on page 1-132.

## Clutch Locking and Unlocking

Locking requires that the:

- Relative speed (absolute slip)  $|\omega|$  be smaller than a velocity threshold  $\omega_{tol}$ .
- Kinetic friction torque  $\tau_K$  be positive.

The static friction torque controls the unlocking of a friction clutch. When the clutch is locked, it remains locked unless the torque transferred across the clutch exceeds the static friction torque limits.

## Clutches, Constraints, and Degrees of Freedom

If it locks, a Fundamental Friction Clutch block imposes a constraint on your driveline by requiring that two otherwise independent angular velocities be equal. A locked clutch thus reduces the number of independent degrees of freedom by one.

By the same principle, a clutch unlocking restores one independent degree of freedom to a driveline.

A locking clutch imposes a *dynamic constraint* because its constraint can appear and disappear during the simulation.

## Unlocked State: Kinetic Friction

The kinetic friction torque  $\tau_K$  applied between the base and follower driveshafts is specified by the incoming signal at the tK inport. This signal should be positive or zero.

The Fundamental Friction Clutch applies this torque as long as the clutch remains unlocked.

# Fundamental Friction Clutch

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## Locked State: Static Friction

Once the friction clutch locks, it remains locked as long as the total torque  $\tau$  transferred across the clutch remains within the range defined by the static friction torque limits:

$$\tau_S^- < \tau < \tau_S^+ .$$

You specify the static friction torque limits  $\tau_S^\pm$  by the incoming signals at the t+ and t- inports.  $\tau_S^+$  and  $\tau_S^-$  are independent, as long as

$$\tau_S^- < 0 < \tau_S^+ .$$

## How the Friction Clutch Locks and Unlocks

The locking and unlocking of a friction clutch proceed through an intermediate Wait state.

### Wait State

The Wait state is a virtual state that continues the motion of the clutch's previous state but tests for locking or unlocking.

- If the clutch moved to Wait from Locked, it remains locked while in Wait.
- If the clutch moved to Wait from Unlocked, it remains unlocked while in Wait.

### Clutch Locking

The friction clutch locks the two connected driveline axes together when both these conditions hold:

- $\tau_K > 0$
- Either of these conditions:
  - $|\omega| \leq \omega_{Tol}$
  - $\tau$  changes sign while the clutch is unlocked

If the  $\omega$  changes sign while the clutch is unlocked, but  $\tau_K = 0$ , the clutch enters the Wait state. While the clutch is in the Wait state, the driveshafts continue to slip relative to one another, subject to  $\tau_K$ . While in the Wait state, if the clutch locking conditions become true, the clutch moves to Locked.

## Clutch Unlocking

If the total torque across the two driveline axes moves outside the static friction limit range, the clutch enters the Wait state. While the clutch is in the Wait state, it remains locked but tests for unlocking.

The unlocking of a friction clutch is a conditional, multistep process implemented internally through mode iteration. The Wait state encompasses the steps that test the entire driveline for unlocking.

- 1 The block first checks the relative acceleration  $\alpha = d\omega/dt$  of the two driveline axes, based on the torques present when the clutch enters the Wait state.

The clutch returns to the Locked state if:

- The whole driveline requires the axes to turn in the relative forward direction, but  $\alpha$  is negative.
  - The whole driveline requires the axes to turn in the relative reverse direction, but  $\alpha$  is positive.
- 2 If the clutch remains in the Wait state instead of returning to Locked, the relative acceleration is integrated in time to obtain the absolute value of the virtual angular speed. The block checks this result against angular velocity tolerance  $\omega_{Tol}$ . If the result is less than  $\omega_{Tol}$ , the clutch returns to the start of the Wait state and the relative acceleration check. If the result exceeds  $\omega_{Tol}$ , the clutch unlocks.
  - 3 In the Unlocked state, the clutch begins applying kinetic friction again.

# Fundamental Friction Clutch

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**Tip** For more information about mode iteration and solving constraints in Simscape, see “Simulation”.

---

## Examples

The Disk Friction Clutch is a subsystem built from the Fundamental Friction Clutch. For further details, consult its block reference page.

The Disk Friction Clutch and models using it are SimDriveline examples of friction clutches.

## See Also

Differential, Disk Friction Clutch, Torque Converter, Variable Ratio Transmission

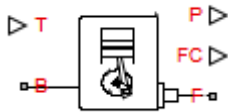
“Gear Coupling Control with Clutches”



**Purpose** Internal combustion engine with throttle and rotational inertia and time lag

**Library** Engines

## Description



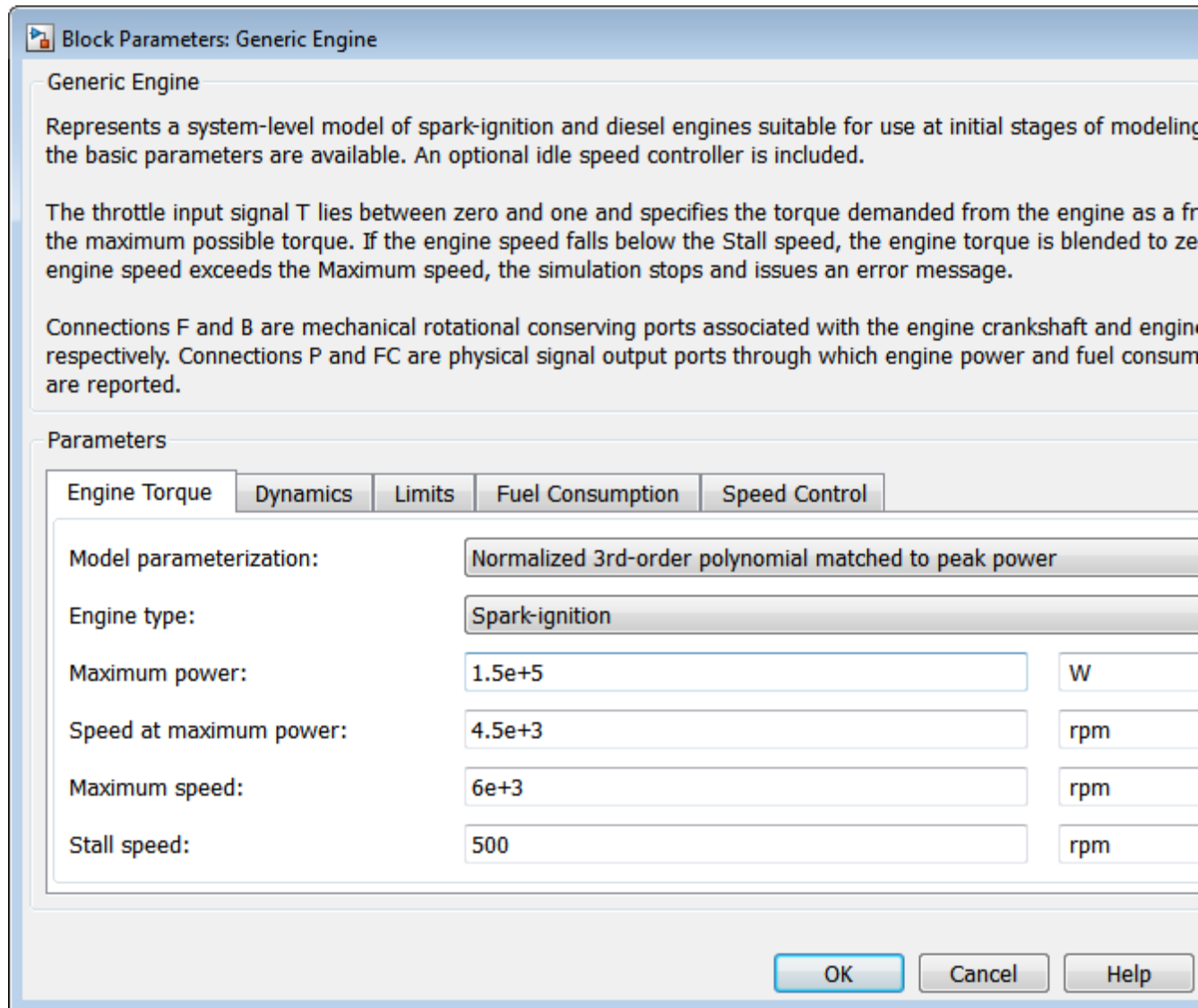
The block represents a general internal combustion engine. Engine types include spark-ignition and diesel. Speed-power and speed-torque parameterizations are provided. A throttle physical signal input specifies the normalized engine torque. Optional dynamic parameters include crankshaft inertia and response time lag. A physical signal port outputs engine fuel consumption rate based on choice of fuel consumption model. An optional speed controller prevents engine stall and enables cruise control. See “Generic Engine Model” on page 1-164.

# Generic Engine

## Dialog Box and Parameters

The dialog box contains five tabs, **Engine Torque**, **Dynamics**, **Limits**, **Fuel Consumption**, and **Speed Control**.

### Engine Torque



## Model parametrization

Select how to model the engine. The default is Normalized 3rd-order polynomial matched to peak power.

- Normalized 3rd-order polynomial matched to peak power — Parametrize the engine with a power function controlled by power and speed characteristics.

## Normalized Engine Power Polynomial

### Engine type

Choose type of internal combustion engine. The default is Spark-ignition.

### Maximum power

Maximum power  $P_{\max}$  that the engine can output. The default is 150000.

From the drop-down list, choose units. The default is watts (W).

### Speed at maximum power

Engine speed  $\Omega_0$  at which the engine is running at maximum power. The default is 4500.

From the drop-down list, choose units. The default is revolutions per minute (rpm).

### Maximum speed

Maximum speed  $\Omega_{\max}$  at which the engine can generate torque. The default is 6000.

From the drop-down list, choose units. The default is revolutions per minute (rpm).

During simulation, if  $\Omega$  exceeds this maximum, the simulation stops with an error.

---

**Note** The engine maximum speed  $\Omega_{\max}$  cannot exceed the engine speed at which the engine power becomes negative. For more information, see “Generic Engine Model” on page 1-164.

---

## Stall speed

Minimum speed  $\Omega_{\min}$  at which the engine can generate torque. The default is 500.

From the drop-down list, choose units. The default is revolutions per minute (rpm).

During simulation, if  $\Omega$  falls below this minimum, the engine torque is blended to zero.

- Tabulated torque data — Engine is parametrized by speed–torque table that you specify. If you select this option, the panel changes from its default.

## Tabulated Torque Data

**Block Parameters: Generic Engine**

**Generic Engine**

Represents a system-level model of spark-ignition and diesel engines suitable for use at initial stages of model development when the basic parameters are available. An optional idle speed controller is included.

The throttle input signal  $T$  lies between zero and one and specifies the torque demanded from the engine as a fraction of the maximum possible torque. If the engine speed falls below the Stall speed, the engine torque is blended to zero. If the engine speed exceeds the Maximum speed, the simulation stops and issues an error message.

Connections  $F$  and  $B$  are mechanical rotational conserving ports associated with the engine crankshaft and engine flywheel respectively. Connections  $P$  and  $FC$  are physical signal output ports through which engine power and fuel consumption are reported.

**Parameters**

Engine Torque | Dynamics | Limits | Fuel Consumption | Speed Control

Model parameterization: Tabulated torque data

Speed vector: [ 500 1000 2e+3 3e+3 4e+3 5e+3 6e+3 7e+3 ] rpm

Torque vector: [ 380 380 380 380 350 280 200 80 ] N\*m

Interpolation method: Linear

OK Cancel Help

## Speed vector

Vector of values of the engine function's independent variable, the speed  $\Omega$ .

From the drop-down list, choose units. The default is revolutions per minute (rpm).

---

**Note** In this vector, the first and last speeds are interpreted as the stall speed and the maximum speed, respectively. If the speed falls below the stall speed, engine torque is blended to zero. If the speed exceeds the maximum speed, the simulation stops with an error.

---

## Torque vector

Vector of values of the engine function's dependent variable, the torque  $\tau$ .

From the drop-down list, choose units. The default is newton-meters (N\*m).

## Interpolation method

Method to interpolate the engine speed–torque or speed–power function between discrete relative velocity values within the range of definition. The default is Linear.

- **Tabulated power data** — Engine is parametrized by speed-power table that you specify. If you select this option, the panel changes from its default.

## Tabulated Power Data

**Block Parameters: Generic Engine**

**Generic Engine**

Represents a system-level model of spark-ignition and diesel engines suitable for use at initial stages of model development when the basic parameters are available. An optional idle speed controller is included.

The throttle input signal  $T$  lies between zero and one and specifies the torque demanded from the engine as a fraction of the maximum possible torque. If the engine speed falls below the Stall speed, the engine torque is blended to zero. If the engine speed exceeds the Maximum speed, the simulation stops and issues an error message.

Connections  $F$  and  $B$  are mechanical rotational conserving ports associated with the engine crankshaft and engine flywheel respectively. Connections  $P$  and  $FC$  are physical signal output ports through which engine power and fuel consumption are reported.

**Parameters**

Engine Torque | Dynamics | Limits | Fuel Consumption | Speed Control

Model parameterization: Tabulated power data

Speed vector: [ 500 1000 2e+3 3e+3 4e+3 5e+3 6e+3 7e+3 ] rpm

Power vector: 4e+4 7.8e+4 1.2e+5 1.45e+5 1.48e+5 1.25e+5 6e+4 ] W

Interpolation method: Linear

OK Cancel Help

## Speed vector

Vector of values of the engine function's independent variable, the speed  $\Omega$ .

From the drop-down list, choose units. The default is revolutions per minute (rpm).

---

**Note** In this vector, the first and last speeds are interpreted as the stall speed and the maximum speed, respectively. If the speed falls below the stall speed, engine torque is blended to zero. If the speed exceeds the maximum speed, the simulation stops with an error.

---

## Power vector

Vector of values of the engine function's dependent variable, the power  $P$ .

From the drop-down list, choose units. The default is watts (W).

## Interpolation method

Method to interpolate the engine speed–torque or speed–power function between discrete relative velocity values within the range of definition. The default is Linear.

## Dynamics

### Inertia

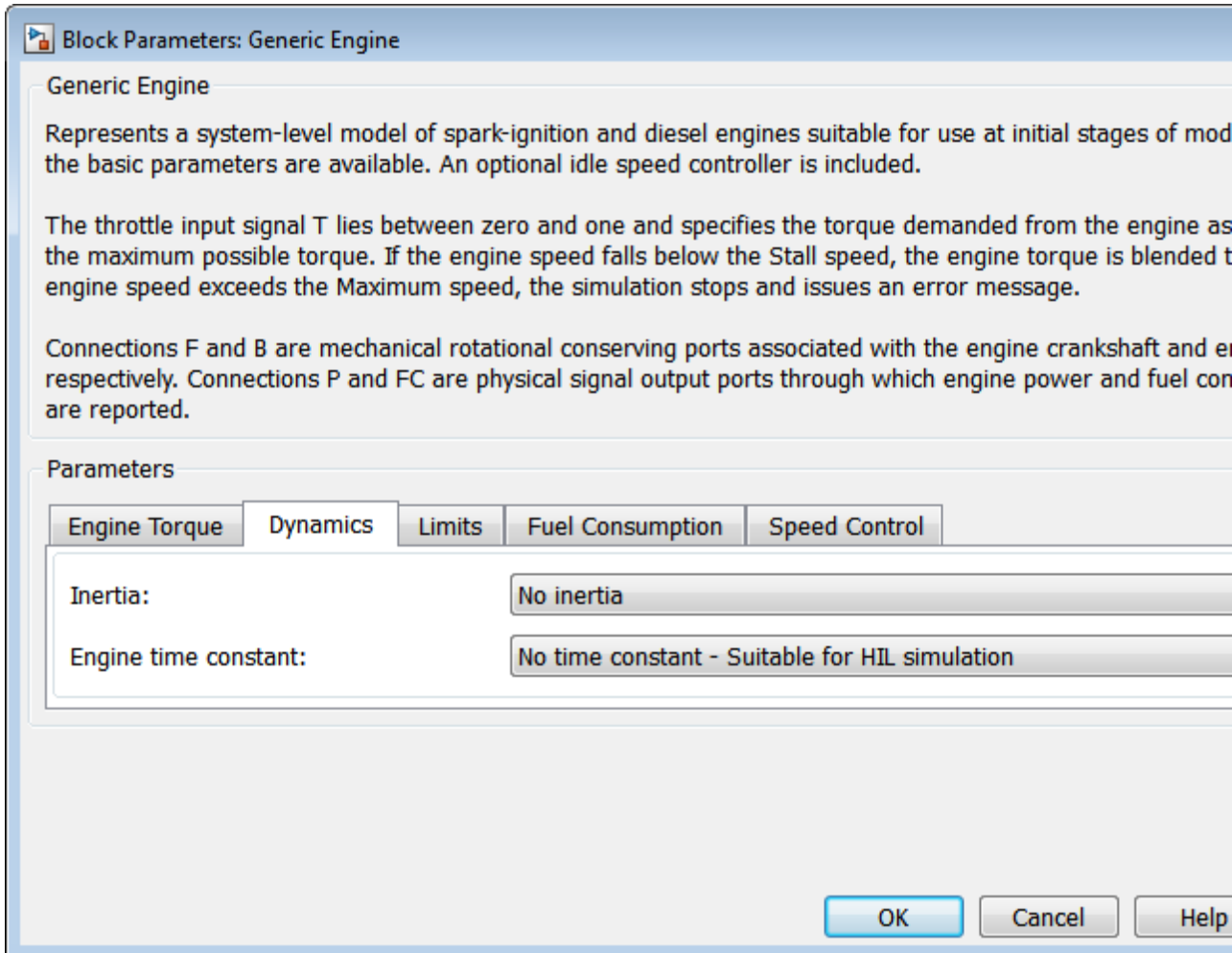
Select how to model the rotational inertia of the engine block. The default is No inertia.

- No inertia — Engine crankshaft is modeled with no inertia.
- Specify inertia and initial velocity — Engine crankshaft is modeled with rotational inertia and initial



angular velocity. If you select this option, the panel changes from its default.

## Rotational Inertia and Initial Velocity



## **Inertia**

Rotational inertia of the engine crankshaft.

From the drop-down list, choose units. The default is kilograms-meters-squared ( $\text{kg}\cdot\text{m}^2$ ).

## **Initial velocity**

Initial angular velocity  $\Omega(0)$  of the engine crankshaft. The default is 800.

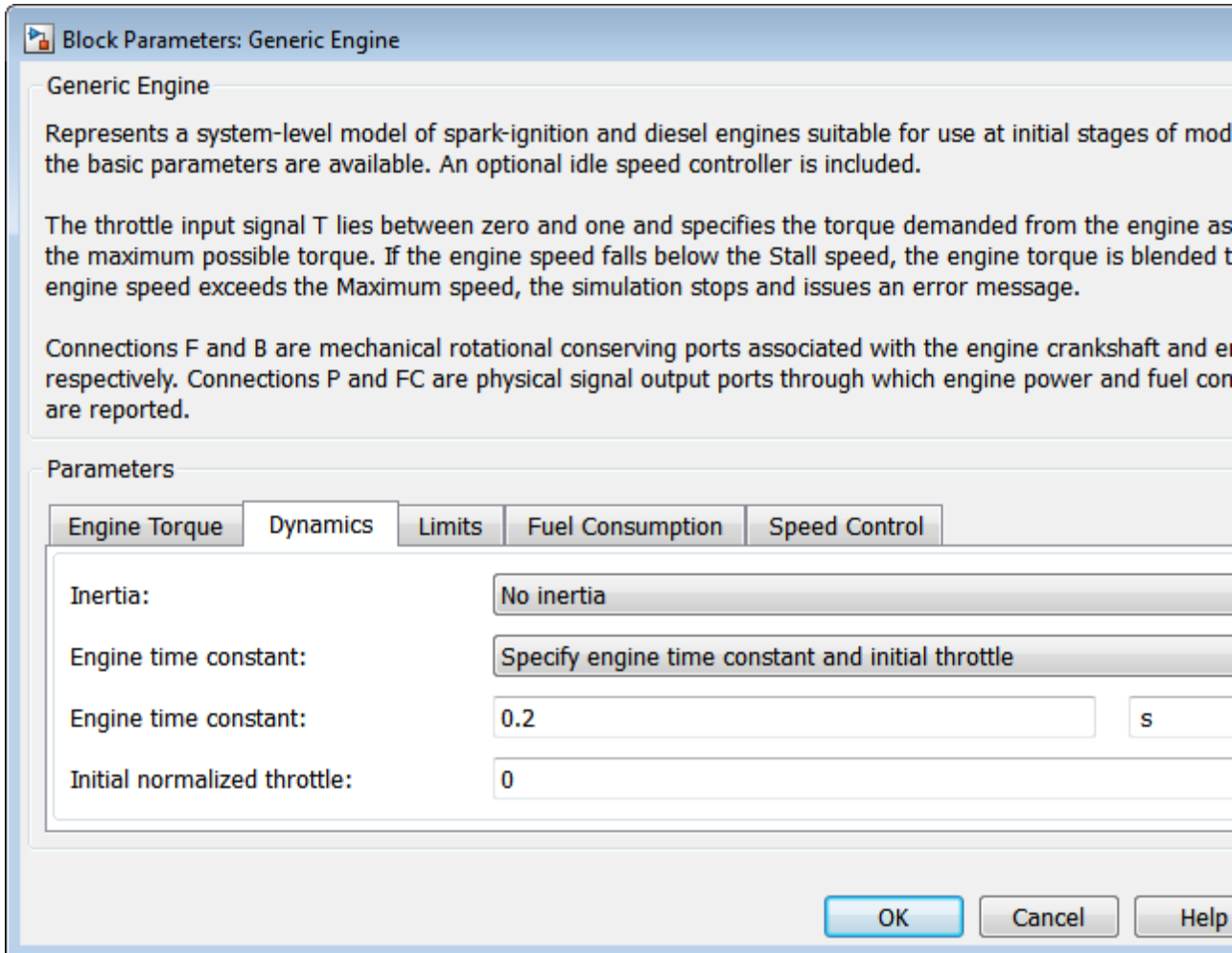
From the drop-down list, choose units. The default is revolutions per minute (rpm).

## **Engine time constant**

Select how to model the time lag of the engine response. The default is No time constant Suitable for HIL simulation.

- No time constant Suitable for HIL simulation — Engine reacts with no time lag.
- Specify engine time constant and initial throttle — Engine reacts with a time lag. If you select this option, the panel changes from its default.

## Engine Time Constant and Initial Throttle



### Engine time constant

Engine time lag. The default is 0.2.

# Generic Engine

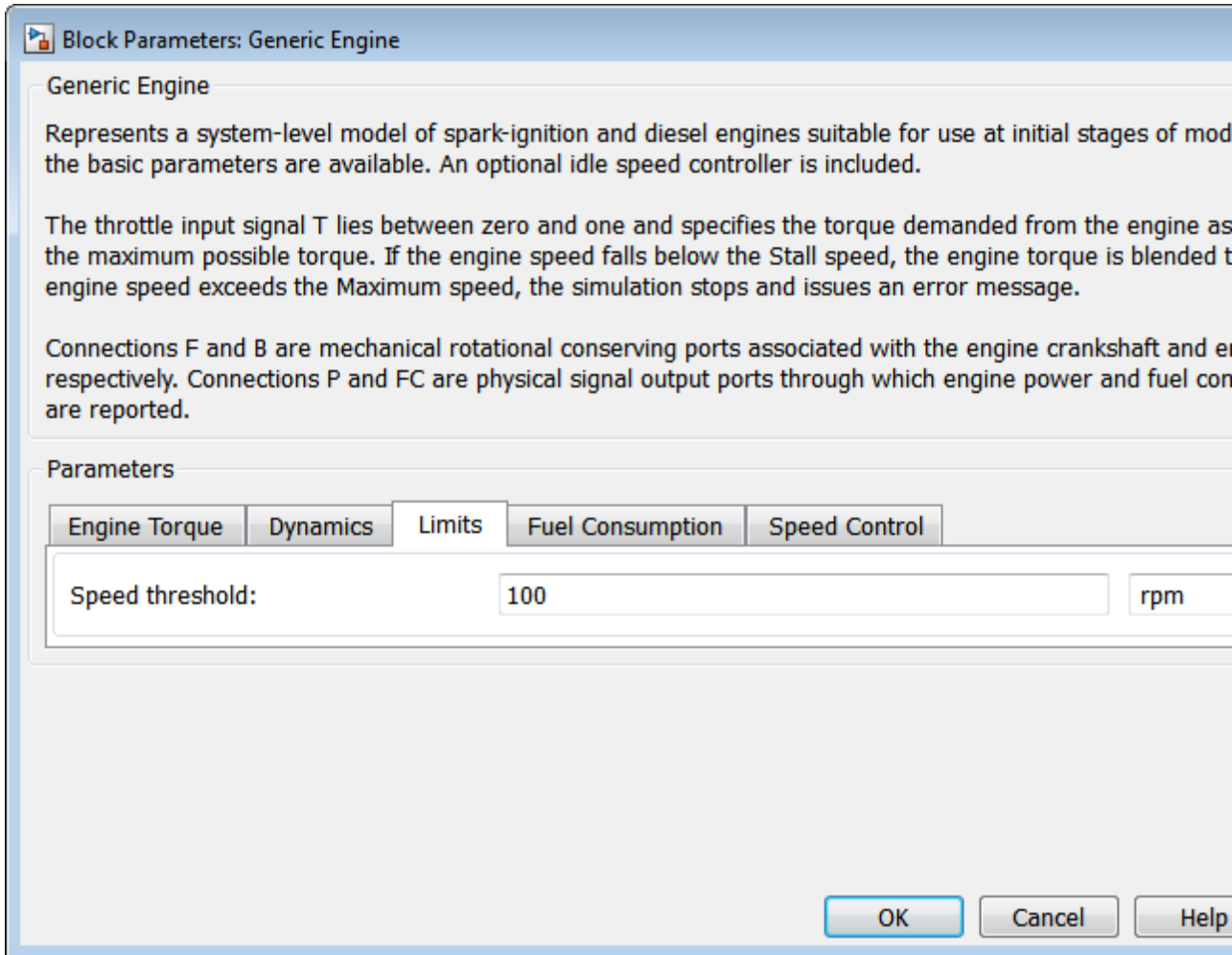
---

From the drop-down list, choose units. The default is seconds (s).

## **Initial normalized throttle**

Initial normalized engine throttle  $T(0)$ , ranging between zero and one. The default is 0.

## Limits



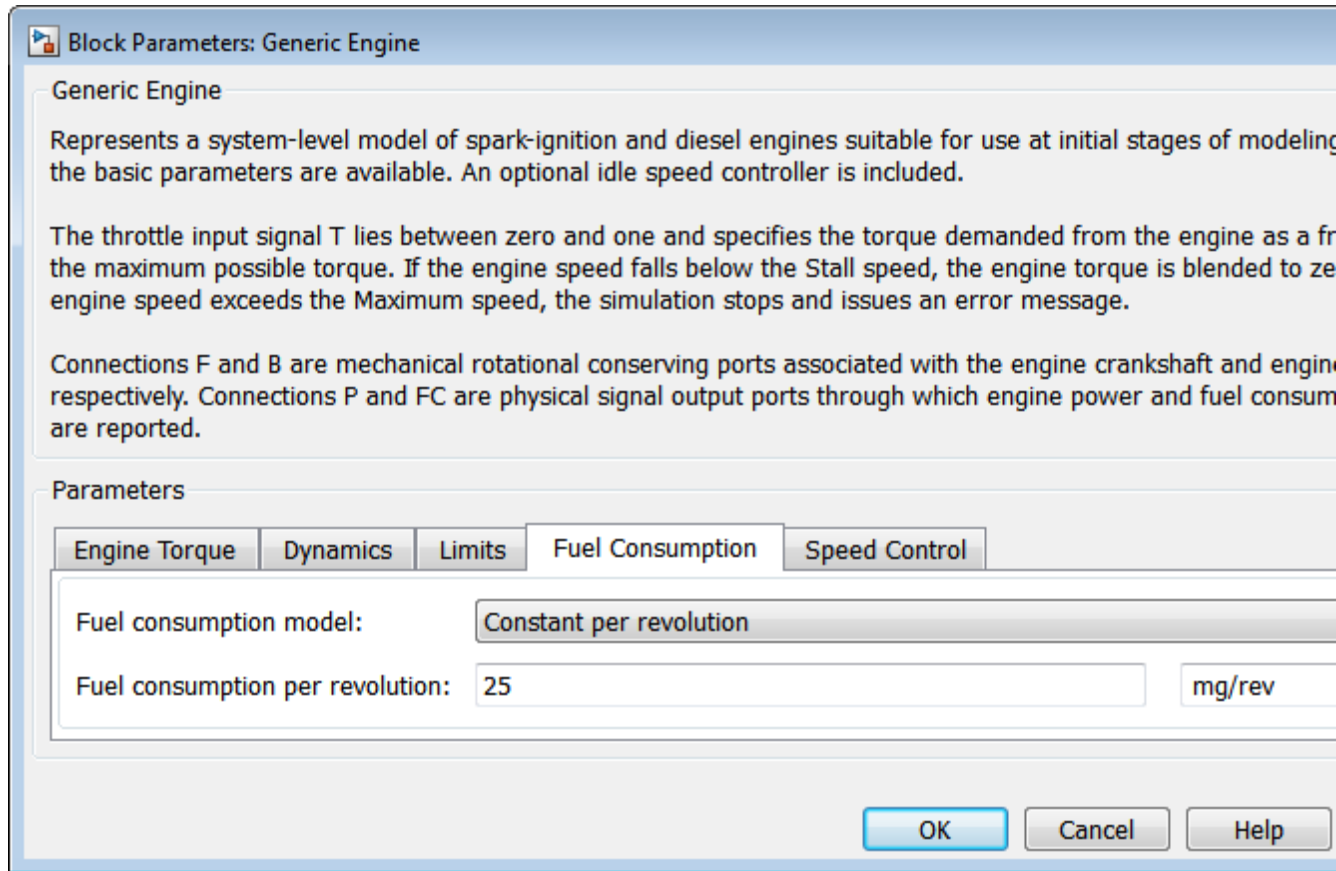
### Speed threshold

Width of the speed range over which the engine torque is blended to zero as  $\Omega$  approaches the stall speed. The default is 100.

# Generic Engine

From the drop-down list, choose units. The default is revolutions per minute (rpm).

## Fuel Consumption



### Fuel consumption model

Select model to specify engine fuel consumption. Models range from simple to advanced parameterizations compatible with

standard industrial data. The default model is Constant per revolution.

## **Constant per revolution**

### **Fuel consumption per revolution**

Enter the volume of fuel consumed in one crankshaft revolution. The default value is 25. the default unit is mg/rev.

# Generic Engine

## Fuel consumption by speed and torque

**Block Parameters: Generic Engine**

**Generic Engine**

Represents a system-level model of spark-ignition and diesel engines suitable for use at initial stages of modeling the basic parameters are available. An optional idle speed controller is included.

The throttle input signal  $T$  lies between zero and one and specifies the torque demanded from the engine as a fraction of the maximum possible torque. If the engine speed falls below the Stall speed, the engine torque is blended to zero. If the engine speed exceeds the Maximum speed, the simulation stops and issues an error message.

Connections  $F$  and  $B$  are mechanical rotational conserving ports associated with the engine crankshaft and engine flywheel respectively. Connections  $P$  and  $FC$  are physical signal output ports through which engine power and fuel consumption are reported.

**Parameters**

Engine Torque	Dynamics	Limits	Fuel Consumption	Speed Control
Fuel consumption model:	Fuel consumption by speed and torque			
Speed vector:	[ 1000 2e+3 3e+3 4e+3 5e+3 6e+3 ]			rpm
Torque vector:	[ 0 80 160 200 240 320 360 400 ]			N*m
Fuel consumption table:	9.4 10.8 14 16.2 18.6; 3.1 6 10.3 11.9 13.8 18.4 22 26.5 ]			g/s
Interpolation method:	Linear			

OK Cancel Help



## **Speed vector**

Enter vector of engine speeds used in lookup table parameterizations. Vector size must match **Torque vector size**. The default unit is rpm.

## **Torque vector**

Enter vector of engine torques used in the lookup table parameterizations. Vector size must match **Speed vector** size. The default unit is N\*m.

## **Fuel consumption table**

Enter matrix with fuel consumption rates corresponding to engine speed and torque vectors. The number of rows must equal the number of elements in the **Speed vector**. The number of columns must equal the number of elements in the **Torque vector**. The default unit is g/s.

## **Interpolation method**

Select the interpolation method used to calculate fuel consumption at intermediate speed-torque values. Methods are **linear**, **cubic**, and **spline**. Outside the data range, fuel consumption is held constant at the last value given in the lookup table.

# Generic Engine

## Brake specific fuel consumption by speed and torque

**Block Parameters: Generic Engine**

**Generic Engine**

Represents a system-level model of spark-ignition and diesel engines suitable for use at initial stages of modeling the basic parameters are available. An optional idle speed controller is included.

The throttle input signal  $T$  lies between zero and one and specifies the torque demanded from the engine as a fraction of the maximum possible torque. If the engine speed falls below the Stall speed, the engine torque is blended to zero. If the engine speed exceeds the Maximum speed, the simulation stops and issues an error message.

Connections  $F$  and  $B$  are mechanical rotational conserving ports associated with the engine crankshaft and engine flywheel respectively. Connections  $P$  and  $FC$  are physical signal output ports through which engine power and fuel consumption are reported.

**Parameters**

Engine Torque | Dynamics | Limits | **Fuel Consumption** | Speed Control

Fuel consumption model: Brake specific fuel consumption by speed and torque

Speed vector: [ 1000 2e+3 3e+3 4e+3 5e+3 6e+3 ] rpm

Torque vector: [ 0 80 160 200 240 320 360 400 ] N\*m

Brake specific fuel consumption table: [ 320 310 300 310 320; 450 430 370 340 330 330 350 380 ] g/hr/kW

Interpolation method: Linear

OK Cancel Help

## **Speed vector**

Enter vector of engine speeds used in lookup table parameterizations. Vector size must match **Torque vector size**. The default unit is rpm.

## **Torque vector**

Enter vector of engine torques used in the lookup table parameterizations. Vector size must match **Speed vector** size. The default unit is N\*m.

## **Brake specific fuel consumption table**

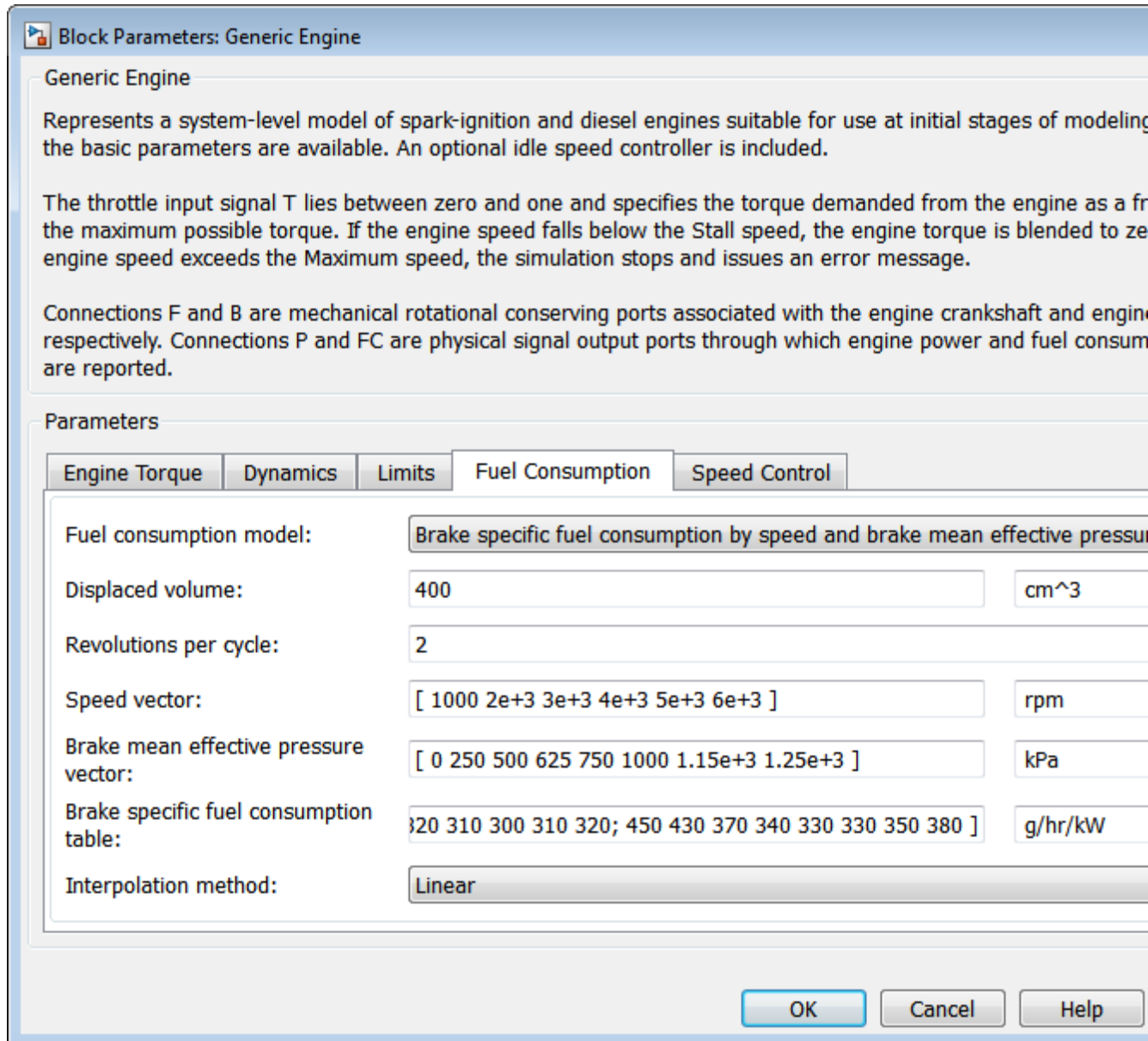
Enter matrix with brake specific fuel consumption (BSFC) rates corresponding to engine speed and torque vectors. BSFC is the ratio of the fuel consumption rate to the output power. The number of rows must equal the number of elements in the **Speed vector**. The number of columns must equal the number of elements in the **Torque vector**. The default unit is g/s.

## **Interpolation method**

Select the interpolation method used to calculate fuel consumption at intermediate speed-torque values. Methods are **linear**, **cubic**, and **spline**. Outside the data range, fuel consumption is held constant at the last value given in the lookup table.

# Generic Engine

## Brake specific fuel consumption by speed and brake mean effective pressure

The image shows a software dialog box titled "Block Parameters: Generic Engine". It contains a description of the engine model, a "Parameters" section with several tabs, and a "Fuel Consumption" tab that is currently selected. The "Fuel Consumption" tab shows various input fields for engine parameters, including fuel consumption model, displaced volume, revolutions per cycle, speed vector, brake mean effective pressure vector, and a brake specific fuel consumption table. The table contains numerical values for different engine speeds and pressures. At the bottom of the dialog are "OK", "Cancel", and "Help" buttons.

Block Parameters: Generic Engine

Generic Engine

Represents a system-level model of spark-ignition and diesel engines suitable for use at initial stages of modeling the basic parameters are available. An optional idle speed controller is included.

The throttle input signal T lies between zero and one and specifies the torque demanded from the engine as a fraction of the maximum possible torque. If the engine speed falls below the Stall speed, the engine torque is blended to zero. If the engine speed exceeds the Maximum speed, the simulation stops and issues an error message.

Connections F and B are mechanical rotational conserving ports associated with the engine crankshaft and engine flywheel respectively. Connections P and FC are physical signal output ports through which engine power and fuel consumption are reported.

Parameters

Engine Torque   Dynamics   Limits   **Fuel Consumption**   Speed Control

Fuel consumption model: Brake specific fuel consumption by speed and brake mean effective pressure

Displaced volume: 400 cm<sup>3</sup>

Revolutions per cycle: 2

Speed vector: [ 1000 2e+3 3e+3 4e+3 5e+3 6e+3 ] rpm

Brake mean effective pressure vector: [ 0 250 500 625 750 1000 1.15e+3 1.25e+3 ] kPa

Brake specific fuel consumption table: [ 320 310 300 310 320; 450 430 370 340 330 330 350 380 ] g/hr/kW

Interpolation method: Linear

OK   Cancel   Help

## Displaced volume

Enter the volume displaced by a piston stroke. The default value is 400. The default unit is cm<sup>3</sup>.

## Revolutions per cycle

Enter the number of crankshaft revolutions in one combustion cycle — e.g. 2 for a four-stroke engine, or 1 for a two-stroke engine. The default value is 2.

## Speed vector

Enter vector of speeds used in lookup table parameterizations. The default unit is rpm.

## Brake mean effective pressure vector

Enter vector of brake mean effective pressure (BMEP) values. The BMEP satisfies the expression:

$$BMEP = T \frac{2\pi n_c}{\sum V_d \cdot \dot{\tau}}$$

where:

- $T$  — Output torque
- $n_c$  — Number of cycles per revolution
- $V_d$  — Cylinder displaced volume

## Brake specific fuel consumption table

Enter matrix with brake specific fuel consumption (BSFC) rates corresponding to engine speed and brake mean effective pressure (BMEP) vectors. BSFC is the ratio of the fuel consumption rate to the output power. The number of rows must equal the number of elements in the **Speed vector**. The number of columns must equal the number of elements in the **Brake mean effective pressure vector**. The default unit is g/s.

## Interpolation method

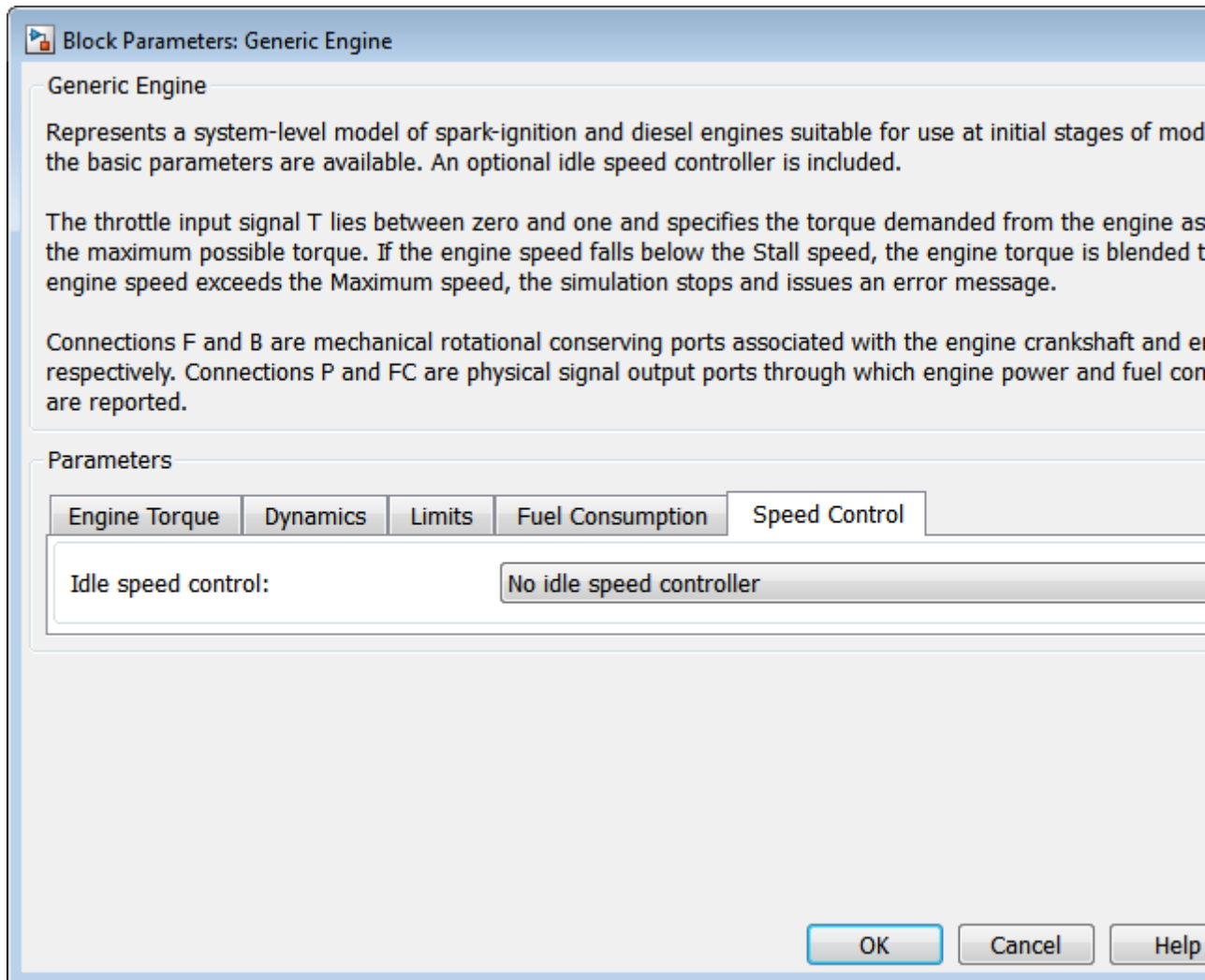
Select the interpolation method used to calculate fuel consumption at intermediate speed-torque values. Methods are linear, cubic,

# Generic Engine

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and spline. Outside the data range, fuel consumption is held constant at the last value given in the lookup table.

## Speed Control

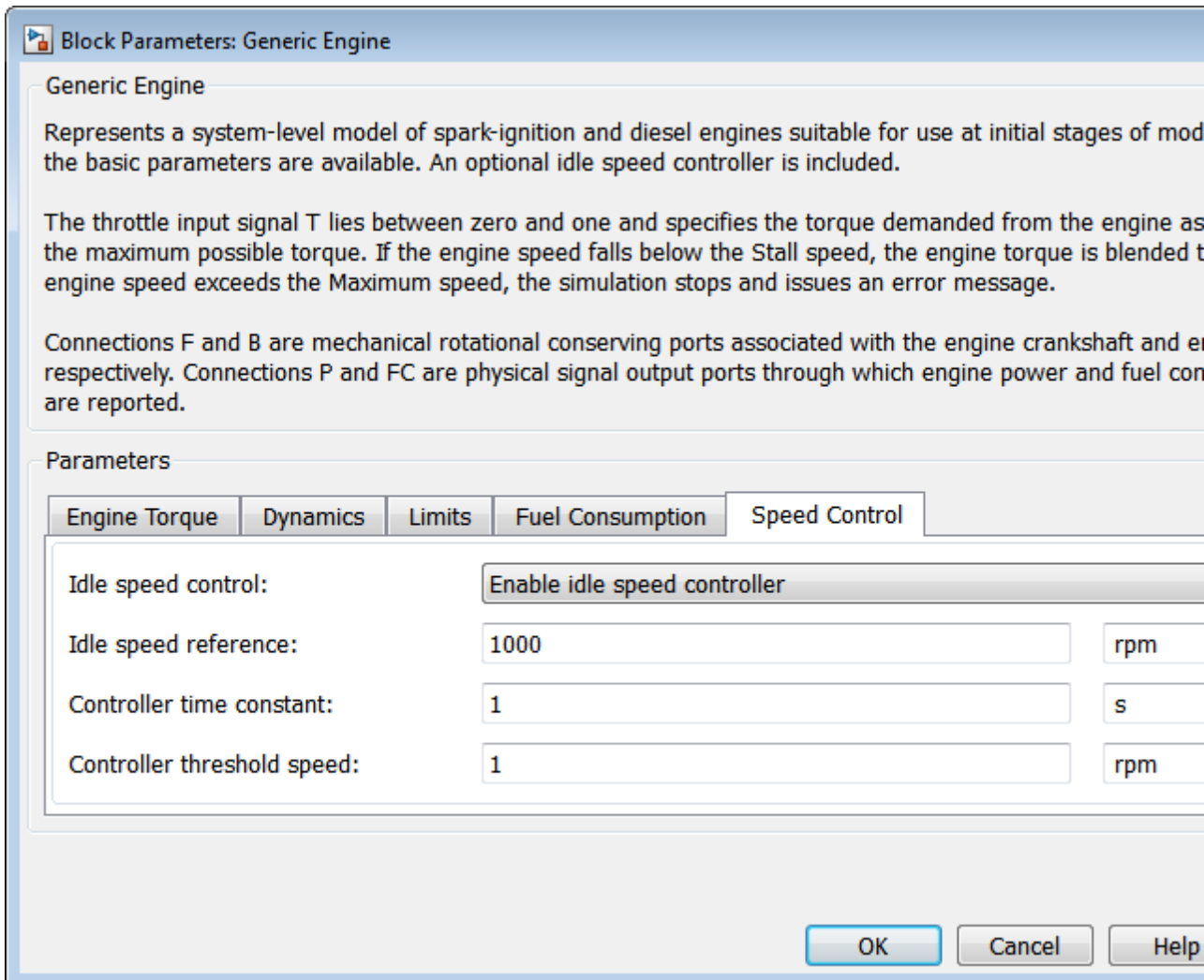


## Idle speed control

Select speed control model. Options include No idle speed controller and Enable idle speed controller.

- No idle speed controller — Omit idle speed controller. Throttle input is used directly.
- Enable idle speed controller — Include idle speed controller to prevent engine stalling. For more information, see “Idle Speed Controller Model” on page 1-167.





# Generic Engine

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## Idle speed reference

Enter the value of the speed reference below which speed increases, and above which speed decreases. The default value is 1000. The default unit is rpm.

## Controller time constant

Enter the value of the time constant associated with an increase or decrease of the controlled throttle. The constant value must be positive. The default value is 1. The default unit is s.

## Controller threshold speed

Parameter used to smooth the controlled throttle value when the engine's rotational speed crosses the idle speed reference. For more information, see "Idle Speed Controller Model" on page 1-167. Large values decrease controller responsiveness. Small values increase computational cost. This parameter must be positive. The default value is 1. The default unit is rpm.

## Generic Engine Model

By default, the Generic Engine model uses a programmed relationship between torque and speed, modulated by the throttle signal.

### Engine Speed, Throttle, Power, and Torque

The engine model is specified by an *engine power demand* function  $g(\Omega)$ . The function provides the maximum power available for a given engine speed  $\Omega$ . The block parameters (maximum power, speed at maximum power, and maximum speed) normalize this function to physical maximum torque and speed values.

The normalized throttle input signal  $T$  specifies the actual engine power. The power is delivered as a fraction of the maximum power possible in a steady state at a fixed engine speed. It modulates the actual power delivered,  $P$ , from the engine:  $P(\Omega, T) = T \cdot g(\Omega)$ . The engine torque is  $\tau = P/\Omega$ .

## Engine Power Demand

The engine power is nonzero when the speed is limited to the operating range,  $\Omega_{\min} \leq \Omega \leq \Omega_{\max}$ . The absolute maximum engine power  $P_{\max}$  defines  $\Omega_0$  such that  $P_{\max} = g(\Omega_0)$ . Define  $w \equiv \Omega/\Omega_0$  and  $g(\Omega) \equiv P_{\max} p(w)$ . Then  $p(1) = 1$  and  $dp(1)/dw = 0$ . The torque function is:

$$\tau = (P_{\max}/\Omega_0) \cdot [p(w)/w] .$$

You can derive forms for  $p(w)$  from engine data and models. Generic Engine uses a third-order polynomial form:

$$p(w) = p_1 \cdot w + p_2 \cdot w^2 - p_3 \cdot w^3$$

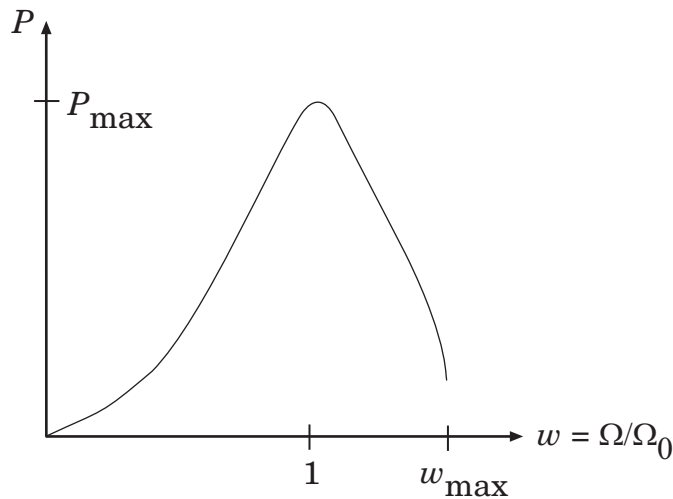
satisfying

$$p_1 + p_2 - p_3 = 1 , p_1 + 2p_2 - 3p_3 = 0 .$$

In typical engines, the  $p_i$  are positive. This polynomial has three zeros, one at  $w = 0$ , and a conjugate pair. One of the pair is positive and physical; the other is negative and unphysical:

$$w_{\pm} = \frac{1}{2} \left( -p_2 \pm \sqrt{p_2^2 + 4p_1p_3} \right)$$

# Generic Engine



## Typical Engine Power Demand Function

### Restrictions on Engine Speed and Power

- For the engine power polynomial, there are restrictions on the polynomial coefficients  $p_i$ , to achieve a valid power-speed curve. These restrictions are detailed below.
- If you use tabulated power or torque data, corresponding restrictions on  $P(\Omega)$  remain.

Set  $w = \Omega/\Omega_0$  and  $p = P(\Omega)/P_0$ , and  $w_{\min} = \Omega_{\min}/\Omega_0$  and  $w_{\max} = \Omega_{\max}/\Omega_0$ .  
Then:

- The engine speed is restricted to a positive range above the minimum speed and below the maximum speed:  $0 \leq w_{\min} \leq w \leq w_{\max}$ .
- The engine power at minimum speed must be nonnegative:  $p(w_{\min}) \geq 0$ . If you use the polynomial form, this condition is a restriction on the  $p_i$ :

$$p(w_{\min}) = p_1 \cdot w_{\min} + p_2 \cdot w_{\min}^2 - p_3 \cdot w_{\min}^3 \geq 0 .$$

- The engine power at maximum speed must be nonnegative:  $p(w_{\max}) \geq 0$ . If you use the polynomial form, this condition is a restriction on  $w_{\max}$ :  $w_{\max} \leq w_+$ .

## Engine Power Forms for Different Engine Types

For the default parametrization, Generic Engine provides two choices of internal combustion engine types, each with different engine power demand parameters.

Power Demand Coefficient	Engine Type:	
	Spark-Ignition	Diesel
$p_1$	1	0.6526
$p_2$	1	1.6948
$p_3$	1	1.3474

## Idle Speed Controller Model

The idle speed controller adjusts the throttle signal to increase engine rotation below a reference speed according to the following expressions:

$$\Pi = \max(\Pi_i, \Pi_c)$$

$$\frac{d(\Pi_c)}{dt} = \frac{0.5 \left( 1 - \tanh \left( 4 \frac{\omega - \omega_r}{\omega_r} \right) - \Pi_c \right)}{\tau}$$

where:

- $\Pi$  — Engine throttle
- $\Pi_i$  — Input throttle (port T)
- $\Pi_c$  — Controller throttle
- $\omega$  — Engine speed
- $\omega_e$  — Idle speed reference

# Generic Engine

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- $\omega_t$  — Controller speed threshold
- $\tau$  — Controller time constant

The controlled throttle increases with a first-order lag from zero to one when engine speed falls below the reference speed. When the engine speed rises above the reference speed, the controlled throttle decreases from one to zero. When the difference between engine velocity and reference speed is smaller than the controller speed threshold, the *tanh* function smooths the time derivative of the controlled throttle. The controlled throttle is limited to the range 0–1. The engine uses the larger of the input and controlled throttle values. If engine time lag is included, the controller changes the input *before* the lag is computed.

## Limitations

This block contains an engine time lag limitation.

### Engine Time Lag

Engines lag in their response to changing speed and throttle. The Generic Engine block optionally supports lag due to a changing throttle only. Time lag simulation increases model fidelity but reduces simulation performance.

## Ports

The block contains the following ports.

Port	Description
B	Rotational conserving port representing the engine block
F	Rotational Conserving port representing the engine crankshaft
T	Physical signal input port specifying the normalized engine throttle level
P	Physical signal output port reporting the instantaneous engine power
FC	Physical signal output port reporting the fuel consumption rate

Port T accepts a signal with values in the range 0–1. The signal specifies the engine torque as a fraction of the maximum torque possible in steady state at fixed engine speed. The signal saturates at zero and one: values below zero are interpreted as zero; values above one are interpreted as one.

## Examples

These SimDriveline example models make use of gasoline engines to power their drivelines:

- `sdl_dual_clutch`
- `sdl_hydraulic_clutch`
- `sdl_manual_transmission`
- `sdl_4wd_dynamics`
- `sdl_vehicle`

## See Also

“Complete Vehicle Model”

“Adjust Model Fidelity”

# Leadscrew

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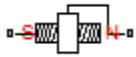
## Purpose

Leadscrew gear set of threaded rotating screw and translating nut, with adjustable thread and friction losses

## Library

Gears/Rotational-Translational

## Description



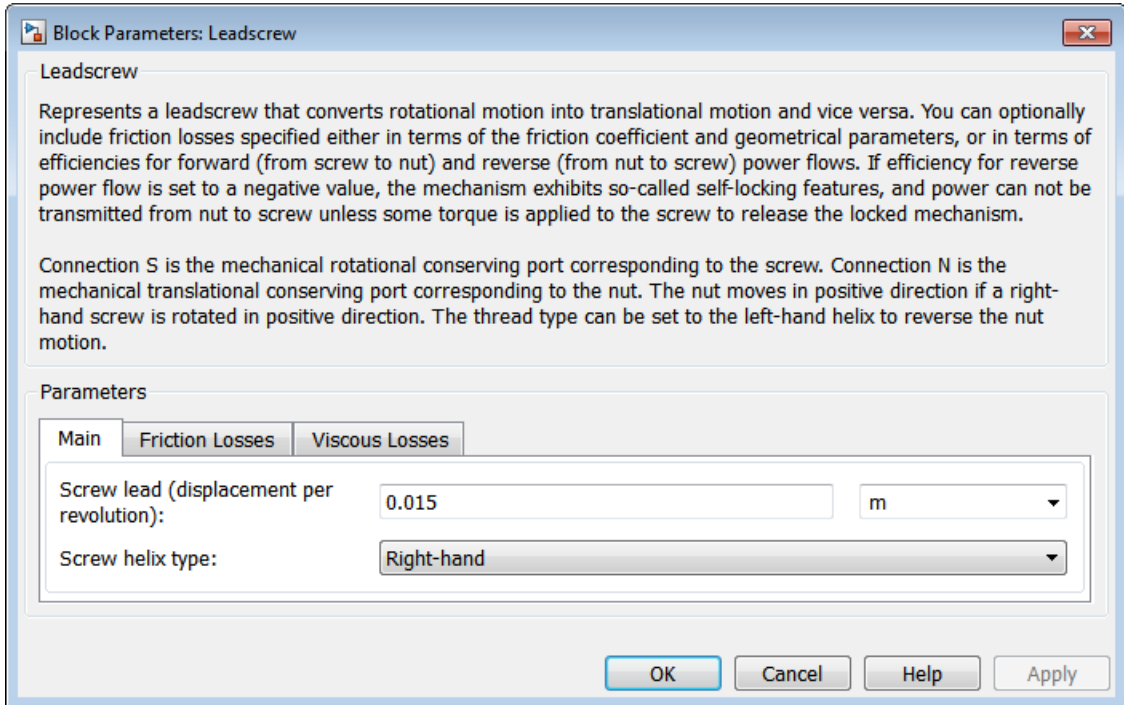
The Leadscrew block represents a threaded rotational-translational gear that constrains the two connected driveline axes, screw (S) and nut (N), to, respectively, rotate and translate together in a fixed ratio that you specify. You can choose whether the nut axis translates in a positive or negative direction, as the screw rotates in a positive right-handed direction. If the screw helix is right-handed,  $\omega_S$  and  $v_N$  have the same sign. If the screw helix is left-handed,  $\omega_S$  and  $v_N$  have opposite signs. For model details, see “Leadscrew Gear Model” on page 1-175.

## Ports

S is a rotational conserving port. N is a translational conserving port. They represent the screw and the nut, respectively.

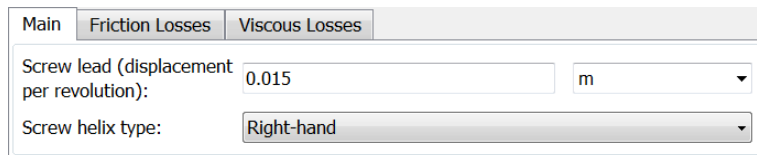


## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



## Screw lead

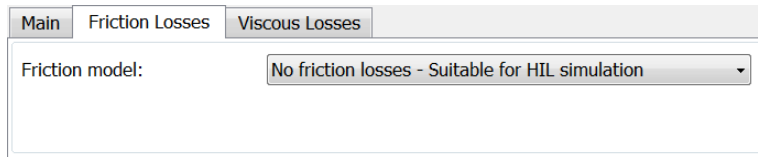
Translational displacement  $L$  of the nut per revolution of the screw. The default is 0.015.

From the drop-down list, choose units. The default is meters (m).

## Screw helix type

Choose the directional sense of screw rotation corresponding to positive nut translation. The default is Right-hand.

## Friction Losses



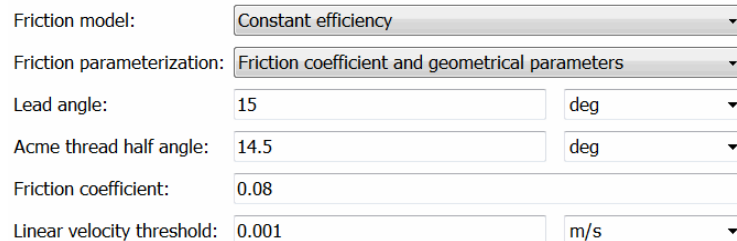
The screenshot shows a software interface with three tabs: 'Main', 'Friction Losses', and 'Viscous Losses'. The 'Friction Losses' tab is active. Below the tabs, there is a label 'Friction model:' followed by a dropdown menu. The dropdown menu is open, showing the selected option: 'No friction losses - Suitable for HIL simulation'.

## Friction model

Select how to implement friction losses from nonideal meshing of gear threads. The default is No friction losses.

- No friction losses Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque and force between screw and nut is reduced by friction. If you select this option, the panel expands.

## Constant Efficiency



The screenshot shows a software interface for the 'Constant Efficiency' friction model. It contains several input fields and dropdown menus:

- Friction model: Constant efficiency (dropdown)
- Friction parameterization: Friction coefficient and geometrical parameters (dropdown)
- Lead angle: 15 (text input) deg (dropdown)
- Acme thread half angle: 14.5 (text input) deg (dropdown)
- Friction coefficient: 0.08 (text input)
- Linear velocity threshold: 0.001 (text input) m/s (dropdown)

## Friction parametrization

Select how to characterize the friction between gear threads. The default is Friction coefficient and geometrical parameters.

- Friction coefficient and geometrical parameters — Friction is determined by contact friction between surfaces.

## Lead angle

Thread helix angle  $\lambda = \arctan[L/(\pi d)]$ .  $d$  is the mean thread diameter. Must be greater than zero. The default is 15.

From the drop-down list, choose units. The default is degrees (deg).

## Acme thread half angle

Half-angle of the acme thread  $\alpha$  in the normal plane. If a square thread is simulated,  $\alpha = 0$ . Must be greater than zero. The default is 14.5.

From the drop-down list, choose units. The default is degrees (deg).

## Friction coefficient

Dimensionless coefficient of kinetic friction in the thread. Must be greater than zero. The default is 0.08.

- Efficiencies — Friction is determined by constant efficiencies  $0 < \eta < 1$ . If you select this option, the panel changes from its default.

## Efficiencies

Friction model:	Constant efficiency	
Friction parameterization:	Efficiencies	
Screw-nut efficiency:	0.75	
Nut-screw efficiency:	0.45	
Linear velocity threshold:	0.001	m/s

### Screw-nut efficiency

Efficiency  $\eta_{SN}$  of the power transfer from screw to nut.

### Nut-screw efficiency

Efficiency  $\eta_{NS}$  of the power transfer from nut to screw.

### Linear velocity threshold

Absolute nut velocity threshold above which full efficiency loss is applied.

From the drop-down list, choose units. The default is meters/second (m/s).

## Viscous Losses

Main	Friction Losses	Viscous Losses	
Viscous friction coefficient:	0	N*m/(rad/s)	

### Viscous friction coefficient

Viscous friction coefficient  $\mu_S$  for the screw. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/ (rad/s)).

## Leadscrew Gear Model

### Ideal Gear Constraint and Gear Ratio

Leadscrew imposes one kinematic constraint on the two connected axes:

$$\omega_S L = 2\pi v_N .$$

The transmission ratio is  $R_{NS} = 2\pi/L$ .  $L$  is the screw lead, the translational displacement of the nut for one turn of the screw. In terms of this ratio, the kinematic constraint is:

$$\omega_S = R_{NS} v_N .$$

The two degrees of freedom are reduced to one independent degree of freedom. The forward-transfer gear pair convention is (1,2) = (S,N).

The torque-force transfer is:

$$R_{NS} \tau_S + F_N - F_{\text{loss}} = 0 ,$$

with  $F_{\text{loss}} = 0$  in the ideal case.

### Nonideal Gear Constraint and Losses

In the nonideal case,  $F_{\text{loss}} \neq 0$ . For general considerations on nonideal gear modeling, see “Model Gears with Losses”.

In a nonideal screw-nut pair (S,N), the angular velocity and geometric constraints are unchanged. But the transferred torque, force, and power are reduced by:

- Coulomb friction between thread surfaces on S and N, characterized by friction coefficient  $k$  or constant efficiencies ( $\eta_{SN}$ ,  $\eta_{NS}$ )
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficient  $\mu$

The loss force has the general form:

# Leadscrew

$$F_{\text{loss}} = F_{\text{Coul}} \tanh(4v_N/v_{\text{th}}) + \mu\omega_S/R_{\text{NS}}.$$

The hyperbolic tangent regularizes the sign change in the Coulomb friction force when the nut velocity changes sign.

Power Flow	Power Loss Condition	Output Driveshaft	Coulomb Friction Force $F_{\text{Coul}}$
Forward	$\omega_S \tau_S > F_N v_N$	Nut, $v_N$	$R_{\text{NS}}  \tau_S  (1 - \eta_{\text{SN}})$
Reverse	$\omega_S \tau_S < F_N v_N$	Screw, $\omega_S$	$ F_N  (1 - \eta_{\text{NS}})$

## Geometric Surface Contact Friction

In the contact friction case,  $\eta_{\text{SN}}$  and  $\eta_{\text{NS}}$  are determined by:

- The screw-nut threading geometry, specified by lead angle  $\lambda$  and acme thread half-angle  $\alpha$ .
- The surface contact friction coefficient  $k$ .

$$\eta_{\text{SN}} = (\cos\alpha - k \tan\alpha)/(\cos\alpha + k/\tan\lambda),$$

$$\eta_{\text{NS}} = (\cos\alpha - k/\tan\lambda)/(\cos\alpha + k \tan\alpha).$$

## Constant Efficiencies

In the constant efficiency case, you specify  $\eta_{\text{SN}}$  and  $\eta_{\text{NS}}$ , independently of geometric details.

## Self-Locking and Negative Efficiency

$\eta_{\text{NS}}$  has two distinct regimes, depending on lead angle  $\lambda$ , separated by the *self-locking point* at which  $\eta_{\text{NS}} = 0$  and  $\cos\alpha = k/\tan\lambda$ .

- In the *overhauling regime*,  $\eta_{\text{NS}} > 0$ . The force acting on the nut can rotate the screw.
- In the *self-locking regime*,  $\eta_{\text{NS}} < 0$ . An external torque must be applied to the screw to release an otherwise locked mechanism. The more negative is  $\eta_{\text{NS}}$ , the larger the torque must be to release the mechanism.

$\eta_{SN}$  is conventionally positive.

## Meshing Efficiency

The efficiencies  $\eta$  of meshing between screw and nut are fully active only if the absolute value of the nut velocity is greater than the velocity tolerance.

If the velocity is less than the tolerance, the actual efficiency is automatically regularized to unity at zero velocity.

## Viscous Friction Force

The viscous friction coefficient  $\mu$  controls the viscous friction torque experienced by the screw from lubricated, nonideal gear threads. The viscous friction torque on a screw driveline axis is  $-\mu_S \omega_S$ .  $\omega_S$  is the angular velocity of the screw with respect to its mounting.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## Example

The `sdl_stepping_mechanism` example model uses the Leadscrew gear to translate a load in one direction (ratcheting).

## See Also

Rack & Pinion

“Represent and Transfer Driveline Motion and Torque”

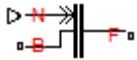
# Loaded-Contact Rotational Friction

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**Purpose** Loaded-contact friction between two rotating bodies

**Library** Brakes & Detents/Rotational

## Description



The Loaded-Contact Rotational Friction block simulates friction between two rotating bodies loaded with a normal force.

The block is implemented as a structural component based on the Fundamental Friction Clutch block. From the locked state, the clutch unlocks if the friction force exceeds the static friction, as defined by the static coefficient of friction and current normal force. For details on how the locking and unlocking are modeled, see the Fundamental Friction Clutch block reference page.

Friction torque is transmitted for normal forces larger than the **Threshold force** parameter.

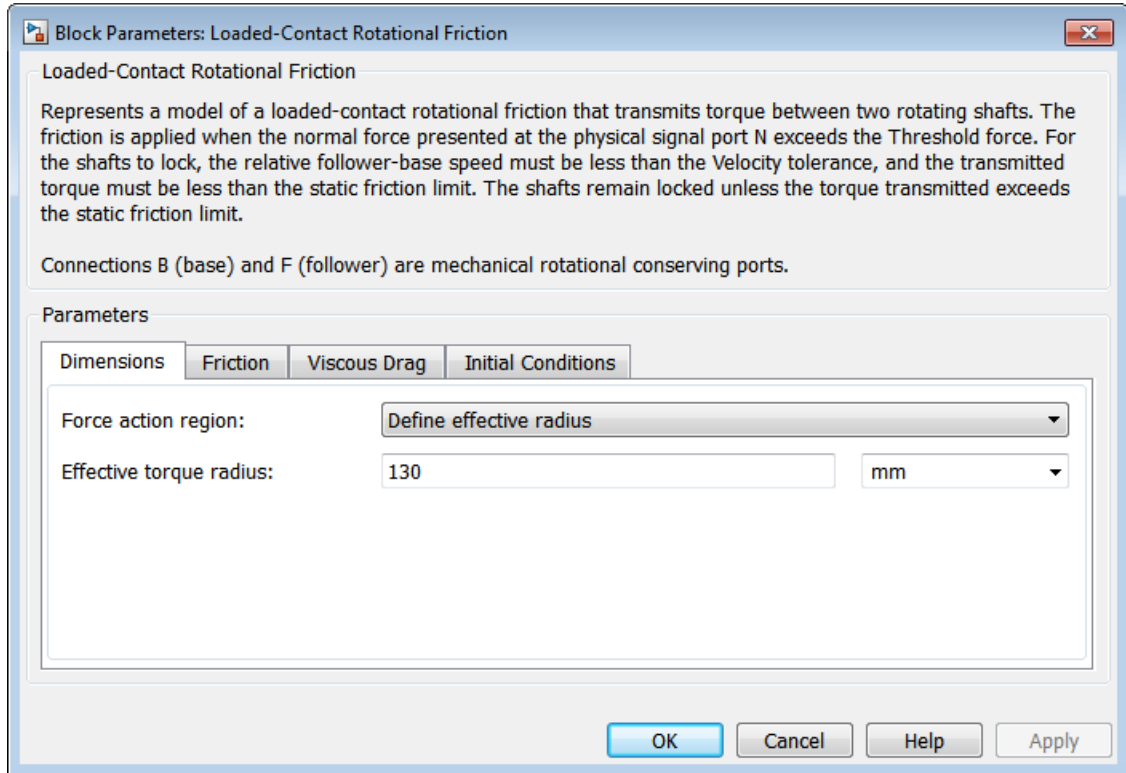
### Ports

B and F are rotational conserving ports associated with the driving and driven shafts, respectively. N is the physical signal terminal through which you import the normal force.



# Loaded-Contact Rotational Friction

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with four tabs.

# Loaded-Contact Rotational Friction

## Dimensions

The screenshot shows a software interface with a tabbed menu at the top containing 'Dimensions', 'Friction', 'Viscous Drag', and 'Initial Conditions'. The 'Dimensions' tab is active. Below the tabs, there are two rows of controls. The first row is labeled 'Force action region:' and has a dropdown menu with 'Define effective radius' selected. The second row is labeled 'Effective torque radius:' and has a text input field containing the number '130' and a unit dropdown menu with 'mm' selected.

### Force action region

Select how to parameterize the loaded-contact friction. The default is **Define effective radius**.

- **Define effective radius** — Provide a value for the friction effective radius.

### Effective torque radius

Effective radius  $r_{\text{eff}}$ . Must be greater than zero. The default is 130.

From the drop-down list, choose units. The default is millimeters (mm).

- **Define annular region** — Define the friction effective radius in terms of the inside and outside diameters of the friction disk. If you select this option, the panel changes from its default.

# Loaded-Contact Rotational Friction

## Define annular region

Dimensions	Friction	Viscous Drag	Initial Conditions
Force action region:	Define annular region		
Friction disk outside diameter:	150	mm	
Friction disk inside diameter:	100	mm	

## Friction disk outside diameter

Outside diameter  $r_o$  of the friction disks. Must be greater than zero. The default is 150.

From the drop-down list, choose units. The default is millimeters (mm).

## Friction disk inside diameter

Inside diameter  $r_i$  of the friction disks. Must be greater than zero, but less than the friction disk outside diameter. The default is 100.

From the drop-down list, choose units. The default is millimeters (mm).

## Friction

Dimensions	Friction	Viscous Drag	Initial Conditions
Friction model:	Fixed kinetic friction coefficient		
Kinetic friction coefficient:	0.3		
Static friction coefficient:	0.35		
Velocity tolerance:	0.001	rad/s	
Threshold force:	1	N	

# Loaded-Contact Rotational Friction

## Friction model

Select how to specify the kinetic friction coefficient. The default is Fixed kinetic friction coefficient.

- Fixed kinetic friction coefficient — Provide a fixed value for the kinetic friction coefficient.

## Kinetic friction coefficient

The kinetic, or Coulomb, friction coefficient. Must be greater than zero. The default is 0.3.

- Table lookup kinetic friction coefficient — Define the kinetic friction coefficient by one-dimensional table lookup based on the relative angular velocity between disks. If you select this option, the panel changes from its default.

## Table lookup kinetic friction coefficient

Dimensions	Friction	Viscous Drag	Initial Conditions
Friction model:	Table lookup kinetic friction coefficient		
Kinetic friction coefficient relative velocity vector:	[ 0 0.1 0.2 0.3 0.4 0.5 1 2 5 10 100 1000 ]		rad/s
Kinetic friction coefficient vector:	[ 0.29 0.28 0.26 0.22 0.21 0.2 0.2 0.19 0.19 0.19 0.19 ]		
Friction coefficient interpolation method:	Linear		
Friction coefficient extrapolation method:	From last 2 points		
Static friction coefficient:	0.35		
Velocity tolerance:	0.001	rad/s	
Threshold force:	1	N	

## Kinetic friction coefficient relative velocity vector

Specify the vector of input values for the relative velocity. The input values vector must be strictly increasing. The values can be nonuniformly spaced. The

minimum number of values depends on the interpolation method: you must provide at least two values for linear interpolation, at least three values for cubic or spline interpolation. The default values are [0 0.1 0.2 0.3 0.4 0.5 1 2 5 10 100 1000].

From the drop-down list, choose units. The default is radians/second (rad/s).

## **Kinetic friction coefficient vector**

Specify the vector of output values for kinetic friction coefficient. The kinetic friction coefficient vector must be of the same size as the relative velocity vector. All the values must be positive. The default values are [0.29 0.28 0.26 0.22 0.21 0.2 0.2 0.19 0.19 0.19 0.19 0.19].

## **Interpolation method**

Select one of the following interpolation methods for approximating the output value when the input value is between two consecutive grid points:

- **Linear** — Uses a linear interpolation function.
- **Cubic** — Uses the Piecewise Cubic Hermite Interpolation Polynomial (PCHIP).
- **Spline** — Uses the cubic spline interpolation algorithm.

For more information on interpolation algorithms, see the PS Lookup Table (1D) block reference page.

## **Extrapolation method**

Select one of the following extrapolation methods for determining the output value when the input value is outside the range specified in the argument list:

- **From last 2 points** — Extrapolates using the linear method (regardless of the interpolation method

# Loaded-Contact Rotational Friction

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specified), based on the last two output values at the appropriate end of the range. That is, the block uses the first and second specified output values if the input value is below the specified range, and the two last specified output values if the input value is above the specified range.

- **From last point** — Uses the last specified output value at the appropriate end of the range. That is, the block uses the last specified output value for all input values greater than the last specified input argument, and the first specified output value for all input values less than the first specified input argument.

For more information on extrapolation algorithms, see the PS Lookup Table (1D) block reference page.

## Static friction coefficient

The static, or peak, value of the friction coefficient. Must be greater than the kinetic friction coefficient. The default is 0.35.

## Velocity tolerance

Relative velocity below which the two surfaces can lock. For the surfaces to lock, the torque across the B and F rotational ports must be less than the product of the effective radius, the static friction coefficient, and the applied normal force. The default is 0.001.

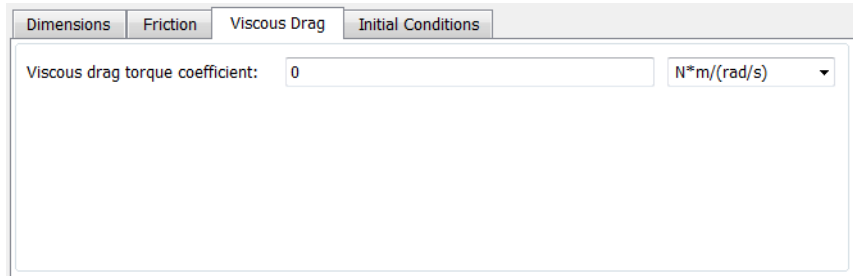
From the drop-down list, choose units. The default is radians/second (rad/s).

## Threshold force

The normal force applied to the physical signal port N must exceed the **Threshold force** parameter value to be applied to the contact. Forces below the **Threshold force** are not applied, and there is consequently no transmitted frictional torque. The default is 1.

From the drop-down list, choose units. The default is newton (N).

## Viscous Drag



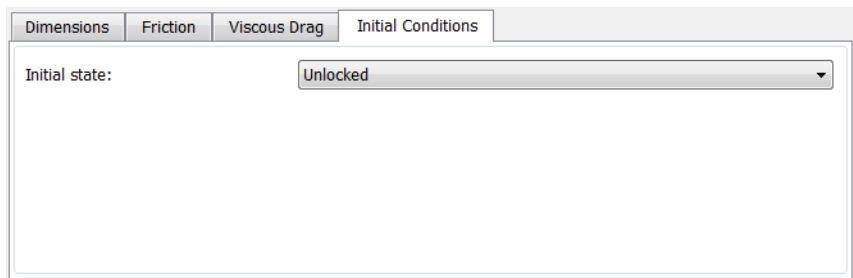
The screenshot shows a software interface with four tabs: "Dimensions", "Friction", "Viscous Drag", and "Initial Conditions". The "Viscous Drag" tab is selected. It contains a text input field labeled "Viscous drag torque coefficient:" with the value "0" entered. To the right of the input field is a drop-down menu showing the units "N\*m/(rad/s)".

### Viscous drag torque coefficient

Viscous drag coefficient  $\mu_{\text{visc}}$  for computing the drag torque. The coefficient depends on the type of operating fluid, fluid temperature, and the maximum distance between the disks. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/ (rad/s)).

## Initial Conditions



The screenshot shows a software interface with four tabs: "Dimensions", "Friction", "Viscous Drag", and "Initial Conditions". The "Initial Conditions" tab is selected. It contains a drop-down menu labeled "Initial state:" with the value "Unlocked" selected.

### Initial state

Select the initial condition applied to the Fundamental Friction Clutch state inside the block:

# Loaded-Contact Rotational Friction

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- **Locked** — Rotational ports B and F are initially locked together. This is the default.
- **Unlocked** — Rotational ports B and F are initially sliding relative to each other.

## Loaded-Contact Rotational Friction Model

The block simulates friction between two rotating bodies loaded with a normal force. When the two rotating bodies are not locked, the friction torque is determined with the following equations:

$$\tau_{\text{fr}} = N \cdot \mu \cdot r_{\text{eff}} \cdot \text{sign}(\omega) + \tau_{\text{visc}} ,$$

$$r_{\text{eff}} = \frac{2}{3} \cdot \frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} ,$$

$$\tau_{\text{visc}} = \mu_{\text{visc}} \cdot \omega ,$$

where:

$\tau_{\text{fr}}$	Friction torque
$N$	Normal force
$\mu$	Friction coefficient
$r_{\text{eff}}$	Effective radius
$r_o$	Disk outside radius
$r_i$	Disk inside radius
$\omega$	Relative angular velocity
$\tau_{\text{visc}}$	Viscous drag torque
$\mu_{\text{visc}}$	Viscous drag torque coefficient

## Limitations

The model does not account for inertia. Add inertia terms externally to the B and F ports as required.



The model computes the torque assuming a uniform distribution of the normal force.

## **See Also**

Fundamental Friction Clutch , Loaded-Contact Translational Friction  
“Control Gear Couplings with Clutches”.

# Loaded-Contact Translational Friction

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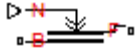
## Purpose

Loaded-contact friction between two sliding bodies

## Library

Brakes & Detents/Translational

## Description



The Loaded-Contact Translational Friction block simulates friction between two sliding bodies loaded with a normal force.

The block is implemented as a structural component based on the Fundamental Friction Clutch block. From the locked state, the clutch unlocks if the friction force exceeds the static friction, as defined by the static coefficient of friction and current normal force. For details on how the locking and unlocking are modeled, see the Fundamental Friction Clutch block reference page.

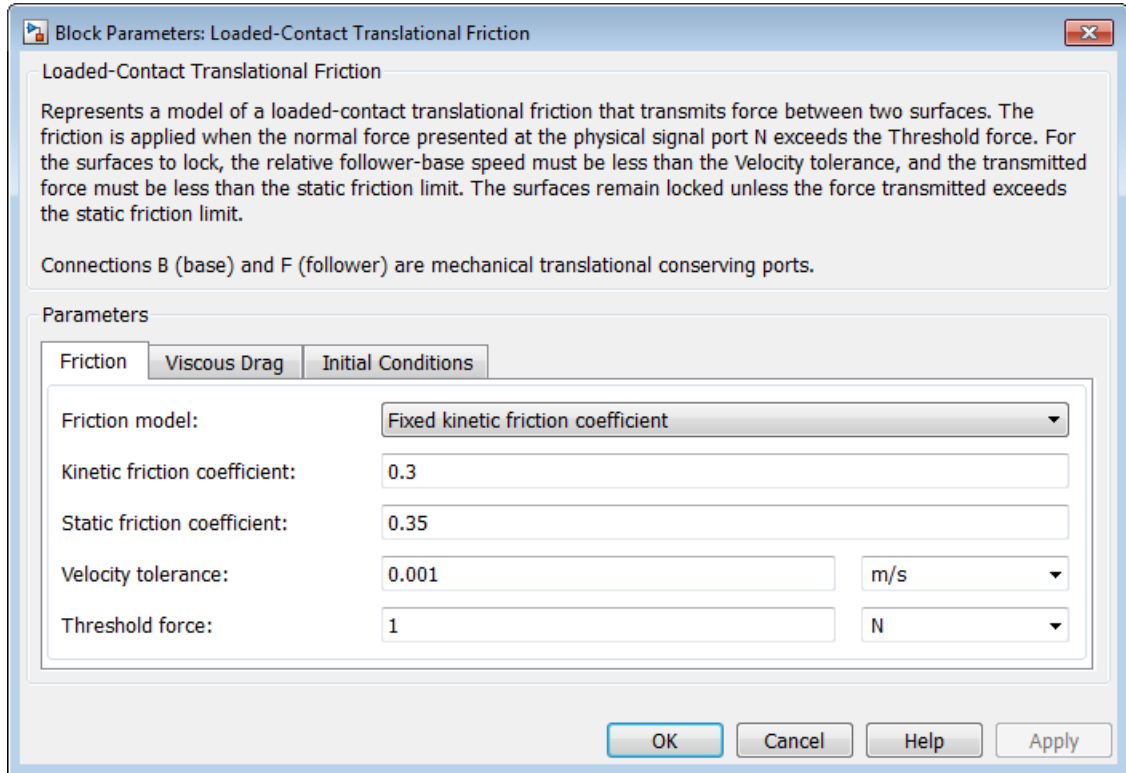
Friction force is transmitted for normal forces larger than the **Threshold force** parameter.

## Ports

B and F are translational conserving ports associated with the base and follower, respectively. N is the physical signal terminal through which you import the normal force.

# Loaded-Contact Translational Friction

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

# Loaded-Contact Translational Friction

## Friction

Friction	Viscous Drag	Initial Conditions
Friction model:	Fixed kinetic friction coefficient	
Kinetic friction coefficient:	0.3	
Static friction coefficient:	0.35	
Velocity tolerance:	0.001	m/s
Threshold force:	1	N

### Friction model

Select how to specify the kinetic friction coefficient. The default is Fixed kinetic friction coefficient.

- Fixed kinetic friction coefficient — Provide a fixed value for the kinetic friction coefficient.

### Kinetic friction coefficient

The kinetic, or Coulomb, friction coefficient. Must be greater than zero. The default is 0.3.

- Table lookup kinetic friction coefficient — Define the kinetic friction coefficient by one-dimensional table lookup based on the relative angular velocity between disks. If you select this option, the panel changes from its default.

# Loaded-Contact Translational Friction

## Table lookup kinetic friction coefficient

Friction	Viscous Drag	Initial Conditions
Friction model:	Table lookup kinetic friction coefficient	
Kinetic friction coefficient relative velocity vector:	[ 0 0.1 0.2 0.3 0.4 0.5 1 2 5 ]	m/s
Kinetic friction coefficient vector:	[ 0.29 0.28 0.26 0.22 0.21 0.2 0.2 0.19 0.19 ]	
Friction coefficient interpolation method:	Linear	
Friction coefficient extrapolation method:	From last 2 points	
Static friction coefficient:	0.35	
Velocity tolerance:	0.001	m/s
Threshold force:	1	N

### Kinetic friction coefficient relative velocity vector

Specify the vector of input values for the relative velocity. The input values vector must be strictly increasing. The values can be nonuniformly spaced. The minimum number of values depends on the interpolation method: you must provide at least two values for linear interpolation, at least three values for cubic or spline interpolation. The default values are [0 0.1 0.2 0.3 0.4 0.5 1 2 5].

From the drop-down list, choose units. The default is meters/second (m/s).

### Kinetic friction coefficient vector

Specify the vector of output values for kinetic friction coefficient. The kinetic friction coefficient vector must be of the same size as the relative velocity vector. All the values must be positive. The default values are [0.29 0.28 0.26 0.22 0.21 0.2 0.2 0.19 0.19 0.19 0.19].

# Loaded-Contact Translational Friction

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## Interpolation method

Select one of the following interpolation methods for approximating the output value when the input value is between two consecutive grid points:

- **Linear** — Uses a linear interpolation function.
- **Cubic** — Uses the Piecewise Cubic Hermite Interpolation Polynomial (PCHIP).
- **Spline** — Uses the cubic spline interpolation algorithm.

For more information on interpolation algorithms, see the PS Lookup Table (1D) block reference page.

## Extrapolation method

Select one of the following extrapolation methods for determining the output value when the input value is outside the range specified in the argument list:

- **From last 2 points** — Extrapolates using the linear method (regardless of the interpolation method specified), based on the last two output values at the appropriate end of the range. That is, the block uses the first and second specified output values if the input value is below the specified range, and the two last specified output values if the input value is above the specified range.
- **From last point** — Uses the last specified output value at the appropriate end of the range. That is, the block uses the last specified output value for all input values greater than the last specified input argument, and the first specified output value for all input values less than the first specified input argument.

For more information on extrapolation algorithms, see the PS Lookup Table (1D) block reference page.

# Loaded-Contact Translational Friction

## Static friction coefficient

The static, or peak, value of the friction coefficient. Must be greater than the kinetic friction coefficient. The default is 0.35.

## Velocity tolerance

Relative velocity below which the two surfaces can lock. For the surfaces to lock, the force across the B and F translational ports must be less than the product of the static friction coefficient and the applied normal force. The default is 0.001.

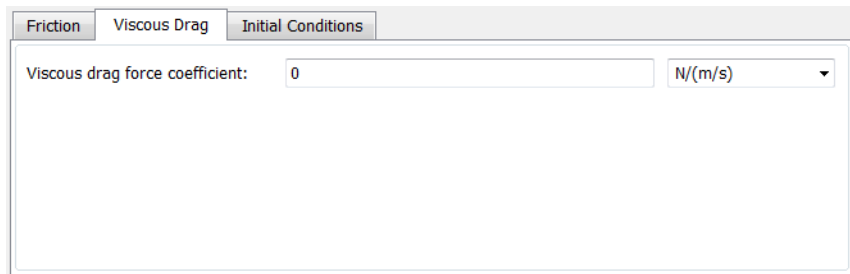
From the drop-down list, choose units. The default is meters/second (m/s).

## Threshold force

The normal force applied to the physical signal port N must exceed the **Threshold force** parameter value to be applied to the contact. Forces below the **Threshold force** are not applied, and there is consequently no transmitted frictional torque. The default is 1.

From the drop-down list, choose units. The default is newton (N).

## Viscous Drag



The image shows a software interface for configuring Viscous Drag. It features three tabs: 'Friction', 'Viscous Drag', and 'Initial Conditions'. The 'Viscous Drag' tab is active, displaying a text input field labeled 'Viscous drag force coefficient' with the value '0' and a dropdown menu for units set to 'N/(m/s)'.

## Viscous drag force coefficient

Viscous drag coefficient  $\mu_{\text{visc}}$  for computing the drag force. The coefficient depends on the type of operating fluid, fluid temperature, and the maximum distance between the sliding bodies. The default is 0.

# Loaded-Contact Translational Friction

From the drop-down list, choose units. The default is newton/(meters/second) (N/ (m/s)).

## Initial Conditions



### Initial state

Select the initial condition applied to the Fundamental Friction Clutch state inside the block:

- **Locked** — Translational ports B and F are initially locked together. This is the default.
- **Unlocked** — Translational ports B and F are initially sliding relative to each other.

## Loaded-Contact Translational Friction Model

The block simulates friction between two sliding bodies loaded with a normal force. When the two sliding bodies are not locked, the friction force is determined with the following equations:

$$F_{\text{fr}} = N \cdot \mu \cdot \text{sign}(v_{\text{rel}}) + F_{\text{visc}},$$

$$F_{\text{visc}} = \mu_{\text{visc}} \cdot v_{\text{rel}},$$

where:

$F_{\text{fr}}$  Friction force

$N$  Normal force



# Loaded-Contact Translational Friction

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$\mu$	Friction coefficient
$v_{\text{rel}}$	Relative velocity
$F_{\text{visc}}$	Viscous drag force
$\mu_{\text{visc}}$	Viscous drag force coefficient

## Limitations

The model does not account for body mass. Add mass terms externally to the B and F ports as required.

The model computes the friction force assuming a uniform distribution of the normal force.

## See Also

Fundamental Friction Clutch , Loaded-Contact Rotational Friction  
“Control Gear Couplings with Clutches”

# Nonlinear Rotational Damper

**Purpose** Rotational damper based on polynomial or lookup-table parameterizations

**Library** Couplings & Drives/Springs & Dampers

**Description** The block represents a nonlinear rotational damper. Polynomial and lookup-table parameterizations define the nonlinear relationship between damping torque and relative angular velocity. The damping torque can be symmetric or asymmetric about the zero velocity point. The block applies equal and opposite damping torques on the two rotational conserving ports.



The symmetric polynomial parameterization defines the damping torque for both positive and negative relative velocities according to the expression:

$$T = b_1\omega + \text{sign}(\omega) b_2\omega^2 + b_3\omega^3 + \text{sign}(\omega) b_4\omega^4 + b_5\omega^5$$

where:

- $T$  — Damping torque
- $b_1, b_2, \dots, b_5$  — Damping coefficients
- $\omega$  — Relative angular velocity between ports R and C,  $\omega = \omega_R - \omega_C$
- $\omega_R$  — Absolute angular velocity associated with port R
- $\omega_C$  — Absolute angular velocity associated with port C

Using an odd polynomial ( $b_2, b_4 = 0$ ), eliminates the sign function from the polynomial expression, avoiding zero-crossings that slow down simulation.

The two-sided polynomial parameterization defines the damping torque for both positive and negative relative velocities according to the expression:

$$T = \begin{cases} b_{1p}\omega + b_{2p}\omega^2 + b_{3p}\omega^3 + b_{4p}\omega^4 + b_{5p}\omega^5, & \omega \neq 0 \\ b_{1n}\omega + b_{2n}\omega^2 + b_{3n}\omega^3 + b_{4n}\omega^4 + b_{5n}\omega^5, & \omega < 0 \end{cases}$$

where:

- $b_{1p}, b_{2p}, \dots, b_{5p}$  — Damping coefficients for positive relative velocities
- $b_{1n}, b_{2n}, \dots, b_{5n}$  — Damping coefficients for negative relative velocities

Both polynomial parameterizations use a fifth-order polynomial expression. To use a lower-order polynomial, set the unneeded higher-order coefficients to zero. To use a higher-order polynomial, fit to a lower order polynomial or use the lookup table parameterization.

The lookup table parameterization defines damping torque based on a set of torque and angular velocity vectors. If not specified, the block automatically adds a data point at the origin (zero angular velocity and zero torque).

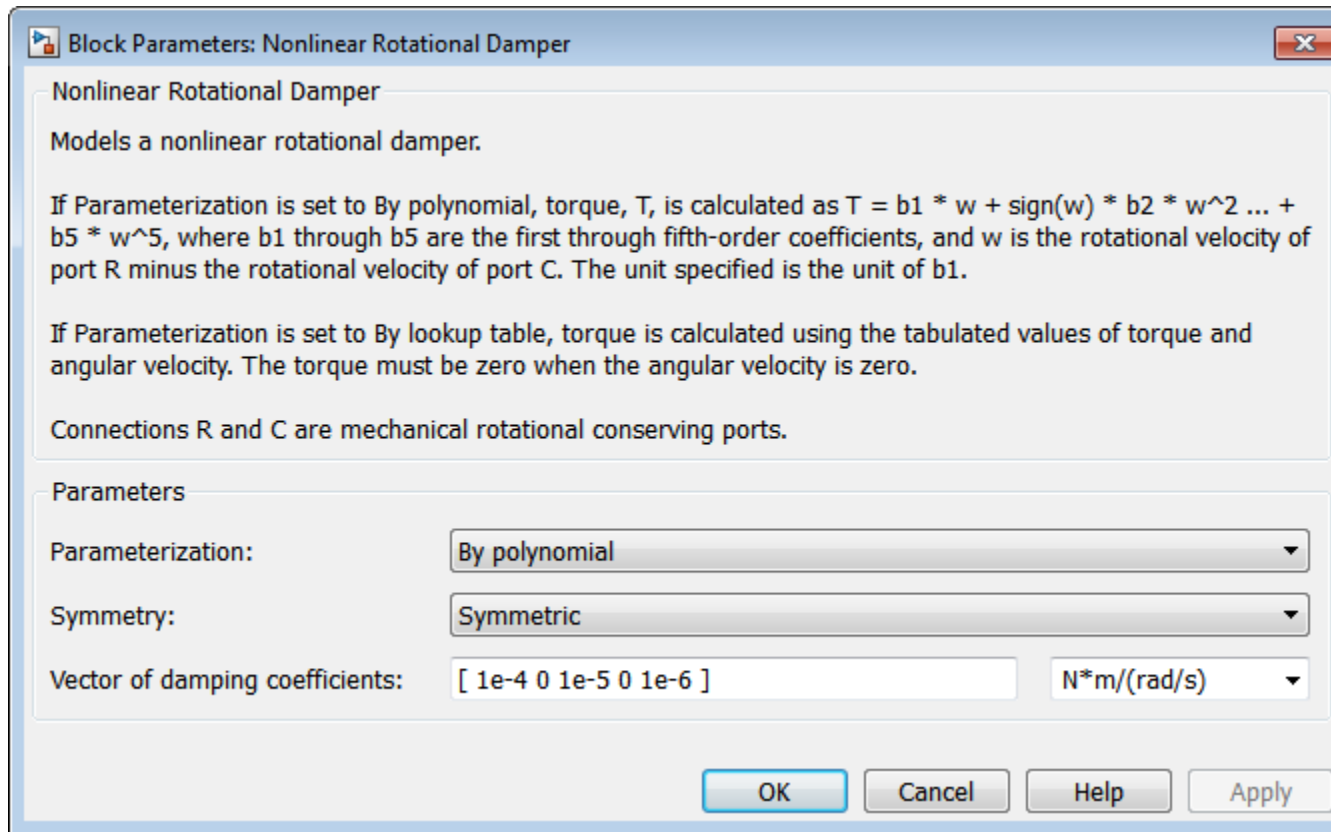
## Assumptions and Limitations

- Damping is of the viscous type. It depends only on velocity.

## Dialog Box and Parameters

The dialog box contains one active area, **Parameters**.

# Nonlinear Rotational Damper



## Parameterization

Select damping parameterization. Options are By polynomial and By lookup table.

## By polynomial

Specify coefficients of polynomial damping function.

## Symmetry

Choose between symmetric and two-sided polynomial parameterizations.

- **Symmetric** — Specify a single set of polynomial coefficients governing damping for both positive and negative relative velocities.

## Vector of damping coefficients

Enter five-element vector with polynomial damping coefficients. Physical units are for the first coefficient.

The default vector is [1e-4 0 1e-5 0 1e-6]. The default unit is N\*m/(rad/s).

- **Two-sided** — Specify two sets of polynomial coefficients governing damping, one for positive relative velocities, the other for negative relative velocities.

## Vector of positive rotation damping coefficients

Enter five-element vector with polynomial damping coefficients for positive relative velocities. Physical units are for the first coefficient.

The default vector is [1e-4 0 1e-5 0 1e-6]. The default unit is N\*m/(rad/s).

## Vector of negative rotation damping coefficients

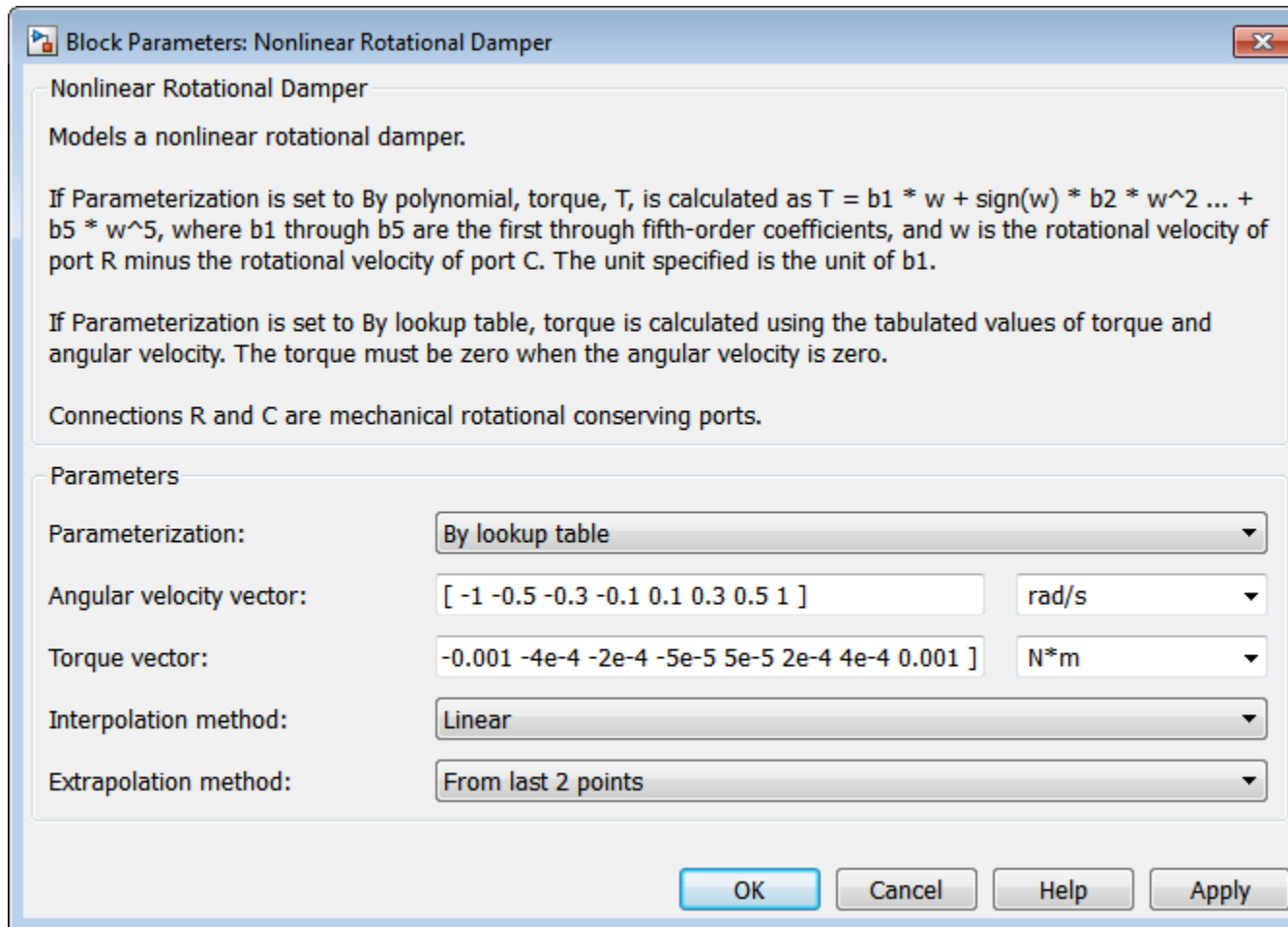
Enter five-element vector with polynomial damping coefficients for negative relative velocities. Physical units are for the first coefficient.

The default vector is [0.001 -1e-5 1e-4 0 0].

## By lookup table

Specify velocity and torque data for damping lookup table.

# Nonlinear Rotational Damper



## Angular velocity vector

Enter vector with relative velocity values. The vector requires a minimum number of elements, based on the selected interpolation method — two for Linear, and three for Cubic or Spline. The number of elements must match the torque vector.

The default vector is [-1 -0.5 -0.3 0.1 0.1 0.3 0.5 1]. The default unit is rad/s.

## **Torque vector**

Enter vector with damping torque values corresponding to velocity vector. The vector requires a minimum number of elements, based on the selected interpolation method — two for **Linear**, three for **Cubic** or **Spline**. The number of elements must match the velocity vector.

The default vector is [-0.001 -4e-4 -2e-4 -5e-5 5e-5 2e-4 4e-4 0.001]. The unit is N\*m.

## **Interpolation Method**

Select method used to find velocity–torque values between specified data points.

- **Linear** — Interpolate between two points using a first-order polynomial function.
- **Cubic** — Interpolate between two points using a third-order polynomial function.
- **Spline** — Interpolate between two points using a piecewise polynomial function.

## **Extrapolation Method**

Select method used to calculate values outside the lookup-table data range.

- **From last two points** — Extrapolate by extending the straight line between the last two data points.
- **From last point** — Extrapolate by extending the horizontal straight line passing through the last data point.

# Nonlinear Rotational Damper

---

## Ports

Port	Description
C	Rotational conserving port
R	Rotational conserving port

## See Also

Variable Rotational Damper, Nonlinear Translational damper, Variable Translational Damper



## Purpose

Torsional spring based on polynomial or lookup table parameterizations

## Library

Couplings & Drives/Springs & Dampers

## Description



The block represents a torsional spring with nonlinear torque-displacement curve. The spring torque magnitude is a general function of displacement. It need not satisfy Hooke's law. Polynomial and lookup-table parameterizations provide two ways to specify the torque-displacement relationship. The spring torque can be symmetric or asymmetric with respect to zero deformation.

The symmetric polynomial parameterization defines spring torque according to the expression:

$$T = k_1\theta + \text{sign}(\theta) k_2\theta^2 + k_3\theta^3 + \text{sign}(\theta) k_4\theta^4 + k_5\theta^5$$

where:

- $T$  — Spring force
- $k_1, k_2, \dots, k_5$  — Spring coefficients
- $\theta$  — Relative displacement between ports R and C,  $\theta = \theta_{init} + \theta_R - \theta_C$
- $\theta_{init}$  — Initial spring deformation
- $\theta_R$  — Absolute angular position of port R
- $\theta_C$  — Absolute angular position of port C

At simulation start ( $t=0$ ),  $\theta_R$  and  $\theta_C$  are zero, making  $\theta$  equal to  $\theta_{init}$ .

Specifying an odd polynomial ( $b_2, b_4 = 0$ ) eliminates the sign function from the polynomial expression. This avoids zero-crossings that slow down simulation.

The two-sided polynomial parameterization defines spring torque according to the expression:

# Nonlinear Rotational Spring

---

$$T = \begin{cases} k_{1t}\theta + k_{2t}\theta^2 + k_{3t}\theta^3 + k_{4t}\theta^4 + k_{5t}\theta^5, & \theta \geq 0 \\ k_{1c}\theta - k_{2c}\theta^2 + k_{3c}\theta^3 - k_{4c}\theta^4 + k_{5c}\theta^5, & \theta < 0 \end{cases}$$

where:

- $k_{1t}, k_{2t}, \dots, k_{5t}$  — Spring tension coefficients
- $k_{1c}, k_{2c}, \dots, k_{5c}$  — Spring compression coefficients

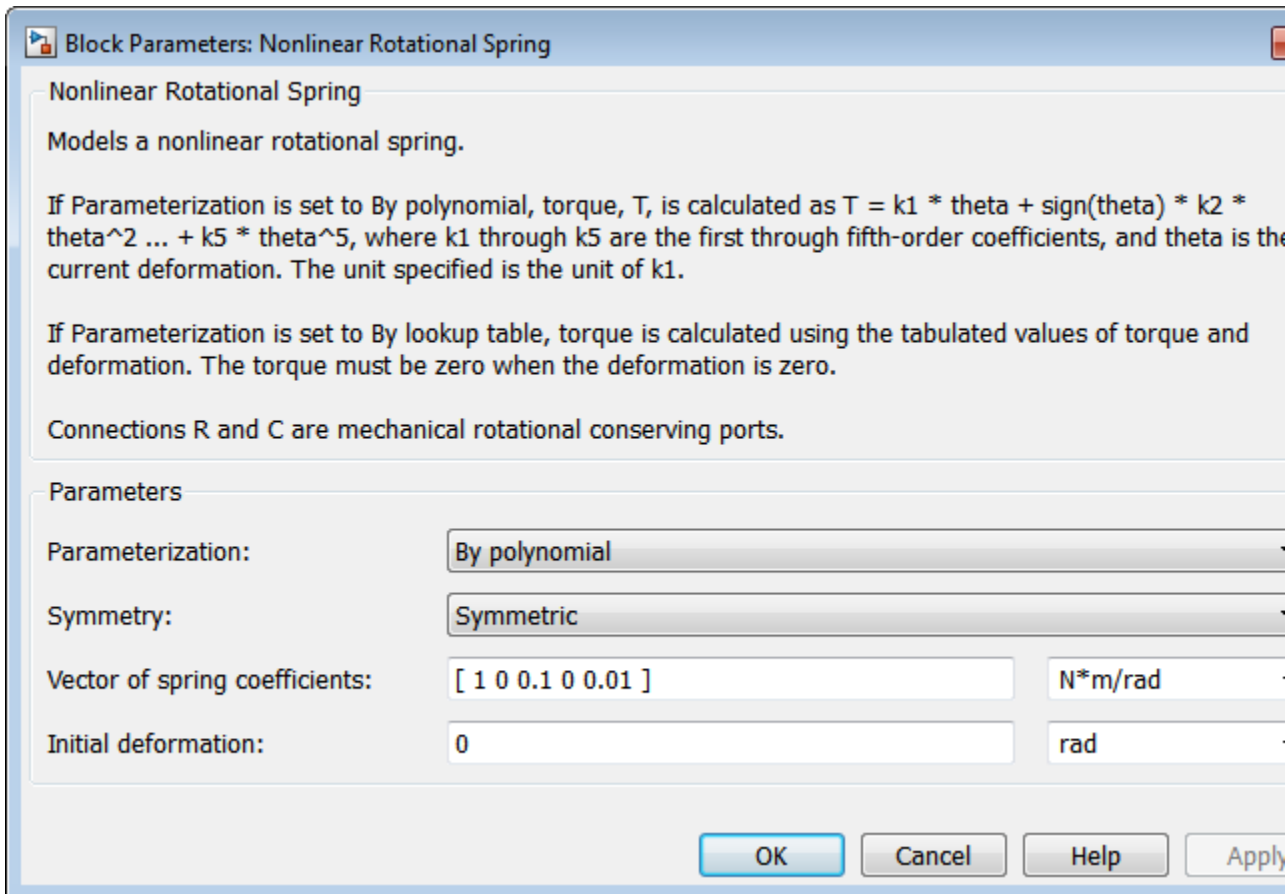
Both polynomial parameterizations use a fifth-order polynomial expression. To use a lower-order polynomial, set the unneeded higher-order coefficients to zero. To use a higher-order polynomial, fit to a lower order polynomial or use the lookup table parameterization.

The lookup table parameterization defines spring torque based on a set of torque and angular velocity vectors. If not specified, the block automatically adds a data point at the origin (zero angular velocity and zero torque).

## Dialog Box and Parameters

The dialog box includes active area, **Parameters**.

# Nonlinear Rotational Spring



## Parameterization

Select spring parameterization. Options are By polynomial and By lookup table.

### By Polynomial

Specify nonlinear spring function in terms of polynomial coefficients.

# Nonlinear Rotational Spring

---

## Symmetry

Choose between symmetric and two-sided polynomial parameterizations.

- **Symmetric** — Specify one set of polynomial coefficients governing spring torque in both tension and compression.

## Vector of spring coefficients

Enter five-element vector with polynomial spring coefficients. The highest non-zero order must be positive. Physical units are for the first coefficient.

The default vector is [1 0 0.1 0 0.01]. The default unit is N\*m/rad.

- **Two-sided** — Specify two sets of polynomial coefficients governing spring torque, one for positive deformation, the other for negative deformation.

## Vector of spring tension coefficients

Enter a five-element vector containing the coefficients of the polynomial spring *tension* function. The highest order non-zero coefficient must be positive. The specified physical unit is the unit of the first polynomial coefficient.

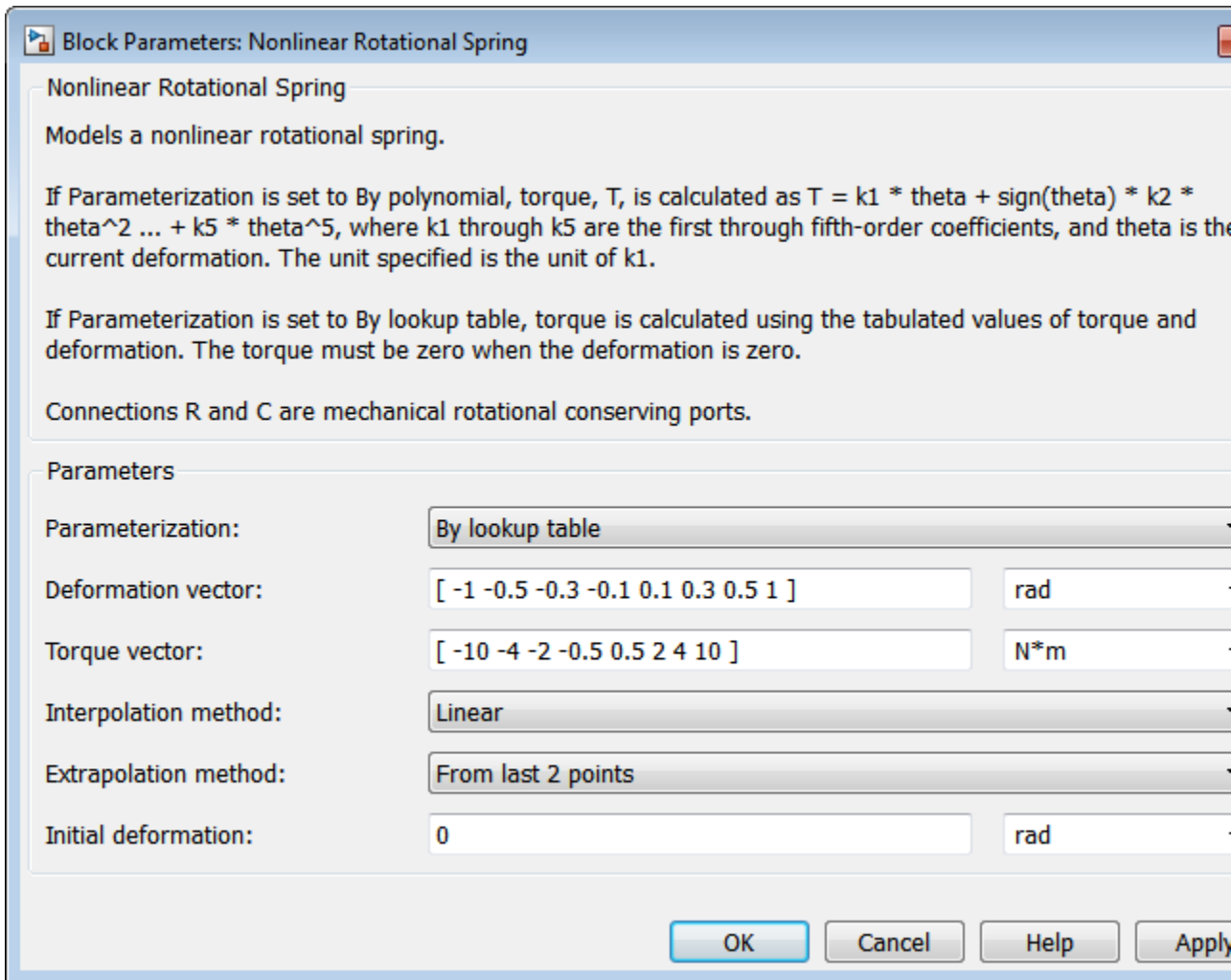
## Vector of spring compression coefficients

Enter a five-element vector containing the coefficients of the polynomial spring *compression* function. The highest order non-zero coefficient must be positive. The specified physical unit is the unit of the first polynomial coefficient.

## By lookup table

Specify nonlinear spring function in terms of deformation and torque vectors.

# Nonlinear Rotational Spring



## Deformation vector

Enter a vector containing the deformation values used in the lookup table. The vector must contain a minimum number of elements based on the interpolation method: two for

# Nonlinear Rotational Spring

---

Linear, and three for Cubic or Spline. The vector must also contain the same number of elements as the torque vector.

## **Torque vector**

Enter a vector containing the torque values that correspond to the deformation vector values. The vector must contain a minimum number of elements based on the interpolation method: two for Linear, and three for Cubic or Spline. The vector must also contain the same number of elements as the deformation vector. If not included in the vectors, the block automatically adds a point at the (0,0) deformation–torque coordinate.

## **Interpolation method**

Select method used to calculate deformation-torque values between lookup-table data points.

- **Linear** — Interpolate using first-order polynomial. Two points or more required.
- **Cubic** — Interpolate using third-order polynomial. Three points or more are required.
- **Spline** — Interpolate using piecewise polynomial. Three points or more are required.

## **Extrapolation method**

Select method used to calculate deformation-torque values outside the lookup-table data range.

- **From last two points** — Extrapolate by extending the straight line between the last two data points.
- **From last point** — Extrapolate by extending the horizontal straight line passing through the last data point.

# Nonlinear Rotational Spring

---

## Ports

Port	Description
C	Rotational conserving port
R	Rotational conserving port

## See Also

Variable Rotational Spring, Nonlinear Translational Spring, Variable Translational Spring

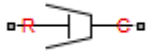
# Nonlinear Translational Damper

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**Purpose** Translational damper based on polynomial or lookup table parameterizations

**Library** Couplings & Drives/Springs & Dampers

**Description** The block represents a nonlinear translational damper. Polynomial and lookup table parameterizations define the nonlinear relationship between damping force and relative linear velocity. The damping force can be symmetric or asymmetric about the zero velocity point. The block applies equal and opposite damping forces on the two translational conserving ports.



The symmetric polynomial parameterization defines the damping force for both positive and negative relative velocities according to the expression:

$$F = b_1 v + \text{sign}(v) (b_2 v^2 + b_3 v^3 + \text{sign}(v) (b_4 v^4 + b_5 v^5))$$

where:

- $F$  — Damping force
- $b_1, b_2, \dots, b_5$  — Damping coefficients
- $v$  — Relative linear velocity between ports R and C,  $v = v_R - v_C$
- $v_R$  — Absolute linear velocity associated with port R
- $v_C$  — Absolute linear velocity associated with port C

Using an odd polynomial ( $b_2, b_4 = 0$ ), eliminates the sign function from the polynomial expression, avoiding zero-crossings that slow down simulation.

The two-sided polynomial parameterization defines the damping force for both positive and negative relative velocities according to the expression:



# Nonlinear Translational Damper

$$F = \begin{cases} b_{1e}v + b_{2e}v^2 + b_{3e}v^3 + b_{4e}v^4 + b_{5e}v^5, & v \neq 0 \\ b_{1c}v - b_{2c}v^2 + b_{3c}v^3 - b_{4c}v^4 + b_{5c}v^5, & v < 0 \end{cases}$$

where:

- $b_{1e}, b_{2e}, \dots, b_{5e}$  — Damping coefficients for positive relative velocities
- $b_{1c}, b_{2c}, \dots, b_{5c}$  — Damping coefficients for negative relative velocities

Positive relative velocities correspond to damper extension (ports R and C moving away from each other). Negative relative velocities correspond to damper contraction (ports R and C moving towards each other).

Both polynomial parameterizations use a fifth-order polynomial expression. To use a lower-order polynomial, set the unneeded higher-order coefficients to zero. For polynomials of order greater than five, fit to a polynomial of order smaller than or equal to five, or use the lookup table parameterization.

The lookup table parameterization defines damping force based on a set of velocity and force vectors. If not included in the vectors, the block automatically adds a data point at the origin (zero velocity and zero force).

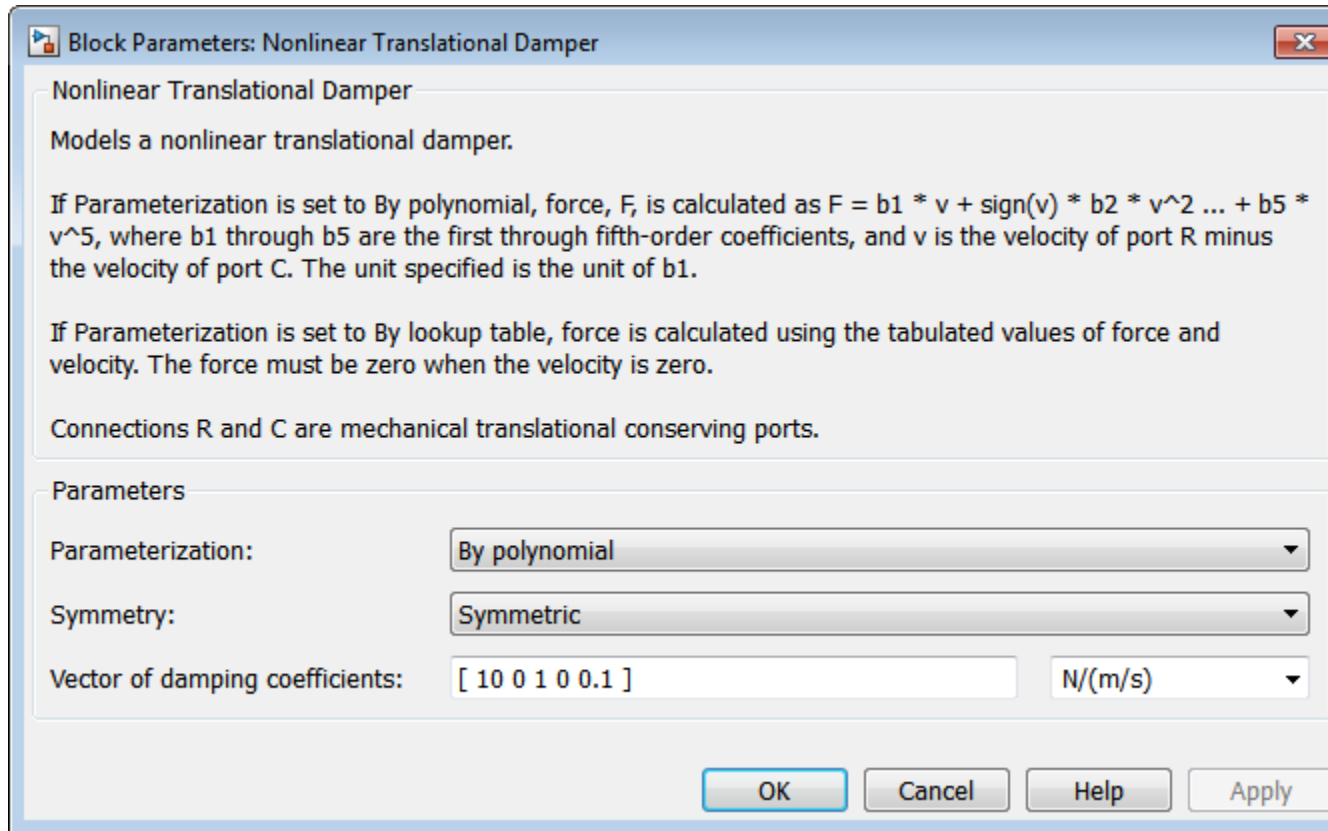
## Assumptions and Limitations

- The block assumes viscous damping. The damping force depends only on velocity.

## Dialog Box and Parameters

The dialog box contains one active area, **Parameters**.

# Nonlinear Translational Damper



## Parameterization

Select damping parameterization. Options are By polynomial and By lookup table.

## By polynomial

Specify coefficients of polynomial damping function.

## Symmetry

Choose between symmetric and two-sided polynomial parameterizations.

# Nonlinear Translational Damper

---

- **Symmetric** — Specify a single set of polynomial coefficients governing damping for both positive and negative relative velocities.

## Vector of damping coefficients

Enter five-element vector with polynomial damping coefficients. Physical units are for the first coefficient.

The default vector is [10 0 1 0 0.1]. The default unit is N/(m/s).

- **Two-sided** — Specify two sets of polynomial coefficients governing damping, one for positive relative velocities, the other for negative relative velocities.

## Vector of extension damping coefficients

Enter five-element vector with polynomial damping coefficients for positive relative velocities. Physical units are for the first coefficient.

The default vector is [10 0 1 0 0.1]. The default unit is N/(m/s).

## Vector of contraction damping coefficients

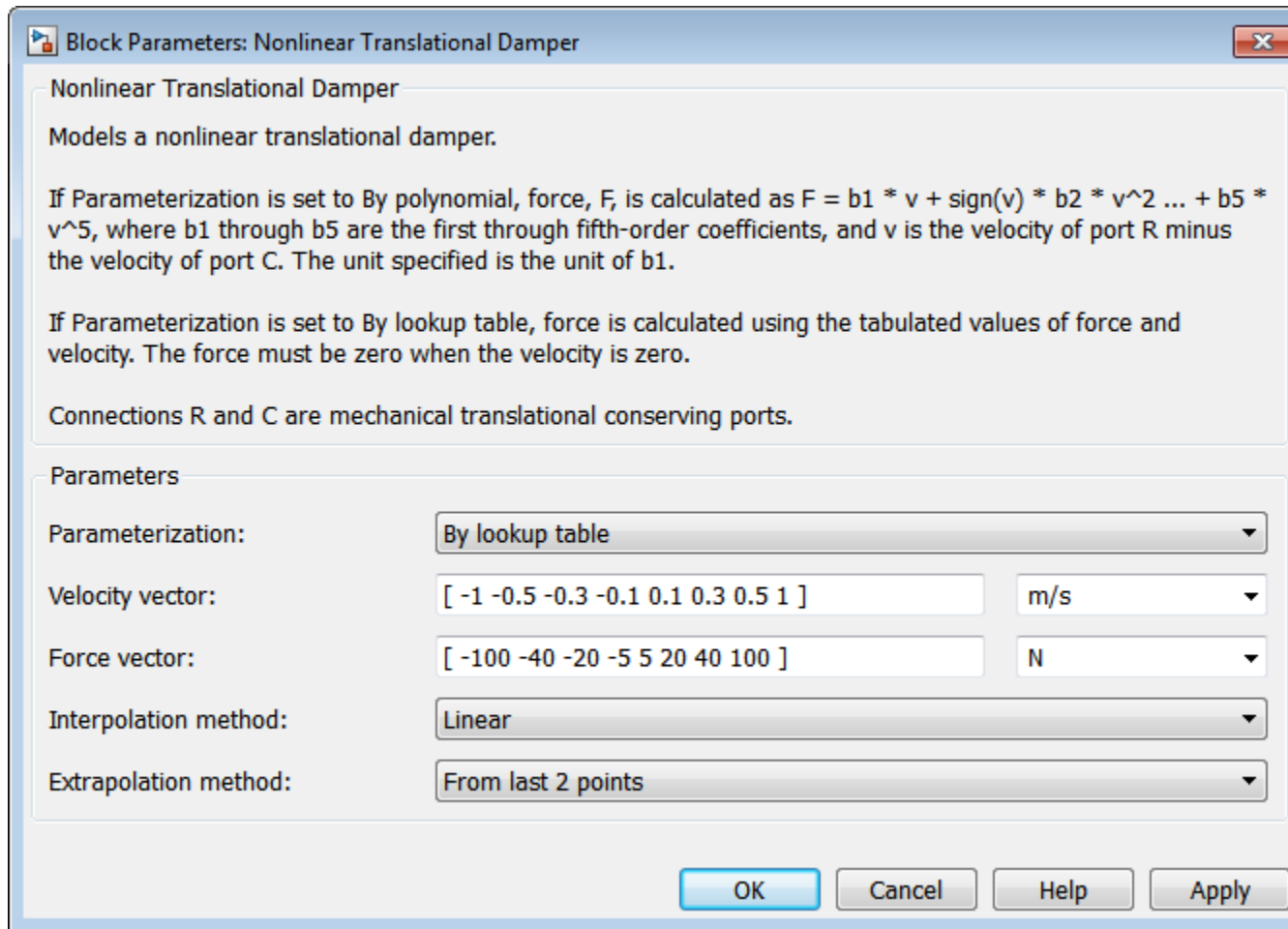
Enter five-element vector with polynomial damping coefficients for negative relative velocities. Physical units are for the first coefficient.

The default vector is [10 0 1 0 0.1]. The default unit is N/(m/s).

## By lookup table

Specify velocity and force data for damping lookup table.

# Nonlinear Translational Damper



## Velocity vector

Enter vector with relative velocity values. The vector requires a minimum number of elements, based on the selected interpolation method — two for Linear, and three for Cubic or Spline. The number of elements must match the force vector.

The default vector is [-1 -0.5 -0.3 -0.1 0.1 0.3 0.5 1]. The default unit is m/s.

## Force vector

Enter vector with damping force values corresponding to velocity vector. The vector requires a minimum number of elements, based on the selected interpolation method — two for **Linear**, three for **Cubic** or **Spline**. The number of elements must match the velocity vector.

The default vector is [-100 -40 -20 -5 5 20 40 100].  
The default unit is N.

## Interpolation Method

Select method used to find intermediate velocity–force values between lookup-table data points.

- **Linear** — Interpolate between two points using a first-order polynomial function.
- **Cubic** — Interpolate between two points using a third-order polynomial function.
- **Spline** — Interpolate between two points using a piecewise polynomial function.

## Extrapolation Method

Select method used to calculate values outside the lookup-table data range.

- **From last two points** — Extrapolate by extending the straight line connecting the last two lookup-table data points.
- **From last point** — Extrapolate by extending the horizontal straight line passing through the last lookup-table data point.

# Nonlinear Translational Damper

---

## Ports

Port	Description
C	Rotational conserving port
R	Rotational conserving port

## See Also

Variable Translational Damper, Nonlinear Rotational Damper, Variable Rotational Damper

## Purpose

Translational spring based on polynomial or lookup table parameterizations

## Library

Couplings & Drives/Springs & Dampers

## Description



The block represents a translational spring with nonlinear force-displacement curve. The spring force magnitude is a general function of displacement. It need not satisfy Hooke's law. Polynomial and lookup-table parameterizations provide two ways to specify the force-displacement relationship. The spring force can be symmetric or asymmetric with respect to zero deformation.

The symmetric polynomial parameterization defines spring force according to the expression:

$$F = k_1 x + \text{sign}(x) k_2 x^2 + k_3 x^3 + \text{sign}(x) k_4 x^4 + k_5 x^5$$

where:

- $F$  — Spring force
- $k_1, k_2, \dots, k_5$  — Spring coefficients
- $x$  — Relative displacement between ports R and C,  $x = x_{init} + x_R - x_C$
- $x_{init}$  — Initial relative displacement
- $x_R$  — Absolute displacement associated with port R
- $x_C$  — Absolute displacement associated with port C

At simulation start ( $t=0$ ),  $x_R$  and  $x_C$  are zero, making  $x$  equal to  $x_{init}$ .

Specifying an odd polynomial ( $b_2, b_4 = 0$ ) eliminates the sign function from the polynomial expression. This avoids zero-crossings that slow down simulation.

The asymmetric polynomial parameterization defines spring force according to the expression:

# Nonlinear Translational Spring

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$$F = \begin{cases} k_{1t}x + k_{2t}x^2 + k_{3t}x^3 + k_{4t}x^4 + k_{5t}x^5, & x \geq 0 \\ k_{1c}x + k_{2c}x^2 + k_{3c}x^3 + k_{4c}x^4 + k_{5c}x^5, & x < 0 \end{cases}$$

where:

- $k_1, k_2, \dots, k_5$  — Spring tension coefficients
- $k_1, k_2, \dots, k_5$  — Spring compression coefficients

Both polynomial parameterizations use a fifth-order polynomial expression. To use a lower-order polynomial, set the unneeded higher-order coefficients to zero. To use a higher-order polynomial, fit to a lower order polynomial or use the lookup table parameterization.

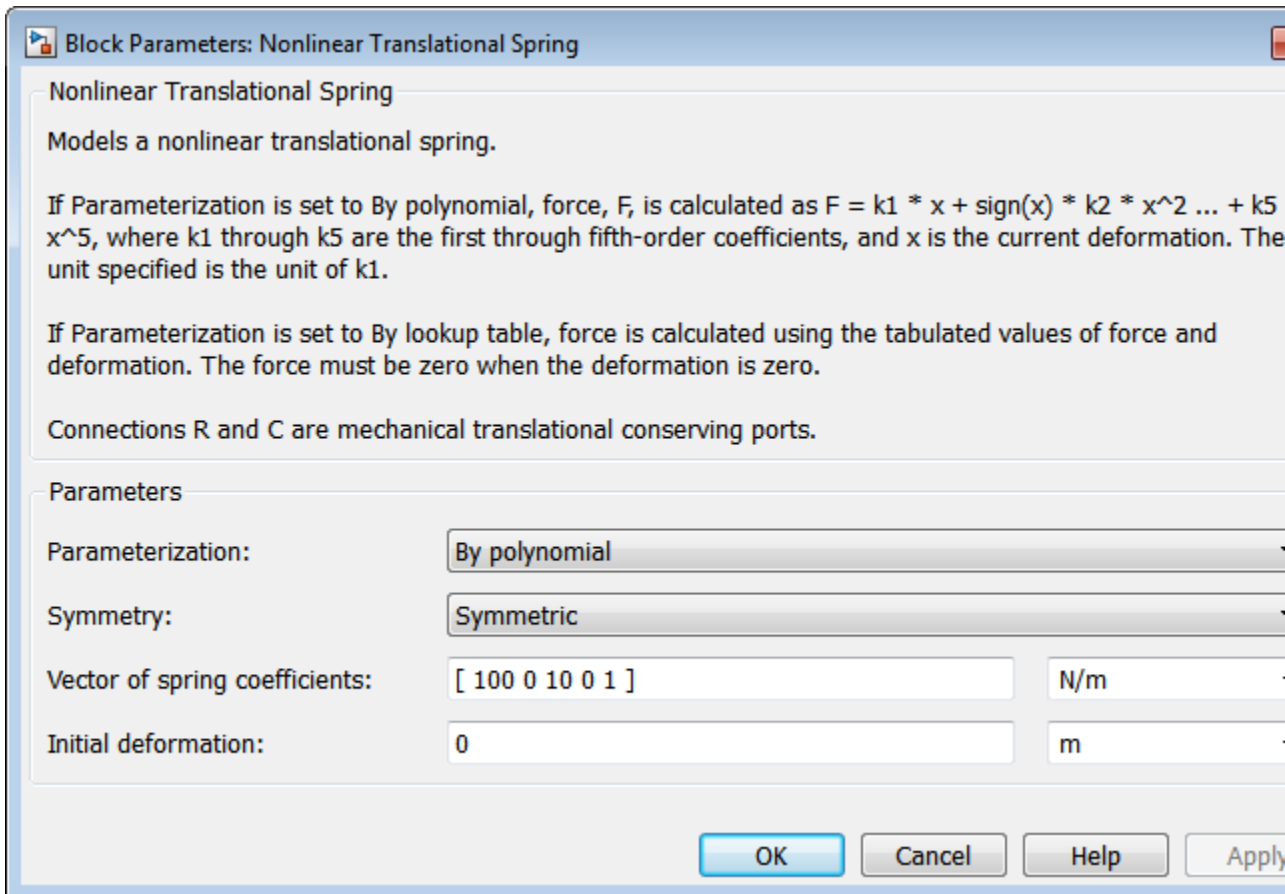
The lookup table parameterization defines damping torque based on a set of torque and angular velocity vectors. If not specified, the block automatically adds a data point at the origin (zero angular velocity and zero torque).

## Dialog Box and Parameters

The dialog box includes one tab, **Parameters**.



# Nonlinear Translational Spring



## Parameterization

Select spring parameterization. Options are By polynomial and By lookup table.

### By polynomial

Specify coefficients of polynomial spring function.

# Nonlinear Translational Spring

---

## Symmetry

Choose between symmetric and two-sided polynomial parameterizations.

- **Symmetric** — Specify one set of polynomial coefficients governing spring force in both tension and compression.

### Vector of spring coefficients

Enter five-element vector with polynomial spring coefficients. The highest non-zero order must be positive. Physical units are for the first coefficient.

The default vector is [100 0 10 0 1]. The default unit is N/m.

- **Two-sided** — Specify two sets of polynomial coefficients governing spring force, one for positive relative displacements, the other for negative relative displacements.

### Vector of spring tension coefficients

Enter five-element vector with polynomial spring tension coefficients. Physical units are for the first coefficient.

The default vector is [100 0 10 0 1]. The default unit is N/m.

### Vector of spring compression coefficients

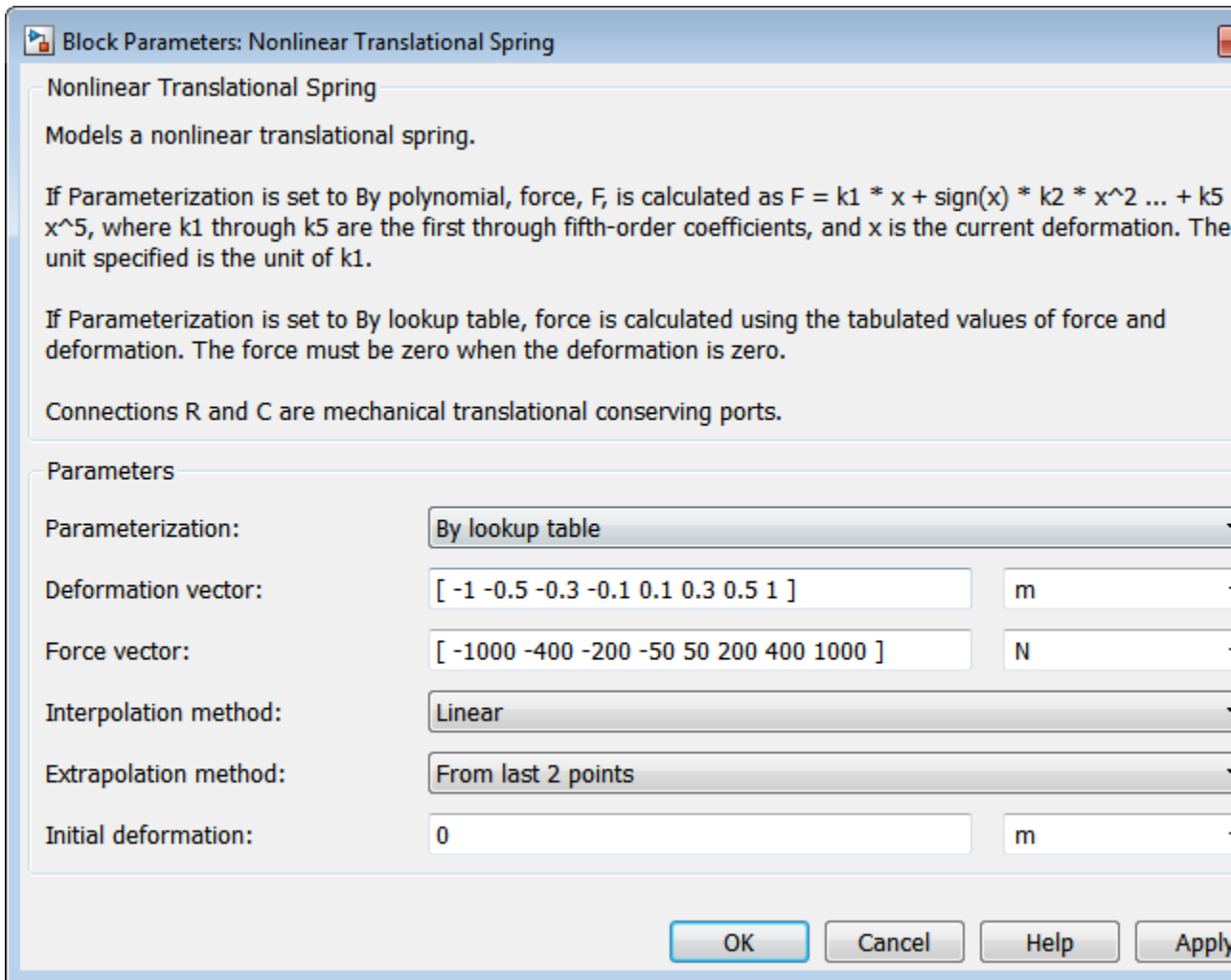
Enter five-element vector with polynomial spring compression coefficients. Physical units are for the first coefficient.

The default vector is [1000 10 100 0 0]. The default unit is N/m.

## By lookup table

Specify deformation and force data for damping lookup table.

# Nonlinear Translational Spring



## Deformation vector

Enter vector with relative displacement values. the vector requires a minimum number of elements, based on the selected interpolation method — two for Linear, and

# Nonlinear Translational Spring

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three for **Cubic** and **Spline**. The number of elements must match the force vector. If not included in the vector, the block automatically adds a point at the origin, corresponding to zero displacement and zero force. If the vector includes a zero value, then the force vector must contain a corresponding zero value.

## Interpolation Method

Select method used to find intermediate velocity–force values between lookup-table data points.

- **Linear** — Interpolate between two points using a first-order polynomial function.
- **Cubic** — Interpolate between two points using a third-order polynomial function.
- **Spline** — Interpolate between two points using a piecewise polynomial function.

## Extrapolation Method

Select method used to calculate values outside the lookup-table data range.

- **From last two points** — Extrapolate by extending the straight line connecting the last two lookup-table data points.
- **From last point** — Extrapolate by extending the horizontal straight line passing through the last lookup-table data point.

## Ports

Port	Description
C	Rotational conserving port
R	Rotational conserving port

## See Also

Variable Translational Spring, Nonlinear Rotational Spring, Variable Rotational Spring

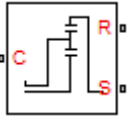
## Purpose

Planetary gear set of carrier, sun, planet, and ring wheels with adjustable gear ratio and friction losses

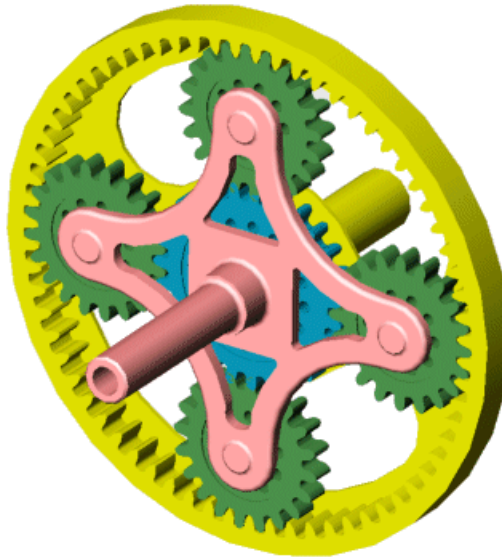
## Library

Gears

## Description



The Planetary Gear block represents a set of carrier, ring, planet, and sun gear wheels. A planetary gear set can be constructed from sun-planet and ring-planet gears. The ring and sun corotate with a fixed gear ratio. For model details, see “Planetary Gear Model” on page 1-226.



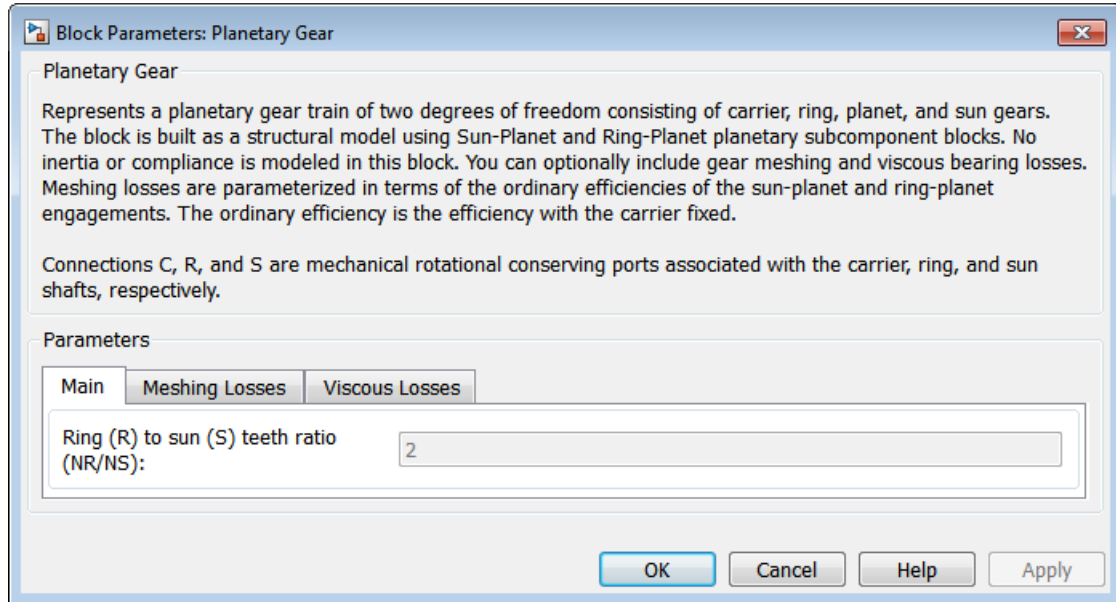
## Planetary Gear Set

### Ports

C, R, and S are rotational conserving ports representing, respectively, the carrier, ring, and sun gear wheels.

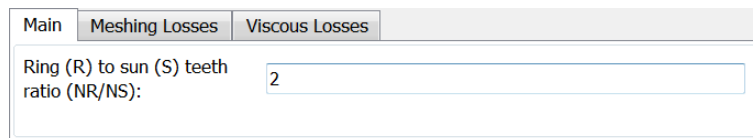
# Planetary Gear

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

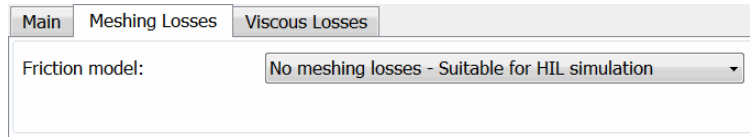
### Main



### Ring (R) to sun (S) teeth ratio (NR/NS)

Ratio  $g_{RS}$  of the ring gear wheel radius to the sun gear wheel radius. This gear ratio must be strictly greater than 1. The default is 2.

## Meshing Losses



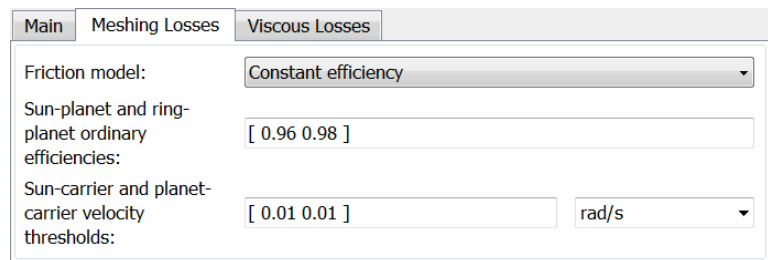
The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Meshing Losses' tab is active. Below the tabs, there is a label 'Friction model:' followed by a dropdown menu. The dropdown menu is open, showing the selected option: 'No meshing losses - Suitable for HIL simulation'.

### Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- **No meshing losses** — Suitable for HIL simulation — Gear meshing is ideal.
- **Constant efficiency** — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

### Constant Efficiency



The screenshot shows the same software interface as above, but with the 'Constant efficiency' option selected in the 'Friction model' dropdown. Below the dropdown, there are three input fields: 'Sun-planet and ring-planet ordinary efficiencies:' with a value of '[ 0.96 0.98 ]', 'Sun-carrier and planet-carrier velocity thresholds:' with a value of '[ 0.01 0.01 ]', and a unit dropdown menu set to 'rad/s'.

### Sun-planet and ring-planet ordinary efficiencies

Vector of torque transfer efficiencies  $[\eta_{SP} \ \eta_{RP}]$  for sun-planet and ring-planet gear wheel pair meshings, respectively.

# Planetary Gear

## Sun-carrier and planet-carrier velocity thresholds

Vector of absolute angular velocity thresholds  $\omega_{th}$  above which full efficiency loss is applied, for sun-carrier and planet-carrier velocities  $[\omega_s \ \omega_p]$ , respectively.

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses	
Sun-carrier and planet-carrier viscous friction coefficients:		<input type="text" value="[ 0 0 ]"/>	N*m/(rad/s) ▾

## Sun-carrier and planet-carrier viscous friction coefficients

Vector of viscous friction coefficients  $[\mu_s \ \mu_p]$  for the sun-carrier and planet-carrier gear motions, respectively. The default is [0 0].

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Planetary Gear Model

### Ideal Gear Constraints and Gear Ratios

Planetary Gear imposes two kinematic and two geometric constraints on the three connected axes and the fourth, internal gear (planet):

$$r_C \omega_C = r_S \omega_S + r_P \omega_P, \quad r_C = r_S + r_P,$$

$$r_R \omega_R = r_C \omega_C + r_P \omega_P, \quad r_R = r_C + r_P.$$

The ring-sun gear ratio  $g_{RS} = r_S/r_S = N_R/N_S$ .  $N$  is the number of teeth on each gear. In terms of this ratio, the key kinematic constraint is:



$$(1 + g_{RS})\omega_C = \omega_S + g_{RS}\omega_R .$$

The four degrees of freedom reduce to two independent degrees of freedom. The gear pairs are (1,2) = (S,P) and (P,R).

## Warning

**The gear ratio  $g_{RS}$  must be strictly greater than one.**

The torque transfer is:

$$g_{RS}t_S + t_R - t_{\text{loss}} = 0 ,$$

with  $t_{\text{loss}} = 0$  in the ideal case.

## Nonideal Gear Constraints and Losses

In the nonideal case,  $t_{\text{loss}} \neq 0$ . See “Model Gears with Losses”.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## See Also

Compound Planetary Gear, Ravigneaux Gear, Ring-Planet, Sun-Planet  
“Represent and Transfer Driveline Motion and Torque”

# Planet-Planet

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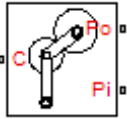
## Purpose

Planetary gear set of carrier, inner planet, and outer planet wheels with adjustable gear ratio and friction losses

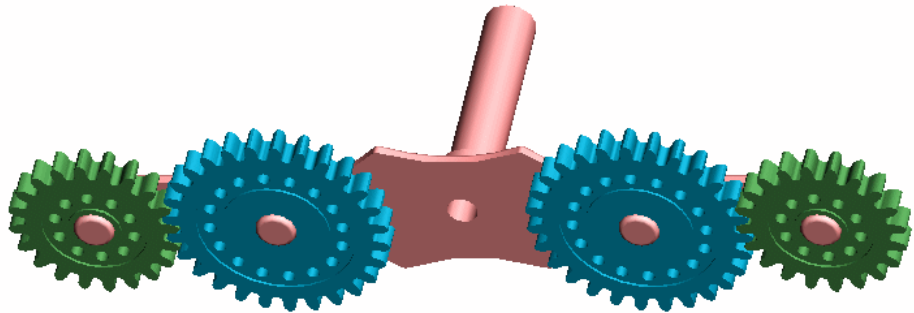
## Library

Gears/Planetary Subcomponents

## Description



The Planet-Planet gear block represents a set of carrier, inner planet, and outer planet gear wheels. Both planetary gears are connected to and rotate with respect to the carrier. The planets corotate with a fixed gear ratio that you specify. For model details, see “Planet-Planet Gear Model” on page 1-231.

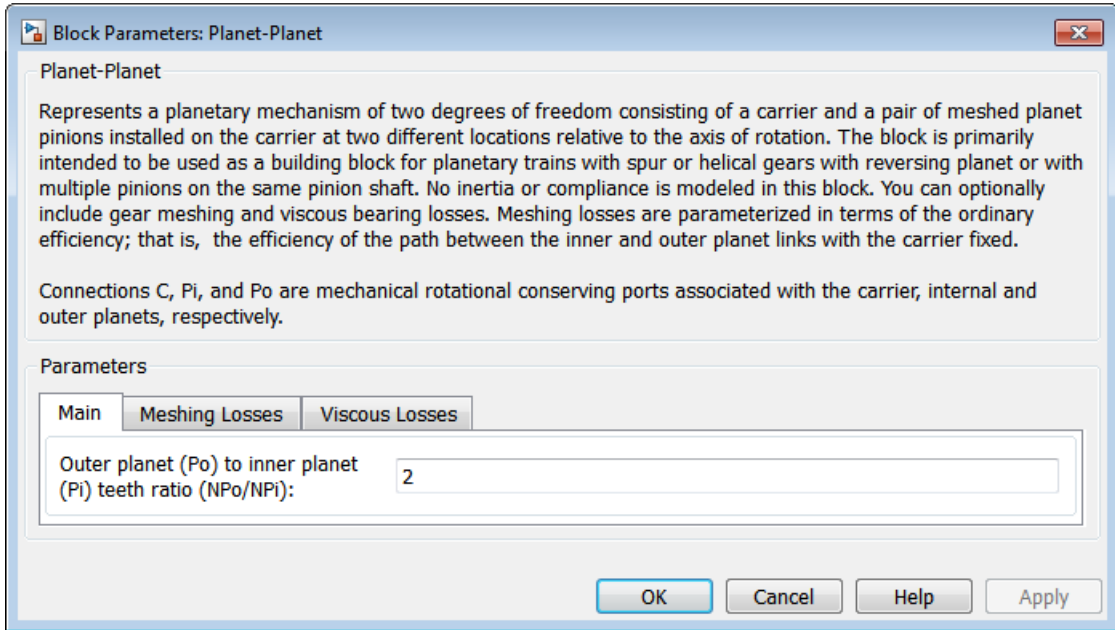


## Planet-Planet Gear Set

### Ports

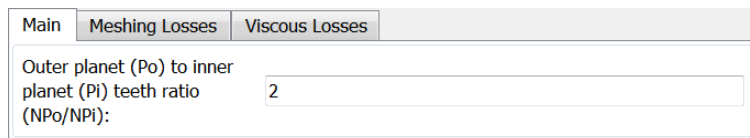
C, Po, and Pi are rotational conserving ports representing, respectively, the carrier, outer planet, and inner planet gear wheels.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

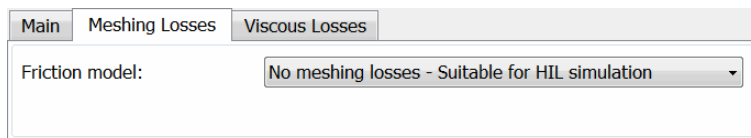
### Main



## Outer planet (Po) to inner planet (Pi) teeth ratio (N<sub>Po</sub>/N<sub>Pi</sub>)

Ratio  $g_{oi}$  of the outer planet gear radius wheel to the inner planet gear wheel radius. This gear ratio must be strictly positive. The default is 2.

## Meshing Losses



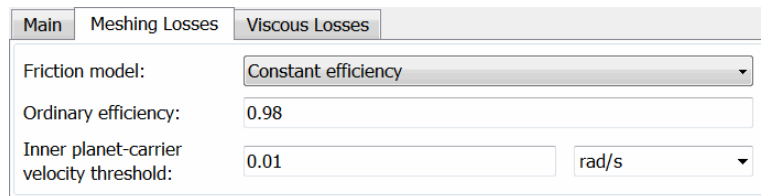
The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Meshing Losses' tab is active. Below the tabs, there is a label 'Friction model:' followed by a dropdown menu. The dropdown menu is open, showing the selected option: 'No meshing losses - Suitable for HIL simulation'.

## Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses — Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

## Constant Efficiency



The screenshot shows the same software interface as above, but with the 'Constant efficiency' option selected in the 'Friction model' dropdown. Below the dropdown, there are two input fields: 'Ordinary efficiency:' with the value '0.98' and 'Inner planet-carrier velocity threshold:' with the value '0.01' and a unit dropdown set to 'rad/s'.

## Ordinary efficiency

Torque transfer efficiency  $\eta_{pp}$  for planet-planet gear wheel pair meshing.

### Inner planet-carrier velocity threshold

Absolute angular velocity threshold  $\omega_{th}$  above which full efficiency loss is applied, for inner planet-carrier velocity  $\omega_{P_i}$ .

From the drop-down list, choose units. The default is radians/second (rad/s).

### Viscous Losses

Main	Meshing Losses	Viscous Losses	
Inner planet-carrier viscous friction coefficient:			
		<input type="text" value="0"/>	N*m/(rad/s) ▾

### Inner planet-carrier viscous friction coefficient

Viscous friction coefficient  $\mu_{P_i}$  for the inner planet-carrier gear motion. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/ (rad/s)).

## Planet-Planet Gear Model

### Ideal Gear Constraints and Gear Ratios

Planet-Planet imposes one kinematic and one geometric constraint on the three connected axes:

$$r_C \omega_C = r_{P_o} \omega_{P_o} + r_{P_i} \omega_{P_i}, \quad r_C = r_{P_o} + r_{P_i}.$$

The outer planet-to-inner planet gear ratio  $g_{oi} = r_{P_o}/r_{P_i} = N_{P_o}/N_{P_i}$ .  $N$  is the number of teeth on each gear. In terms of this ratio, the key kinematic constraint is:

$$(1 + g_{oi}) \omega_C = \omega_{P_i} + g_{oi} \omega_{P_o}.$$

# Planet-Planet

---

The three degrees of freedom reduce to two independent degrees of freedom. The gear pair is (1,2) = (Pi,Po).

The torque transfer is:

$$g_{oi}\tau_{Pi} + \tau_{Po} - \tau_{loss} = 0 ,$$

with  $\tau_{loss} = 0$  in the ideal case.

## **Nonideal Gear Constraints and Losses**

In the nonideal case,  $\tau_{loss} \neq 0$ . See “Model Gears with Losses”.

## **Limitations**

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## **See Also**

Planetary Gear, Ravigneaux Gear, Ring-Planet, Sun-Planet, Sun-Planet Bevel

“Represent and Transfer Driveline Motion and Torque”.

## Purpose

Rack and pinion gear coupling translational and rotational motion, with adjustable pinion radius and friction losses

## Library

Gears/Rotational-Translational

## Description



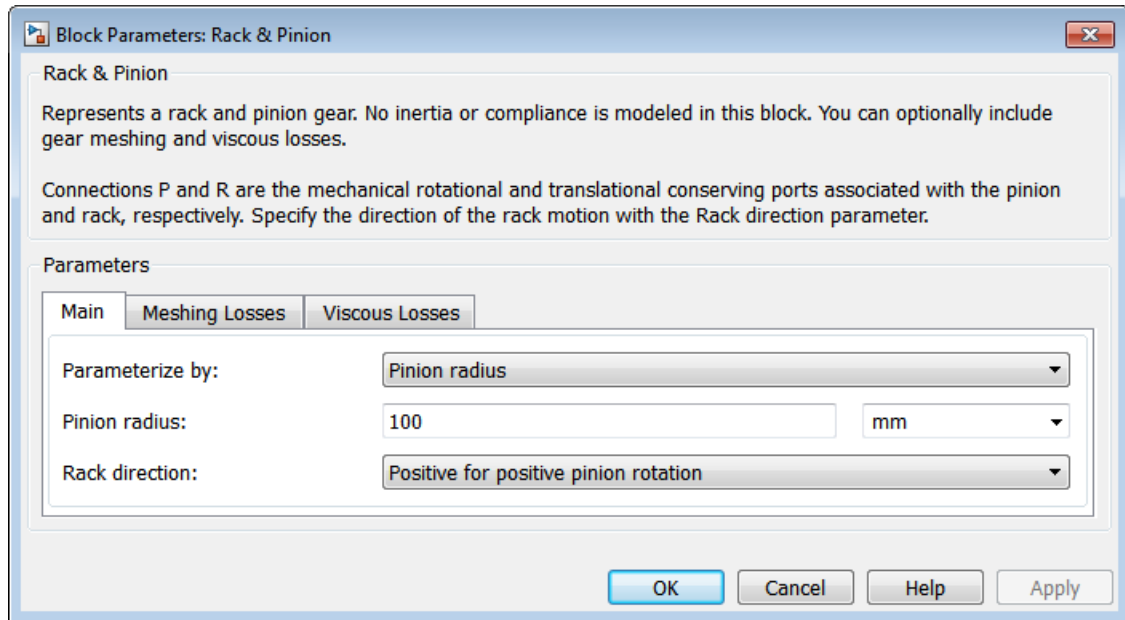
The Rack & Pinion block represents rack and pinion gear that converts between translational and rotational motion. The rotational-translational gear constrains the pinion (P) and rack (R) to, respectively, rotate and translate together in a fixed ratio that you specify. You can choose whether the rack axis translates in a positive or negative direction, as the pinion rotates in a positive direction, by using the **Rack direction** parameter.

## Ports

P is a rotational conserving port. R is a translational conserving port. They represent the pinion and the rack, respectively.

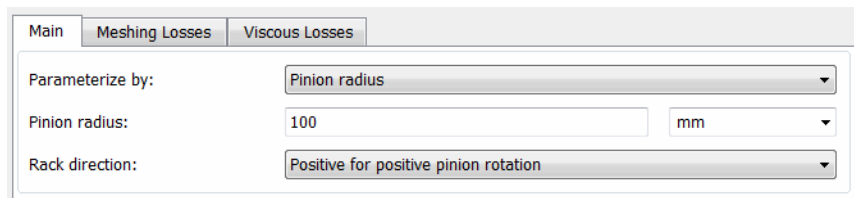
# Rack & Pinion

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main





## Parameterize by

Select how to parameterize the rack and pinion gear. The default is Pinion radius.

- **Pinion radius** — Gear ratio is defined by the effective radius of the pinion.

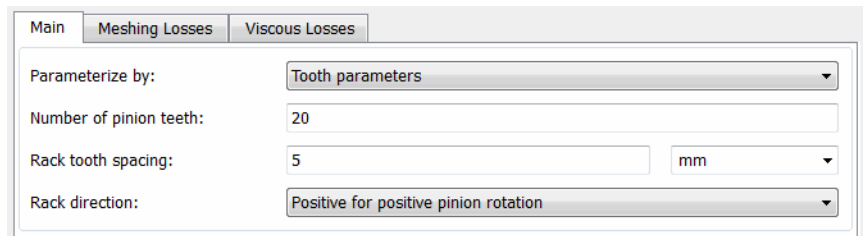
### Pinion radius

Effective radius of the pinion  $r_p$ . Must be greater than zero. The default is 100.

From the drop-down list, choose units. The default is millimeters (mm).

- **Tooth parameters** — Gear ratio is defined by the number of teeth on the pinion gear and the rack tooth spacing. If you select this option, the panel changes from its default.

### Tooth parameters



The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Main' tab is selected. Below the tabs, there are four rows of controls:

- 'Parameterize by:' with a dropdown menu set to 'Tooth parameters'.
- 'Number of pinion teeth:' with a text input field containing '20'.
- 'Rack tooth spacing:' with a text input field containing '5' and a unit dropdown menu set to 'mm'.
- 'Rack direction:' with a dropdown menu set to 'Positive for positive pinion rotation'.

### Number of pinion teeth

Number of teeth on the pinion  $N_p$ . Must be greater than zero. The default is 20.

### Rack tooth spacing

Spacing between the teeth on the rack  $x_R$ . Must be greater than zero. The default is 5.

# Rack & Pinion

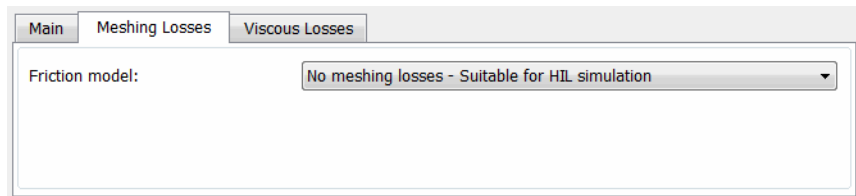
---

From the drop-down list, choose units. The default is millimeters (mm).

## Rack direction

Choose whether the rack axis translates in a positive or negative direction when the pinion rotates in a positive direction. The default is Positive for positive pinion rotation.

## Meshing Losses



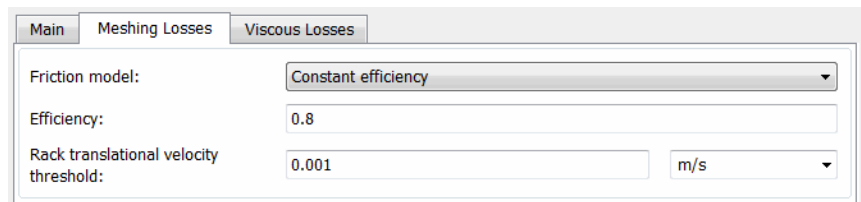
The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Meshing Losses' tab is active. Below the tabs, there is a label 'Friction model:' followed by a dropdown menu. The dropdown menu is open, showing the selected option: 'No meshing losses - Suitable for HIL simulation'.

## Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque and force between pinion and rack is reduced by constant efficiency  $0 < \eta \leq 1$ . If you select this option, the panel expands.

## Constant Efficiency



The screenshot shows the same software interface as above, but with the 'Constant efficiency' option selected in the 'Friction model' dropdown. Below the dropdown, there are three input fields: 'Efficiency:' with the value '0.8', 'Rack translational velocity threshold:' with the value '0.001', and a unit dropdown menu set to 'm/s'.

## Efficiency

Torque transfer efficiency  $\eta$  for the rack-pinion gear meshing, which is the same for the forward and reverse power flows. Must be greater than zero but less than, or equal to, one. The default is 0.8.

## Rack translational velocity threshold

Absolute translational velocity threshold  $v_{th}$  above which the full steady-state value of friction force  $F_{Coul}$  is applied, for the rack velocity  $v_R$ . The smaller the parameter value, the steeper is the change of the force. The default is 0.001.

From the drop-down list, choose units. The default is meters/second (m/s).

## Viscous Losses

The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Viscous Losses' tab is active. It contains two input fields, each with a value of '0' and a corresponding drop-down menu for units. The first input is labeled 'Pinion rotational viscous friction coefficient' and has a unit menu set to 'N\*m/(rad/s)'. The second input is labeled 'Rack translational viscous friction coefficient' and has a unit menu set to 'N/(m/s)'.

### Pinion rotational viscous friction coefficient

Viscous friction coefficient  $\mu_p$  for the pinion shaft. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

### Rack translational viscous friction coefficient

Viscous friction coefficient  $\mu_R$  for the rack motion. The default is 0.

From the drop-down list, choose units. The default is newton/(meters/second) (N/(m/s)).

# Rack & Pinion

---

## Gear Model

### Model Variables

$R_{RP}$	Rack-pinion gear ratio
$\omega_P$	Angular velocity of the pinion shaft
$v_R$	Translational velocity of the rack
$r_P$	Effective radius of the pinion
$N_P$	Number of teeth on the pinion
$x_R$	Rack tooth spacing
$\tau_P$	Pinion shaft torque
$F_R$	Rack force
$F_{\text{loss}}$	Total loss force
$F_{\text{Coul}}$	Friction force
$\eta$	Torque transfer efficiency
$v_{\text{th}}$	Absolute translational velocity threshold
$\mu_P$	Viscous friction coefficient for the pinion shaft
$\mu_R$	Viscous friction coefficient for the rack motion

### Ideal Gear Constraint and Gear Ratio

Rack & Pinion imposes one kinematic constraint on the two connected axes:

$$\omega_P = R_{RP} v_R .$$

The transmission ratio is:

$$R_{RP} = 1 / r_P = \omega_P / v_R = \pm 2\pi / N_P v_R .$$

The two degrees of freedom are reduced to one independent degree of freedom. The forward-transfer gear pair convention is (1,2) = (P,R).

The torque-force transfer is:

$$R_{RP}\tau_P + F_R - F_{\text{loss}} = 0 ,$$

with  $F_{\text{loss}} = 0$  in the ideal case.

## Nonideal Gear Constraint

In a nonideal pinion-rack pair (P,R), the angular velocity and geometric constraints are unchanged. But the transferred torque, force, and power are reduced by:

- Coulomb friction between teeth surfaces on P and R, characterized by constant efficiency  $\eta$
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients  $\mu$

The loss force has the general form:

$$F_{\text{loss}} = F_{\text{Coul}} \cdot \tanh(4v_R/v_{\text{th}}) + \mu_P\omega_P R_{RP} + \mu_R v_R.$$

The hyperbolic tangent regularizes the sign change in the Coulomb friction force when the rack velocity changes sign.

<b>Power Flow</b>	<b>Power Loss Condition</b>	<b>Output Driveshaft</b>	<b>Coulomb Friction Force <math>F_{\text{Coul}}</math></b>
Forward	$\omega_P\tau_P > F_R v_R$	Rack, $v_R$	$R_{RP} \cdot  \tau_P  \cdot (1 - \eta)$
Reverse	$\omega_P\tau_P \leq F_R v_R$	Pinion, $\omega_P$	$R_{RP} \cdot  \tau_P  \cdot (1 - \eta) / \eta$

## Meshing Efficiency

The efficiency  $\eta$  of meshing between pinion and rack is fully active only if the absolute value of the rack velocity is greater than the velocity threshold  $v_{\text{th}}$ .

If the velocity is less than the threshold, the actual efficiency is automatically regularized to unity at zero velocity.

Efficiency is assumed equal for both the forward and reverse power flow.

# Rack & Pinion

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## Viscous Friction Force

The viscous friction coefficients  $\mu_P$  and  $\mu_R$  control the viscous friction torque and force experienced by the rack and pinion from lubricated, nonideal bearings. The viscous friction torque on the pinion axis is  $-\mu_P\omega_P$ . The viscous friction force on the rack motion is  $-\mu_Rv_R$ .

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## See Also

Leadscrew, Simple Gear

“Represent and Transfer Driveline Motion and Torque”

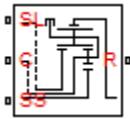
## Purpose

Ravigneaux planetary gear set of carrier, sun, planet, and ring wheels with adjustable gear ratios and friction losses

## Library

Gears

## Description



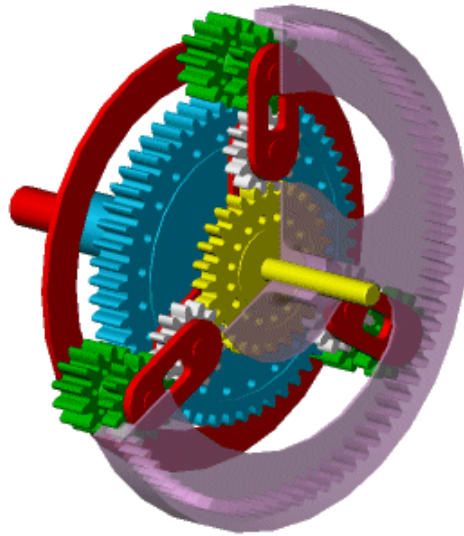
The Ravigneaux block represents a double planetary gear set commonly used in automatic transmissions. This planetary gear set is constructed from three gear pairs, ring-planet, planet-planet, and sun-planet. The Ravigneaux set has two sun gear wheels, a large sun and a small sun, and a single carrier gear with two independent planetary gear wheels connected to it, an inner planet and an outer planet.

- The carrier is one wheel but has two radii to couple with the inner and outer planets, respectively.
- The two planet gears rotate independently of the carrier but corotate in a fixed gear ratio with respect to each other.
  - The inner planet couples to the small sun gear and corotates at a fixed gear ratio.
  - The outer planet couples to the large sun gear and corotates with a fixed gear ratio.
- The ring gear also couples to the outer planet gear and corotates with a fixed gear ratio.

For model details, see “Ravigneaux Gear Model” on page 1-246.

# Ravigneaux Gear

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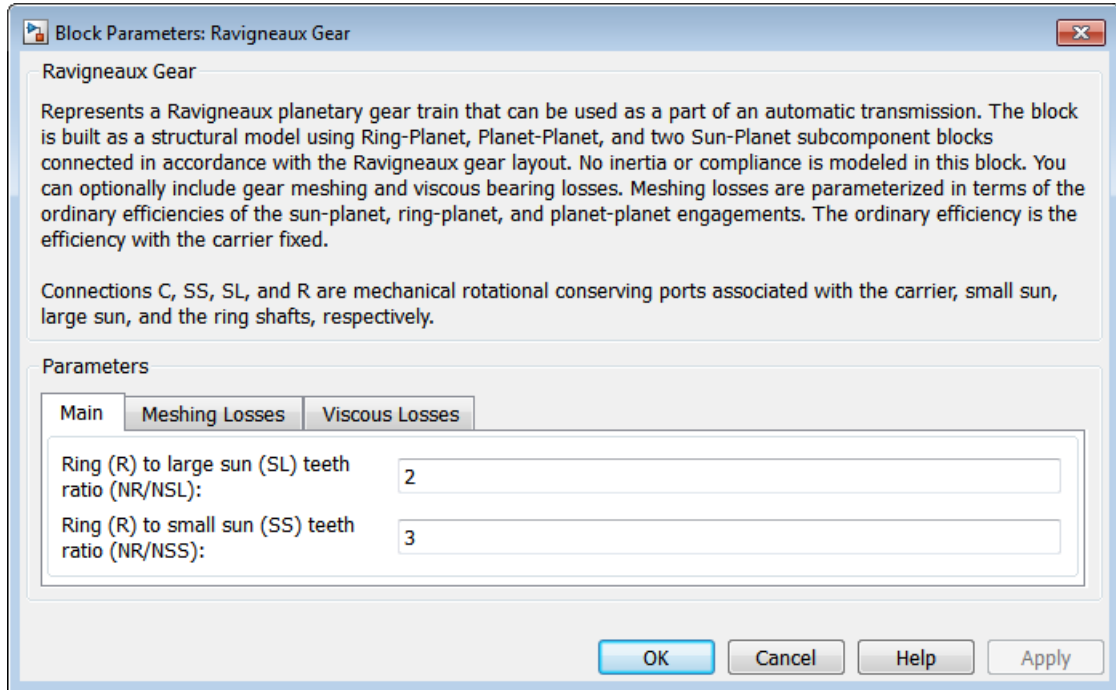
## **Ravigneaux Gear Set**

### **Ports**

C, R, SL, and SS are rotational conserving ports representing, respectively, the carrier, ring, and large and small sun gear wheels.



## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main

Main	Meshing Losses	Viscous Losses
Ring (R) to large sun (SL) teeth ratio (NR/NSL):	2	
Ring (R) to small sun (SS) teeth ratio (NR/NSS):	3	

# Ravigneaux Gear

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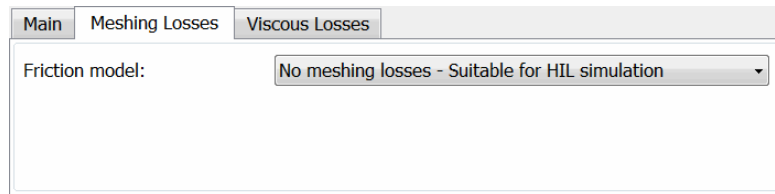
## Ring (R) to large sun (SL) teeth ratio (NR/NSL)

Ratio  $g_{RSL}$  of the ring gear wheel radius to the large sun gear wheel radius. This gear ratio must be strictly greater than 1. The default is 2.

## Ring (R) to small sun (SS) teeth ratio (NR/NSS)

Ratio  $g_{RSS}$  of the ring gear wheel radius to the small sun gear wheel radius. This gear ratio must be strictly greater than the ring-large sun gear ratio. The default is 3.

## Meshing Losses



Friction model: No meshing losses - Suitable for HIL simulation

## Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

## Constant Efficiency

Main	Meshing Losses	Viscous Losses
Friction model: <span style="float: right;">Constant efficiency ▾</span>		
Large sun-planet, small sun-planet, ring-planet and planet-planet ordinary efficiencies:		<input type="text" value="[ 0.98 0.98 0.98 0.99 ]"/>
Large sun-carrier, small sun-carrier, large sun planet-carrier and small sun planet-carrier velocity thresholds:		<input type="text" value="[ 0.01 0.01 0.01 0.01 ]"/> <span style="float: right;">rad/s ▾</span>

### Large sun-planet, small sun-planet, ring-planet, and planet-planet ordinary efficiencies

Vector of torque transfer efficiencies  $[\eta_{LSP} \eta_{SSP} \eta_{RP} \eta_{PP}]$  for large sun-planet, small sun-planet, ring-planet, and planet-planets gear wheel pair meshings, respectively.

### Large sun-carrier, small sun-carrier, large sun planet-carrier, and small sun planet-carrier velocity thresholds

Vector of absolute angular velocity thresholds  $\omega_{th}$  above which full efficiency loss is applied, for large sun-carrier, small sun-carrier, large sun planet-carrier, and small sun planet-carrier velocities  $[\omega_{LS} \omega_{SS} \omega_{LSP} \omega_{SSP}]$ , respectively.

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses
Large sun-carrier, small sun-carrier, large sun planet-carrier and small sun planet-carrier viscous friction coefficients:		
<input type="text" value="[ 0 0 0 0 ]"/>		<input type="text" value="N*m/(rad/s)"/>

### Large sun-carrier, small sun-carrier, large sun planet-carrier, and small sun planet-carrier viscous friction coefficients

Vector of viscous friction coefficients  $[\mu_{LS} \mu_{SS} \mu_{LSP} \mu_{SPP}]$  for the large sun-carrier, small sun-carrier, large sun planet-carrier, and small sun planet-carrier gear motions, respectively. The default is  $[0 \ 0 \ 0 \ 0]$ .

From the drop-down list, choose units. The default is newton-meters/(radians/second) ( $N*m / (rad/s)$ ).

## Ravigneaux Gear Model

### Ideal Gear Constraints and Gear Ratios

Ravigneaux imposes four kinematic and four geometric constraints on the four connected axes and the two internal wheels (inner and outer planets):

$$r_{Ci}\omega_C = r_{SS}\omega_{SS} + r_{Pi}\omega_{Pi}, \quad r_{Ci} = r_{SS} + r_{Pi},$$

$$r_{Co}\omega_C = r_{SL}\omega_{SL} + r_{Po}\omega_{Po}, \quad r_{Co} = r_{SL} + r_{Po},$$

$$(r_{Co} - r_{Ci})\omega_C = r_{Pi}\omega_{Pi} + r_{Po}\omega_{Po}, \quad r_{Co} - r_{Ci} = r_{Po} + r_{Pi},$$

$$r_R\omega_R = r_{Co}\omega_C + r_{Po}\omega_{Po}, \quad r_R = r_{Co} + r_{Po}.$$

The ring-small sun ratio  $g_{RSS} = r_R/r_{SS} = N_R/N_{SS}$  and ring-large sun gear ratio  $g_{RSL} = r_R/r_{SL} = N_R/N_{SL}$ .  $N$  is the number of teeth on each gear. In terms of these ratios, the key kinematic constraints are:

$$(g_{RSS} - 1)\omega_C = g_{RSS}\omega_R - \omega_{SS} ,$$

$$(g_{RSL} + 1)\omega_C = g_{RSL}\omega_R + \omega_{SL} .$$

The six degrees of freedom reduce to two independent degrees of freedom. The gear pairs are (1,2) = (LS,P), (SS,P), (P,R), and (P,P).

## Warning

**The gear ratio  $g_{RSS}$  must be strictly greater than the gear ratio  $g_{RSL}$ . The gear ratio  $g_{RSL}$  must be strictly greater than one.**

The torque transfers are:

$$g_{RSS}\tau_{SS} + \tau_R - \tau_{loss}(SS,R) = 0 , g_{RSL}\tau_{SL} + \tau_R - \tau_{loss}(SL,R) = 0 ,$$

with  $\tau_{loss} = 0$  in the ideal case.

## Nonideal Gear Constraints and Losses

In the nonideal case,  $\tau_{loss} \neq 0$ . See “Model Gears with Losses”.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## Example

The `sdl_Ravigneaux_4_speed` example model uses a Ravigneaux gear as the core of its transmission.

## See Also

Compound Planetary Gear, Planet-Planet, Planetary Gear, Ring-Planet, Sun-Planet

“Represent and Transfer Driveline Motion and Torque”

# Ring-Planet

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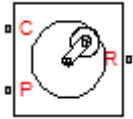
## Purpose

Planetary gear set of carrier, planet, and ring wheels with adjustable gear ratio and friction losses

## Library

Gears/Planetary Subcomponents

## Description



The Ring-Planet gear block represents a set of carrier, planet, and ring gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and ring corotate with a fixed gear ratio that you specify. A ring-planet and a sun-planet gear are basic elements of a planetary gear set. For model details, see “Ring-Planet Gear Model” on page 1-251.

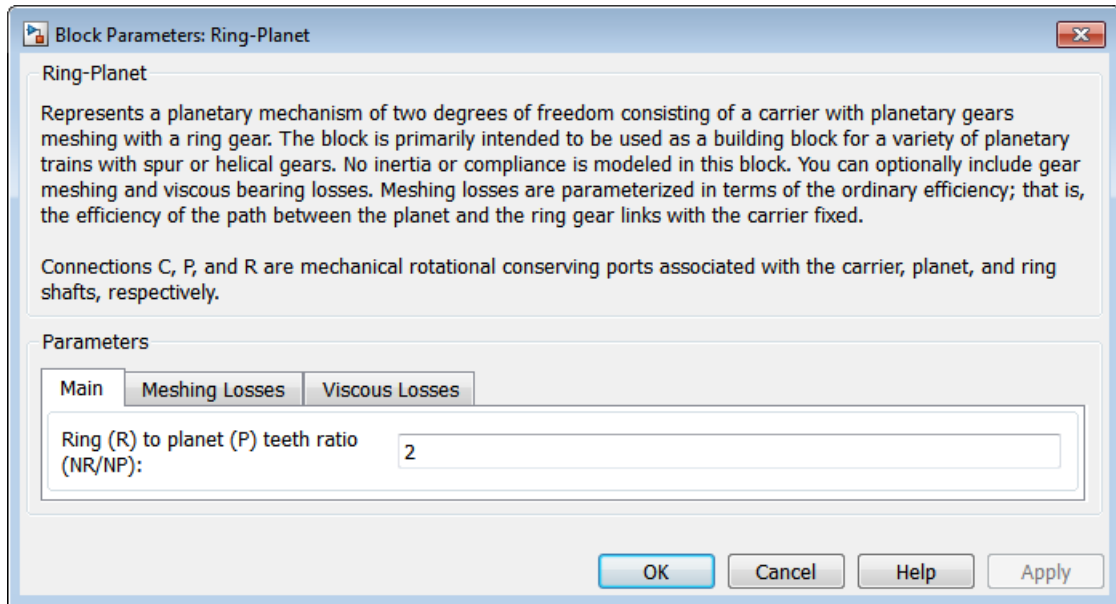


## Ring-Planet Gear Set

### Ports

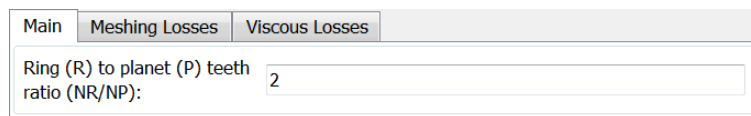
C, P, and R are rotational conserving ports representing, respectively, the carrier, planet, and ring gear wheels.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

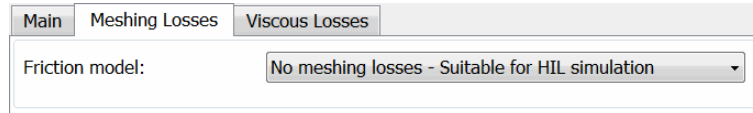
### Main



### Ring (R) to planet (P) teeth ratio (NR/NP)

Ratio  $g_{RP}$  of the ring gear wheel radius to the planet gear wheel radius. This gear ratio must be strictly greater than 1. The default value is 2.

## Meshing Losses



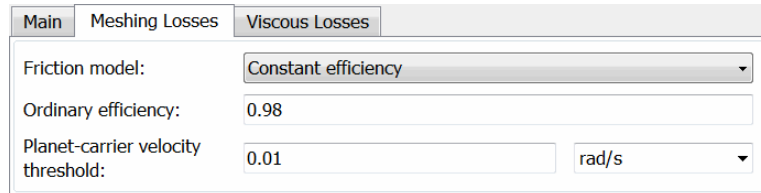
The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Meshing Losses' tab is active. Below the tabs, there is a label 'Friction model:' followed by a dropdown menu. The dropdown menu is open, showing the selected option: 'No meshing losses - Suitable for HIL simulation'.

### Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses — Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

### Constant Efficiency



The screenshot shows the same software interface as above, but with the 'Friction model:' dropdown set to 'Constant efficiency'. Below this, there are two input fields: 'Ordinary efficiency:' with the value '0.98' and 'Planet-carrier velocity threshold:' with the value '0.01'. To the right of the 'Planet-carrier velocity threshold' field is a dropdown menu showing 'rad/s'.

### Ordinary efficiency

Torque transfer efficiency  $\eta_{RP}$  for ring-planet gear wheel pair meshing.

### Planet-carrier velocity threshold

Absolute angular velocity threshold  $\omega_{th}$  above which full efficiency loss is applied, for planet-carrier velocity  $\omega_p$ .

From the drop-down list, choose units. The default is radians/second (rad/s).



## Viscous Losses

Main	Meshing Losses	Viscous Losses	
Planet-carrier viscous friction coefficient:		<input type="text" value="0"/>	N*m/(rad/s) ▾

### Planet-carrier viscous friction coefficient

Viscous friction coefficient  $\mu_p$  for the planet-carrier gear motion. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Ring-Planet Gear Model

### Ideal Gear Constraints and Gear Ratios

Ring-Planet imposes one kinematic and one geometric constraint on the three connected axes:

$$r_R \omega_R = r_C \omega_C + r_P \omega_P, \quad r_R = r_C + r_P.$$

The ring-planet gear ratio  $g_{RP} = r_R/r_P = N_R/N_P$ .  $N$  is the number of teeth on each gear. In terms of this ratio, the key kinematic constraint is:

$$g_{RP} \omega_R = \omega_P + (g_{RP} - 1) \omega_C.$$

The three degrees of freedom reduce to two independent degrees of freedom. The gear pair is (1,2) = (P,R).

### Warning

**The ring-planet gear ratio  $g_{RP}$  must be strictly greater than one.**

The torque transfer is:

$$g_{RP} \tau_P + \tau_R - \tau_{\text{loss}} = 0,$$

# Ring-Planet

---

with  $\tau_{\text{loss}} = 0$  in the ideal case.

## **Nonideal Gear Constraints and Losses**

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . See “Model Gears with Losses”.

### **Limitations**

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

### **Example**

The `sdl_epicyclic_gearbox` example model uses two Ring-Planet gears to model a nonideal epicyclic gear set.

### **See Also**

Compound Planetary Gear, Planet-Planet, Planetary Gear, Sun-Planet, Sun-Planet Bevel

“Represent and Transfer Driveline Motion and Torque”  
“Adjust Model Fidelity”

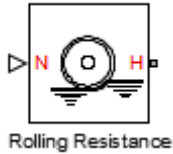
## Purpose

Model rolling resistance

## Library

Tires & Vehicles/Tire Subcomponents

## Description



The block models the resistance force that acts on the wheel hub due to the rolling resistance at the road-wheel contact surface. The model can use a constant resistance coefficient or the pressure and velocity dependence of the SAE J2452 standard. The resistance force is zero when the normal force acting at the wheel-road surface is less than or equal to zero.

### Constant Resistance Coefficient Model

In the constant coefficient model, the rolling resistance is directly proportional to the resistance coefficient:

$$F=N\mu$$

where parameters represent the following quantities:

- $F$ —Rolling resistance force
- $N$ —Normal Force
- $\mu$ —Rolling resistance coefficient

The rolling resistance coefficient has a hyperbolic form that eliminates discontinuity at  $v_{hub}=0$ :

$$\mu=\mu_0\tanh(4v_{hub}/v_{threshold})$$

where parameters represent the following quantities:

- $\mu_0$ —Asymptotic rolling resistance coefficient
- $v_{hub}$ —Hub velocity
- $v_{threshold}$ —Threshold velocity

# Rolling Resistance

---

## Pressure and Velocity Dependent Model

The pressure- and velocity-dependent model uses the following formula:

$$F = \frac{P}{P_0}^\alpha \frac{N}{N_0}^\beta N_0 (A + B|v_{hub}| + Cv_{hub}^2)$$

where parameters represent the following quantities:

- $P$ —Tire pressure
- $v_{hub}$ —Hub velocity
- $\alpha, \beta, A, B, C$ —Approximating coefficients
- $P_0$ —1 Pascal (Pa)
- $N_0$ —1 Newton (N)

In the previous equation, parameters  $P_0$  and  $N_0$  remove the physical units from each exponential expression base.

Connection H is a mechanical translational conserving port that represents the hub of the tire. Connection N is a physical signal input port that represents the normal force acting on the tire. Normal force is positive if it points downward.

## Dialog Box and Parameters

The dialog box has one active area, **Parameters**.

### Resistance Model

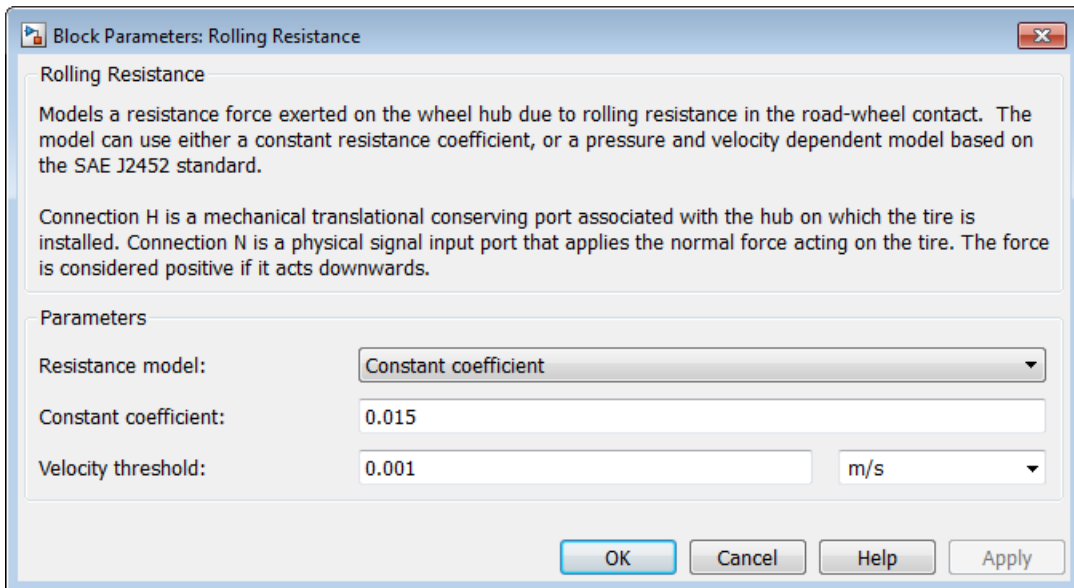
Select the model used to compute the rolling resistance on a wheel hub. The parameter provides two options:

- Constant coefficient
- Pressure and velocity dependent

The default value is Constant coefficient.

## Constant Coefficient

Selecting the Constant coefficient option exposes two model parameters: **Constant coefficient** and **Velocity threshold**.



## Constant Coefficient

Coefficient that sets the proportionality between the normal force and the rolling resistance force. The parameter must be greater than zero. The default value is 255.

## Velocity Threshold

Velocity at which the full rolling resistance force is transmitted to the rolling hub. The parameter ensures the force remains continuous during velocity direction changes, which increases the numerical stability of the simulation. The parameter must be greater than zero. The default value is 0.001 m/s.

# Rolling Resistance

## Pressure and Velocity Dependent

**Block Parameters: Rolling Resistance**

**Rolling Resistance**

Models a resistance force exerted on the wheel hub due to rolling resistance in the road-wheel contact. The model can use either a constant resistance coefficient, or a pressure and velocity dependent model based on the SAE J2452 standard.

Connection H is a mechanical translational conserving port associated with the hub on which the tire is installed. Connection N is a physical signal input port that applies the normal force acting on the tire. The force is considered positive if it acts downwards.

**Parameters**

Resistance model: Pressure and velocity dependent

Tire pressure: 250e3 Pa

Alpha: -0.003

Beta: 0.97

Coefficient A: 84e-4

Coefficient B: 6.2e-4 s/m

Coefficient C: 1.6e-4 s<sup>2</sup>/m<sup>2</sup>

Velocity threshold: 0.001 m/s

OK Cancel Help Apply

### Tire pressure

Inflation pressure of the tire. The parameter must be greater than zero. The default value is 250e+3 Pa.

### Alpha

Exponent of the tire pressure in the model equation. See “Pressure and Velocity Dependent Model” on page 1-254. The default value is -0.003.

**Beta**

Exponent of the normal force model equation. The default value is 0.97.

**Coefficient A**

Velocity independent force component in the model equation. The parameter must be greater than zero. The default value is  $84e-4$ .

**Coefficient B**

Velocity dependent force component in the model equation. The parameter must be greater than zero. The default value is  $6.2e-4$  s/m.

**Coefficient C**

Force component that depends on the square of the velocity term in the model equation. The parameter must be greater than zero. The default value is  $1.6e-4$  s<sup>2</sup>/m<sup>2</sup>.

**Velocity Threshold**

Velocity at which the full rolling resistance force is transmitted to the rolling hub. The parameter ensures the force remains continuous during velocity direction changes, which increases the numerical stability of the simulation. The parameter must be greater than zero. The default value is  $1e-3$  m/s.

**Ports**

Port	Description
N	Physical signal input port that represents the normal force
H	Conserving translational port that represents the wheel hub

**See Also**

Tire-Road Interaction (Magic Formula), Tire (Magic Formula), Tire (Friction Parameterized)

# Rope Drum

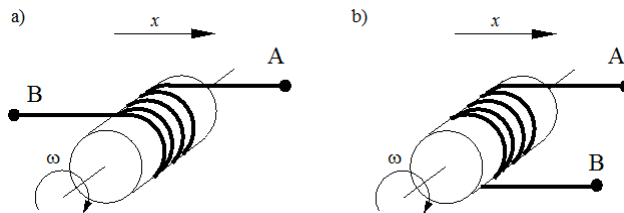
**Purpose** Rope wrapped around a drum

**Library** Couplings & Drives

**Description**



The block represents a drum wrapped with a flexible rope so that no slip exists between the drum and the rope. The block accounts for viscous friction in the drum bearings. You can include drum inertia. The block simulates drums with two types of the rope windup.



In left-hand figure (a), the rope is wound around the drum in such a way that its ends move in the same direction. In the right-hand figure (b), the rope ends move in opposite direction.

Rope compliance is not modeled. The model is valid when both ends of the rope are in tension. An optional warning can display at the MATLAB command line when either rope end loses tension. When assembling a model, ensure that tension is maintained throughout the simulation. Interpret the results with this in mind.

The branch A end moves in a positive direction, if the drum rotates in a positive direction.

The branch B end moves either in a positive or negative direction, depending on whether the **Rope windup type** parameter specifies that ends move in the opposite direction or in the same direction.

Positive translation for end A corresponds to positive rotation of the drum.

The three block equations are:



$$T = F_B \cdot R \cdot or - F_A \cdot R + \mu \cdot \omega$$

$$V_B = -\omega \cdot R \cdot or$$

$$V_A = \omega \cdot R$$

where

- $R$  is the drum radius.
- $T$  is the drum torque.
- $F_B$  is the force acting along branch B.
- $F_A$  is the force acting along branch A.
- $\omega$  is the drum angular velocity.
- $V_A$  is the linear velocity of branch A.
- $V_B$  is the linear velocity of branch B.
- $or$  is the parameter that accounts for the rope windup type.
  - If you set the **Rope windup type** parameter to Ends move in the same direction, then  $or$  is set to -1.
  - If you set the **Rope windup type** parameter to Ends move in opposite direction, then  $or$  is set to 1.

Connection S is a conserving rotational port associated with the drum shaft. Connections A and B are the conserving mechanical translational ports associated with the rope ends.

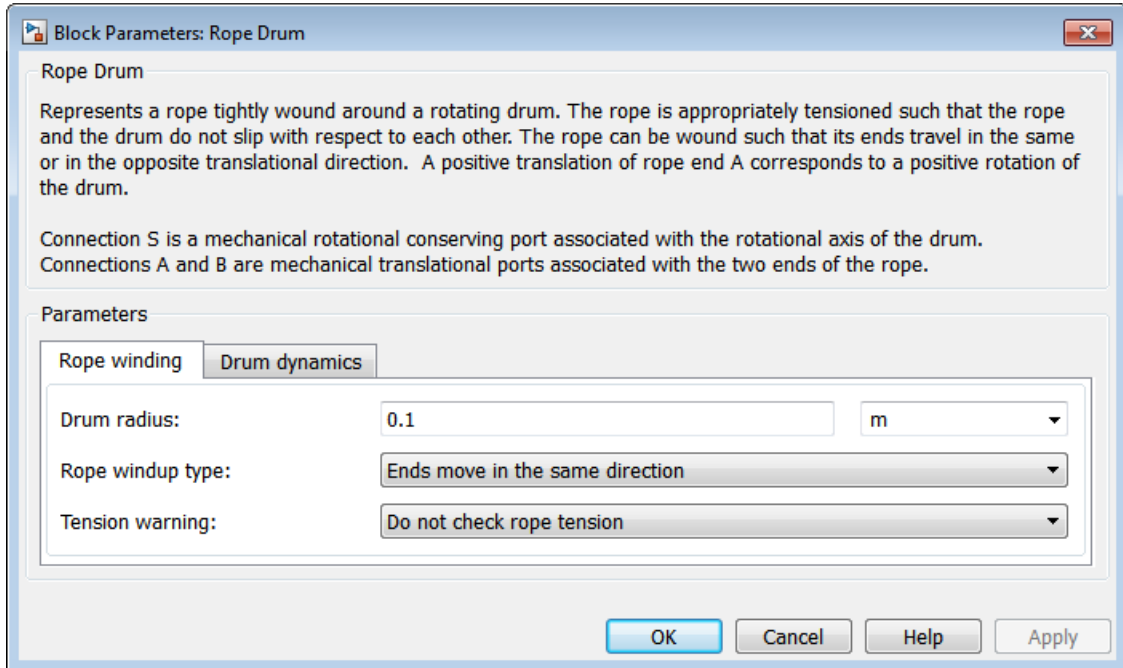
## Assumptions and Limitations

- Both rope ends maintain adequate tension throughout the simulation.
- Rope compliance is not modeled.
- The drum does not translate.

# Rope Drum

## Dialog Box and Parameters

The Rope Drum dialog box includes two tabs. In the **Rope winding** pane, specify the drum radius and type of rope windup.



### Drum radius

The radius of the drum. Default is 0.1 m.

### Rope windup type

Specifies whether the ends of the rope travel in the same or opposite direction.

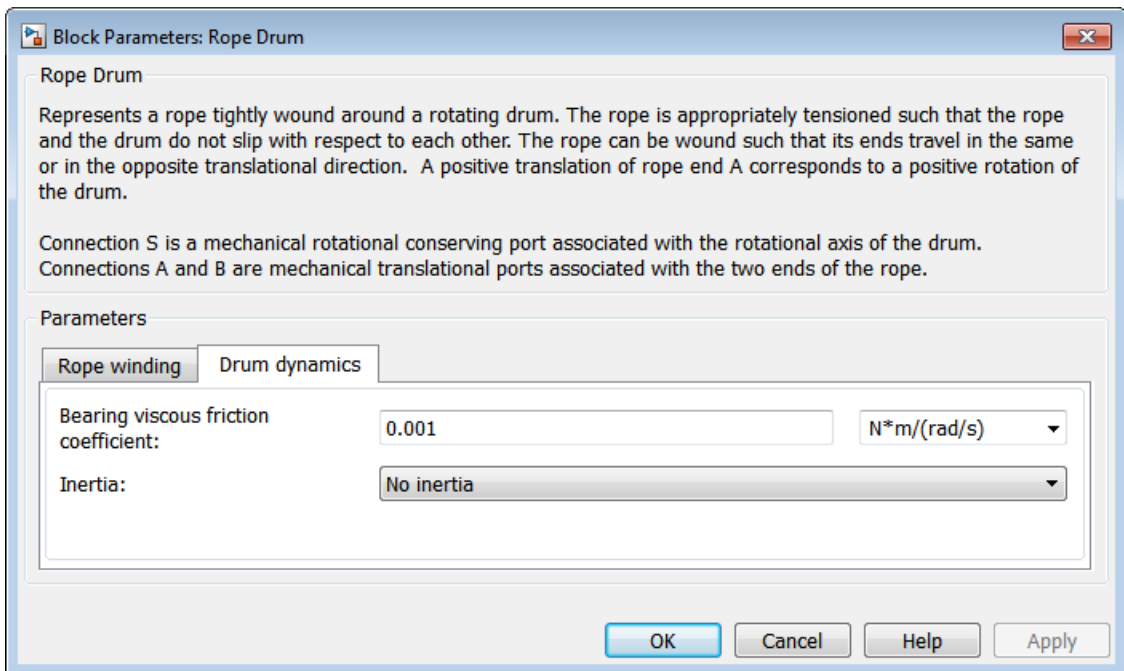
- The ends move in the same direction — (Default)
- The ends move in opposite direction

## Tension Warning

Specifies whether the block writes a warning at the MATLAB command line when the tension at either end of the rope falls below zero.

- Do not check rope tension — (Default)
- Warn if either rope end loses tension

Select the **Drum dynamics** tab to specify a bearing viscous friction coefficient and inertia.



## Bearing viscous friction coefficient

Viscous friction associated with the bearings that hold the axis of the rope drum. Default is  $0.001 \text{ (N}\cdot\text{m) / (rad/s)}$ .

# Rope Drum

---

## Inertia

Specifies whether the block models the rotational inertia of the drum.

- **No inertia** — (Default) Does not model rotational inertia.
- **Specify drum inertia and initial velocity** — Models rotational inertia. Exposes the **Drum inertia** and **Initial rotational velocity** parameters.

## Drum inertia

The rotational inertia of the drum. Default is  $0.01 \text{ kg}\cdot\text{m}^2$ .

The **Drum inertia** parameter is visible only when the **Inertia** parameter is **Specify drum inertia and initial velocity**.

## Drum initial velocity

The initial rotational velocity of the drum. Default is  $0 \text{ rad/s}$ .

The **Drum initial velocity** parameter is visible only when the **Inertia** parameter is **Specify drum inertia and initial velocity**.

## Ports

Port	Description
S	Conserving rotational port associated with the drum shaft
A	Conserving translational port associated with rope end A
B	Conserving translational port associated with rope end B

## Example

Featured example `sdl_power_window` shows an application of the Rope Drum block.

## See Also

Wheel and Axle (in the Simscape documentation)  
“Basic Motion, Torque, and Force Modeling”

**Purpose** Double-sided, spring-loaded rotational detent

**Library** Brakes & Detents/Rotational

## Description



The Rotational Detent block represents a double-sided, spring-loaded rotational detent.

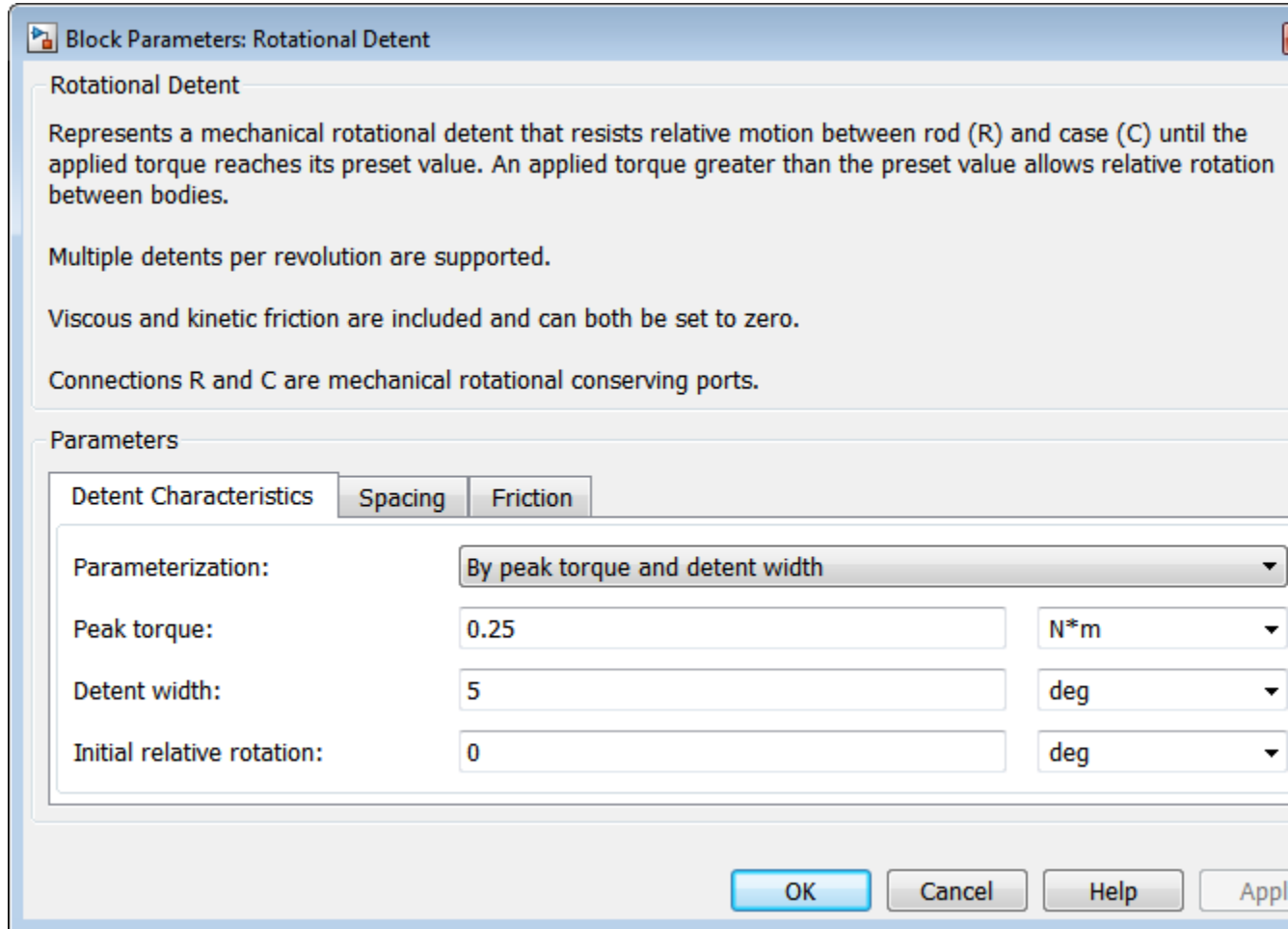
The detent rod rotates over the detent case. The case has notches or magnets that create a net relative torque with the rod as the rod rotates. With the block, you can model rotational detents with peak torque and detent width, or with a lookup table. The model also includes viscous damping and kinetic friction between the rod and case. You can set the friction to zero.

### Ports

R and C are rotational conserving ports associated with the rod and case, respectively.

# Rotational Detent

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

## Detent Characteristics

### Parameterization

Select how to specify the detent characteristics. The default is By peak torque and detent width.

- By peak torque and detent width — Specify detent characteristics by the peak torque and the detent width.

### Peak Torque and Detent Width

#### Peak torque

The peak torque developed inside the detent. The default is 0.25.

From the drop-down list, choose units. The default is newton-meters (N\*m).

#### Detent width

The width of the region where the torque develops. The default is 5.

From the drop-down list, choose units. The default is degrees (deg).

- By lookup table — Define the detent characteristics by one-dimensional table lookup based on the relative displacement between the slider and case. If you select this option, the panel changes from its default option.

# Rotational Detent

## Lookup Table

Block Parameters: Rotational Detent

Rotational Detent

Represents a mechanical rotational detent that resists relative motion between rod (R) and case (C) until the applied torque reaches its preset value. An applied torque greater than the preset value allows relative rotation between bodies.

Multiple detents per revolution are supported.

Viscous and kinetic friction are included and can both be set to zero.

Connections R and C are mechanical rotational conserving ports.

Parameters

Detent Characteristics   Spacing   Friction

Parameterization: By lookup table

Relative rotation vector: [ -5 -4 -3 -1 1 3 4 5 ]   deg

Torque vector: [ 0 0 -0.25 -0.1 0.1 0.25 0 0 ]   N\*m

Initial relative rotation: 0   deg

Interpolation method: Linear

OK   Cancel   Help   Appl



## Relative rotation vector

Specify the vector of input relative rotations that correspond to the values in **Torque vector**. These two vectors must be of the same size. The angles must be specified in strictly increasing order. The default values are [-5.0 -4.0 -3.0 -1.0 1.0 3.0 4.0 5.0].

From the drop-down list, choose units. The default is degrees (deg).

## Torque vector

Specify the vector of input values for the developed torque that correspond to the values in **Relative rotation vector**. The default values are [0 0 -0.25 -0.10 0.10 0.25 0 0].

From the drop-down list, choose units. The default is newton-meters (N\*m).

## Initial relative rotation

The relative angle of the rod and case when simulation starts. The default is 0.

From the drop-down list, choose units. The default is degrees (deg).

## Interpolation method

Select one of the following interpolation methods for approximating the output value when the input value is between two consecutive grid points:

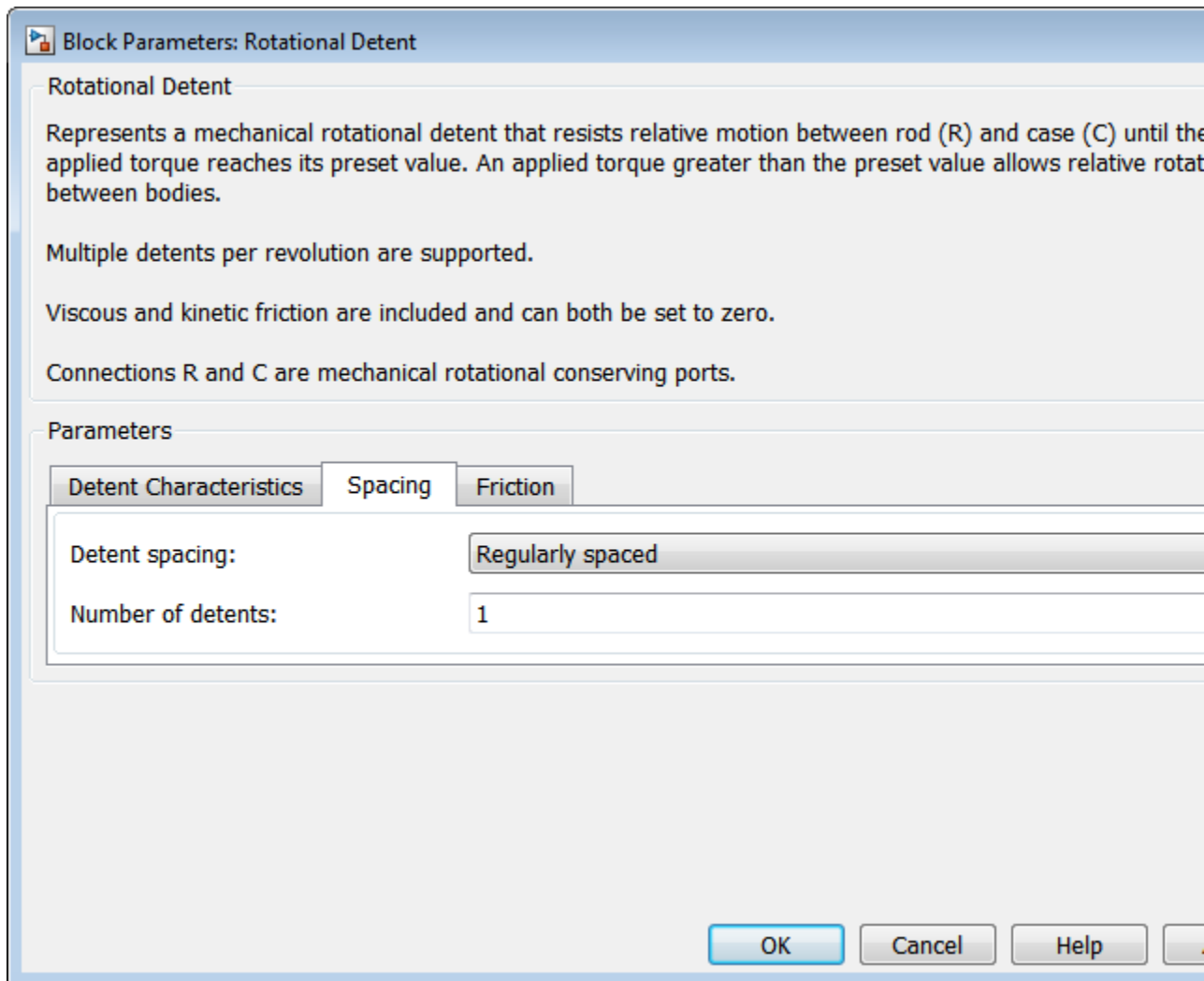
- **Linear** — Uses a linear interpolation function.
- **Cubic** — Uses the Piecewise Cubic Hermite Interpolation Polynomial (PCHIP).
- **Spline** — Uses the cubic spline interpolation algorithm.

# Rotational Detent

---

For more information on interpolation algorithms, see the PS Lookup Table (1D) block reference page.

## Spacing



# Rotational Detent

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## **Detent spacing**

Select how to specify the detent spacing. The default is **Regularly spaced**.

- **Regularly spaced** — Specify detent spacing by regular intervals.

## **Regular spacing**

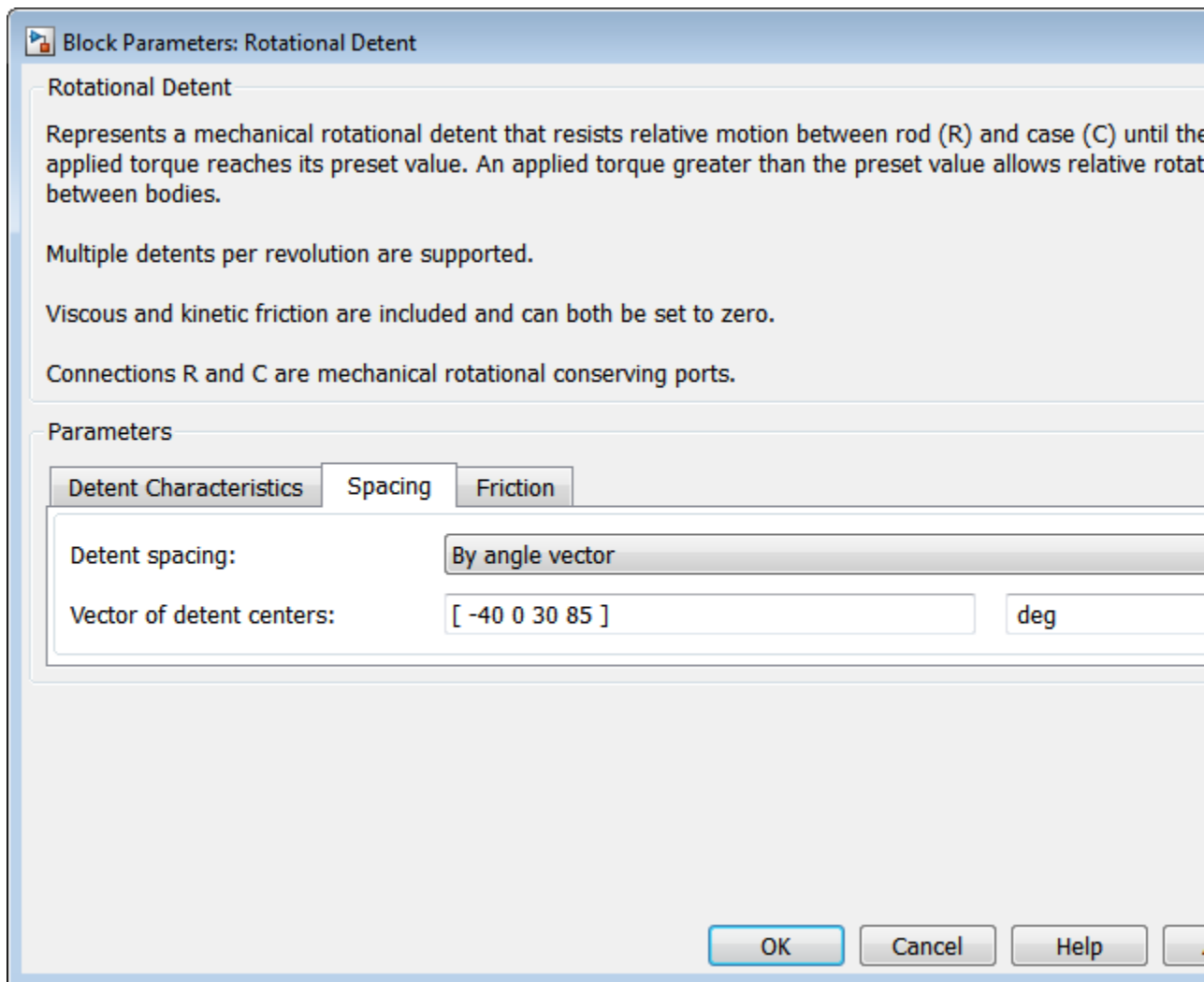
### **Number of detents**

Specify the number of identical, regularly spaced detents. Must be greater than or equal to 1. The default is 1.

Creates a number of detents evenly spaced from  $-180$  to  $+180$  degrees, with one detent center at 0 degrees. Non-integer values are rounded to the nearest integer.

- **By angle vector** — Specify detent spacing by custom intervals. If you select this option, the panel changes from its default option.

## Custom intervals



# Rotational Detent


---

## Vector of detent centers

Specify the locations of the centers of identical detents as a vector of angles. These values must lie between  $-180$  and  $+180$  degrees. The minimum spacing between the detents must be greater than the **Detent width** parameter. The default values are [ -40 0 30 85].

From the drop-down list, choose units. The default is degrees (deg).

## Friction

 Block Parameters: Rotational Detent

**Rotational Detent**

Represents a mechanical rotational detent that resists relative motion between rod (R) and case (C) until the applied torque reaches its preset value. An applied torque greater than the preset value allows relative rotation between bodies.

Multiple detents per revolution are supported.

Viscous and kinetic friction are included and can both be set to zero.

Connections R and C are mechanical rotational conserving ports.

**Parameters**

Detent Characteristics	Spacing	Friction
Viscous friction coefficient:	<input type="text" value="0.1"/>	<input type="text" value="N*m/(rad/s)"/>
Friction to peak torque ratio:	<input type="text" value="0.01"/>	
Friction velocity threshold:	<input type="text" value="10"/>	<input type="text" value="rpm"/>

# Rotational Detent

---

## Viscous friction coefficient

Specify the viscous friction coefficient for the rod-case contact. Must be greater than or equal to 0. The default is 0.1.

From the drop-down list, choose units. The default is newton-meters/(radians/second) ( $\text{N}\cdot\text{m}/(\text{rad}/\text{s})$ ).

## Friction to peak torque ratio

Specify the kinetic friction coefficient for the rod-case contact. Must be greater than or equal to 0. The default is 0.01.

The kinetic friction is this ratio multiplied by the peak torque.

## Friction velocity threshold

Specify the relative angular velocity required for peak kinetic friction in the detent. Must be greater than 0. The default is 10.

From the drop-down list, choose units. The default is revolutions/minute (rpm).

## Rotational Detent Model

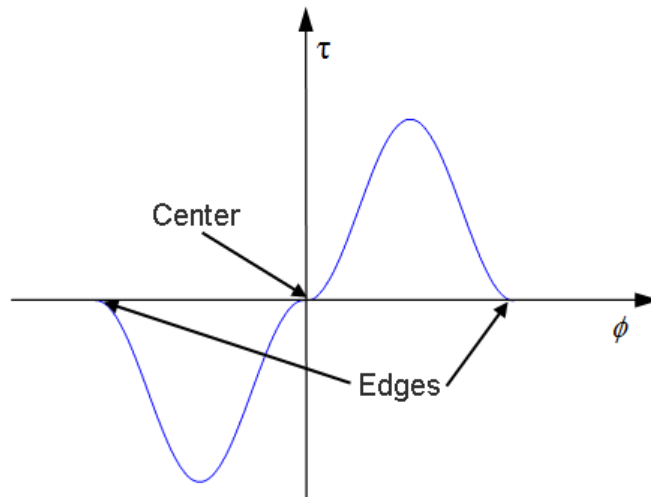
As the rod rotates over the case, the relative rotation  $\varphi = \varphi_R - \varphi_C$  causes a torque  $\tau$  to develop. Depending on your choice of parameterization, the torque model is defined by the peak torque and detent width, or by a lookup table specifying relative rotation versus torque.

Regardless of the parameterization, if you specify multiple detents, all detents have the same torque-relative rotation function.

## Peak Torque and Detent Width

If you choose the peak torque and detent width parameterization, the block ensures that the torque-relative rotation curve provides a continuous torque and torque derivative over the detent region. The peak torques are halfway between the detent center and detent edge, as shown in the following figure.





## Lookup Table

With the lookup table parameterization, you can create an arbitrary function relating torque to relative rotation. If you create such a function, consider the following best practices.

### Energy Conservation

If you want to ensure that the detent conserves energy, the total integral of the torque-relative rotation curve (area under the curve) must be zero.

### Simulation Performance

To stabilize simulation of the detent, avoid discontinuities in the torque-relative rotation function.

## Limitations

The model does not account for inertia. Add inertia terms externally to the R and C ports as required.

If you use the peak torque-detent width or the lookup table parameterization, the kinetic friction is independent of the detent normal force.

## See Also

Loaded-Contact Rotational Friction, Translational Detent

# Rotational Detent

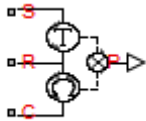
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“Clutches, Clutch-Like Elements, and Coulomb Friction”

**Purpose** Mechanical sensor used to measure average or instantaneous rotational power

**Library** Sensors

## Description



The block represents a sensor that you can use to measure the rotational power change due to one or more driveline components connected between ports R and C. The power output can be an instantaneous or average value. Three averaging options are available: period, vibration, and revolution averaging.

The instantaneous power is the product of the current angular velocity and torque. Use the instantaneous power to analyze peak power transmission and identify heavily-loaded components.

The time-averaged power is the integral of the instantaneous power over one sample period, divided by that period. For simulation times smaller than the sample period, the average power is zero. The power output is constant for each sample period.

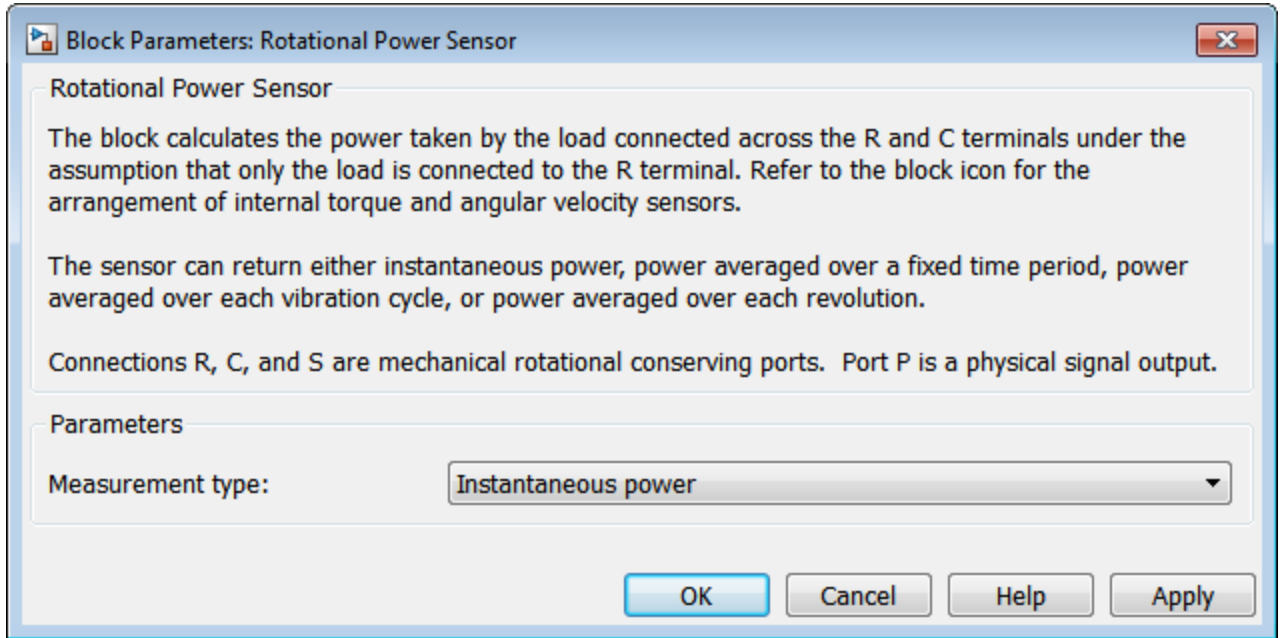
The vibration-averaged power is the integral of the power over one vibration period, divided by that period. The vibration period is the time between two consecutive crossings from negative to positive velocity values. The averaging period can change if the vibration has a variable frequency. For simulation times smaller than the vibration period, the average power is zero. The power output is constant for each vibration period.

The revolution-averaged power is the integral of the instantaneous power over one revolution period, divided by that period. The revolution period is defined as the time between two consecutive crossings of zero angular position. The averaging period can change if the revolution has a variable frequency. For simulation times smaller than the revolution period, the average power is zero. The power output is constant for each revolution period.

# Rotational Power Sensor

## Dialog Box and Parameters

The dialog box contains one active area, **Parameters**.

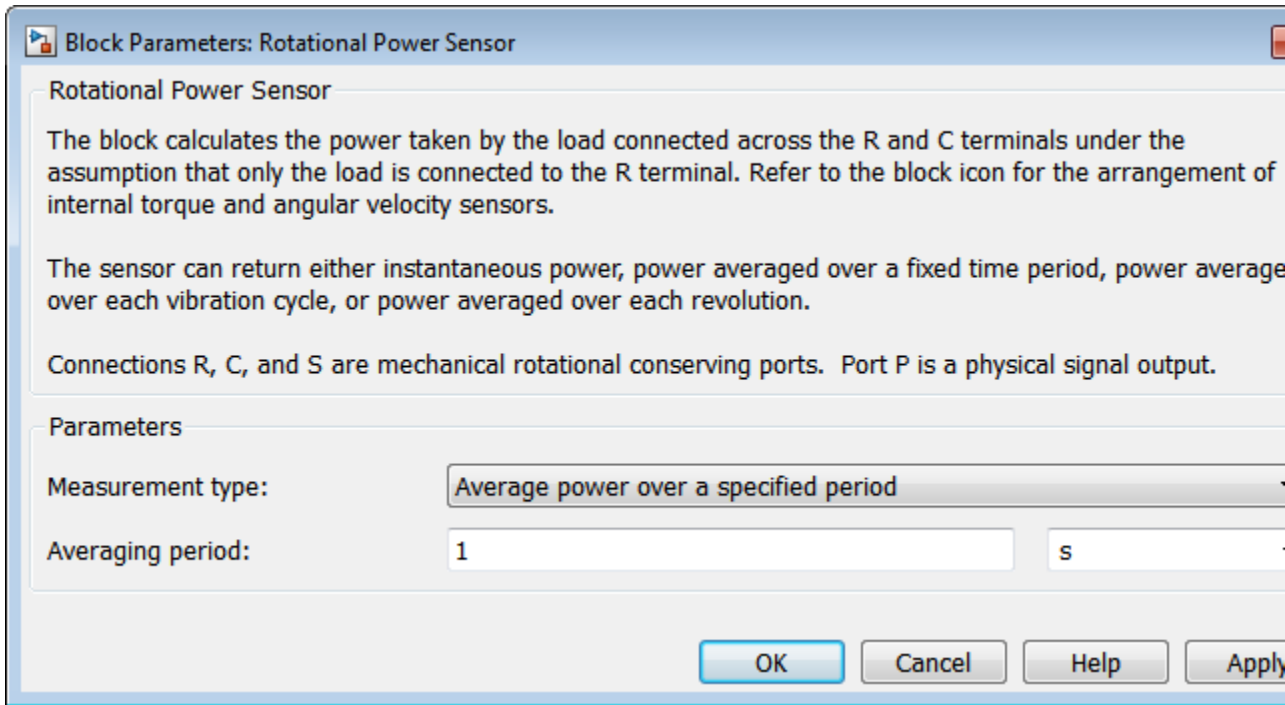


### Measurement Type

Select either instantaneous or average power measurement. The default type is Instantaneous Power.

- Instantaneous power — Output the rotational power measured for the current simulation time step.
- Average power over a specified period — Output the rotational power averaged over a fixed time period.

# Rotational Power Sensor



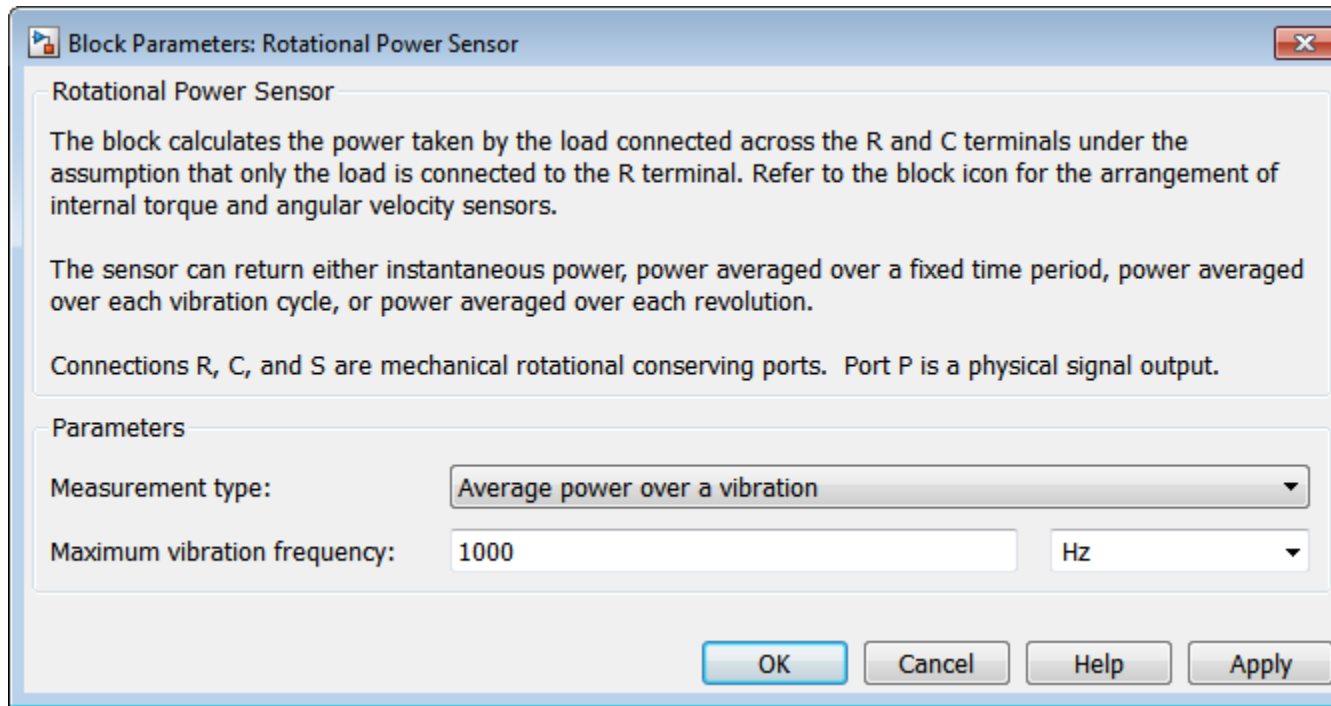
## Averaging period

Enter the time period used to compute the average power. Typical values include the cycle length of a rotating device.

The default value is 1. The default unit is s.

- Average power over a vibration — Output the rotational power averaged over one vibration period.

# Rotational Power Sensor



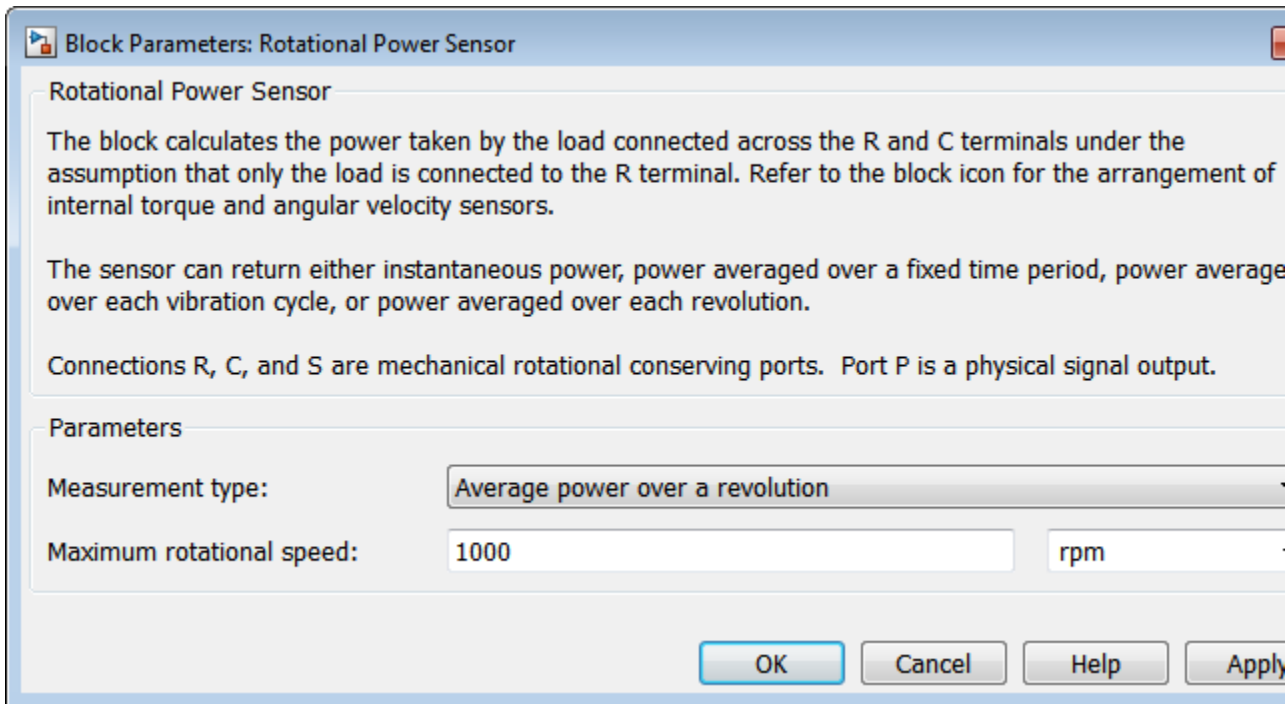
## Maximum vibration frequency

Enter the maximum vibration frequency over which the rotational power should be averaged. Very high values can reduce simulation speed. Very low values can result in lower power measurements.

The default value is 1000. The default unit is Hz.

- Average power over a revolution — Output the rotational power averaged over one revolution period.

# Rotational Power Sensor



## Maximum rotational speed

Enter the maximum speed over which the rotational power should be averaged. Very high values can reduce simulation speed. Very low values can result in lower power measurements.

The default value is 1000. The default unit is rpm.

## Ports

Port	Description
P	Physical signal output port for rotational power
S	Rotational conserving port associated with driving source

# Rotational Power Sensor

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Port	Description
C	Rotational conserving port associated with loaded driveline component(s)
R	Rotational conserving port associated with case holding loaded driveline component(s)

## See Also

Translational Power Sensor



**Purpose** Translational spring-damper system

**Library** Couplings & Drives

**Description** The block represents a spring-damper system commonly used to dampen vibration in mechanical systems. An internal force acts between ports R and C. This force is the sum of spring stiffness, damping, Coulomb friction, and hard-stop contributions. All force contributions are optional. The net internal force satisfies the expression:



$$F = F_{spr} + F_d + F_c + F_{HS}$$

where:

- $F$  — Net internal force
- $F_{spr}$  — Spring force,  $F_k = -k \ x$
- $F_d$  — Damping force,  $F_d = -b \ v$
- $F_c$  — Coulomb friction force,

$$F_c = \begin{cases} -F_k \ \text{sign}(v), & \text{when unlocked} \\ -F_s, & \text{when locked} \end{cases}$$

- $F_{HS}$  — Hard stop force,

$$F_{HS} = \begin{cases} -k_{HS} \ (x - UB) - b_{HS} \ v, & x \neq UB \\ 0 & UB \leq x \leq LB \\ -k_{HS} \ (x - LB) + b_{HS} \ v, & LB > x \end{cases}$$

- $k$  — Restoring spring stiffness
- $b$  — Viscous damping coefficient
- $x$  — Relative displacement between ports R and C
- $v$  — Relative velocity between R and C

# Shock Absorber

---

- $F_k$  — Kinetic friction force
- $F_s$  — Static friction force
- $k_{HS}$  — Hard stop contact stiffness coefficient
- $k_{HS}$  — Hard stop damping coefficient
- $UB$  — Hard stop upper bound
- $LB$  — Hard stop lower bound

The shock absorber block is based on the blocks listed in the table. For more information, see the corresponding block reference pages.

Block	Library
Loaded-Contact Translational Friction	SimDriveline Brakes & Detents/Translational
Translational Spring	Simscape Foundation Library/Mechanical/Translational
Translational Damper	
Translational Hard Stop	

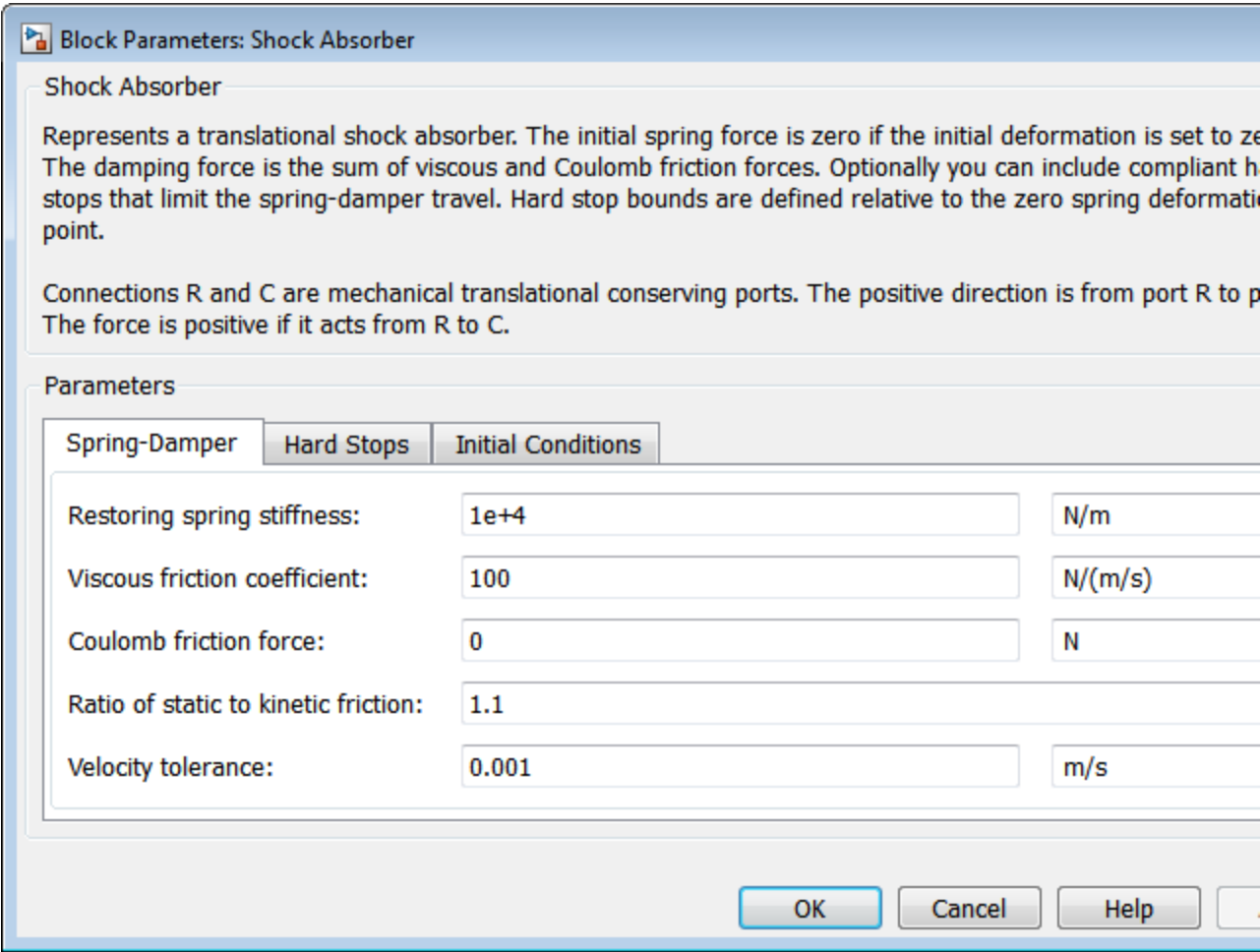
## Assumptions and Limitations

- The use of hard-stop and Coulomb friction enhances model fidelity, but reduces simulation speed. See “Improve Performance” in the product documentation.

## Dialog Box and Parameters

The dialog box contains three tabs, **Spring-Damper**, **Hard Stops**, and **Initial Conditions**.

### Spring-Damper



# Shock Absorber

---

## **Restoring spring stiffness**

Enter the value of the viscous spring stiffness constant,  $k$ . The value must be greater than or equal to zero.

The default value is 1e+4. the default unit is N/m.

## **Viscous friction coefficient**

Enter the value of the viscous damping constant,  $b$ . The value must be greater than or equal to zero.

The default value is 100. The default unit is m/s.

## **Coulomb friction force**

Enter the value of the Coulomb friction force. The value must be greater than or equal to zero. Setting the value to 0 eliminates the Coulomb friction force contribution. This enhances simulation speed, making the model more suitable for HIL testing.

The default value is 0. The default unit is N.

## **Ratio of static to kinetic friction**

Enter the value of the static/kinetic friction ratio ( $F_s/F_k$ ). The value must be greater than one.

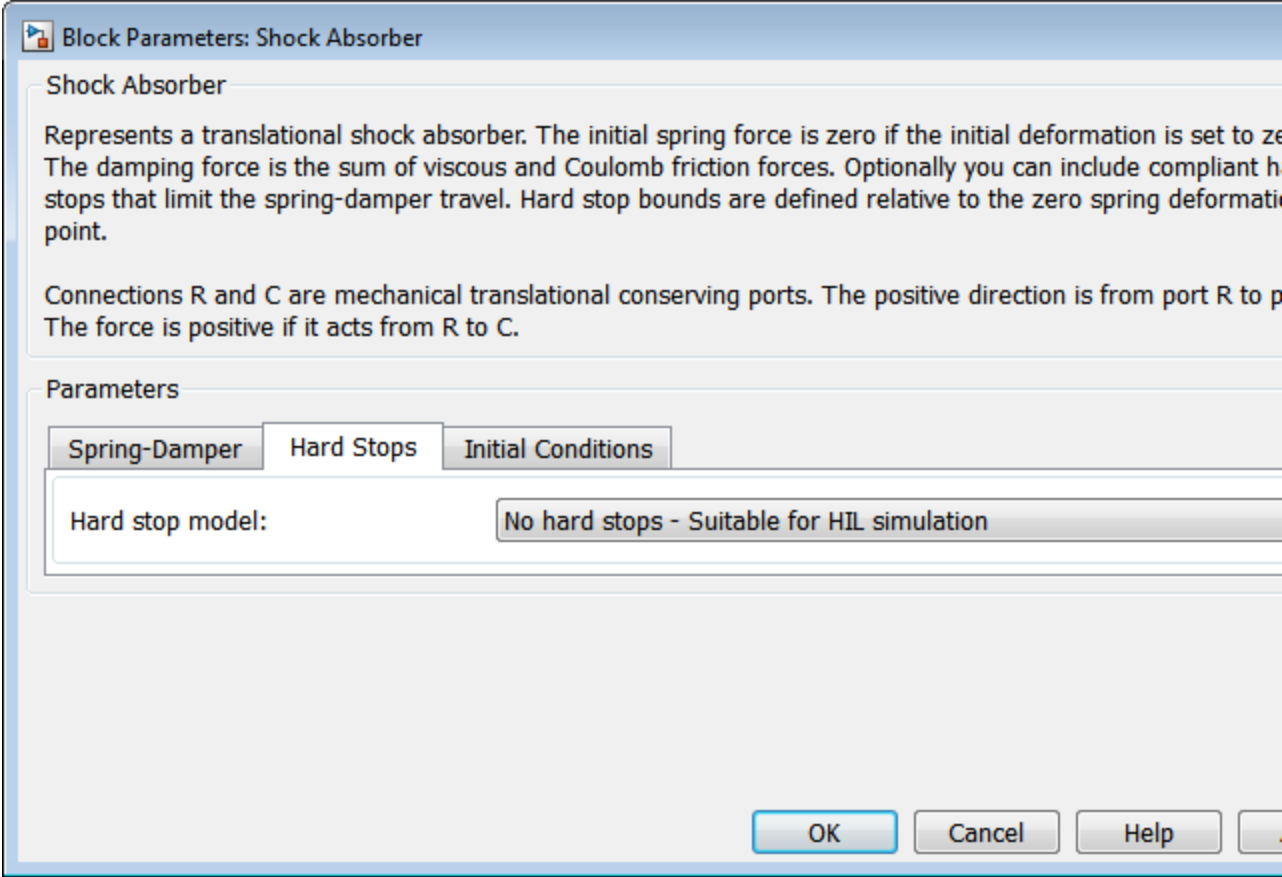
The default value is 1.1.

## **Velocity tolerance**

Enter the value of the relative velocity below which ports R and C lock and translate together. The value must greater than zero.

The default value is 0.001. The default unit is m/s.

## Hard Stops



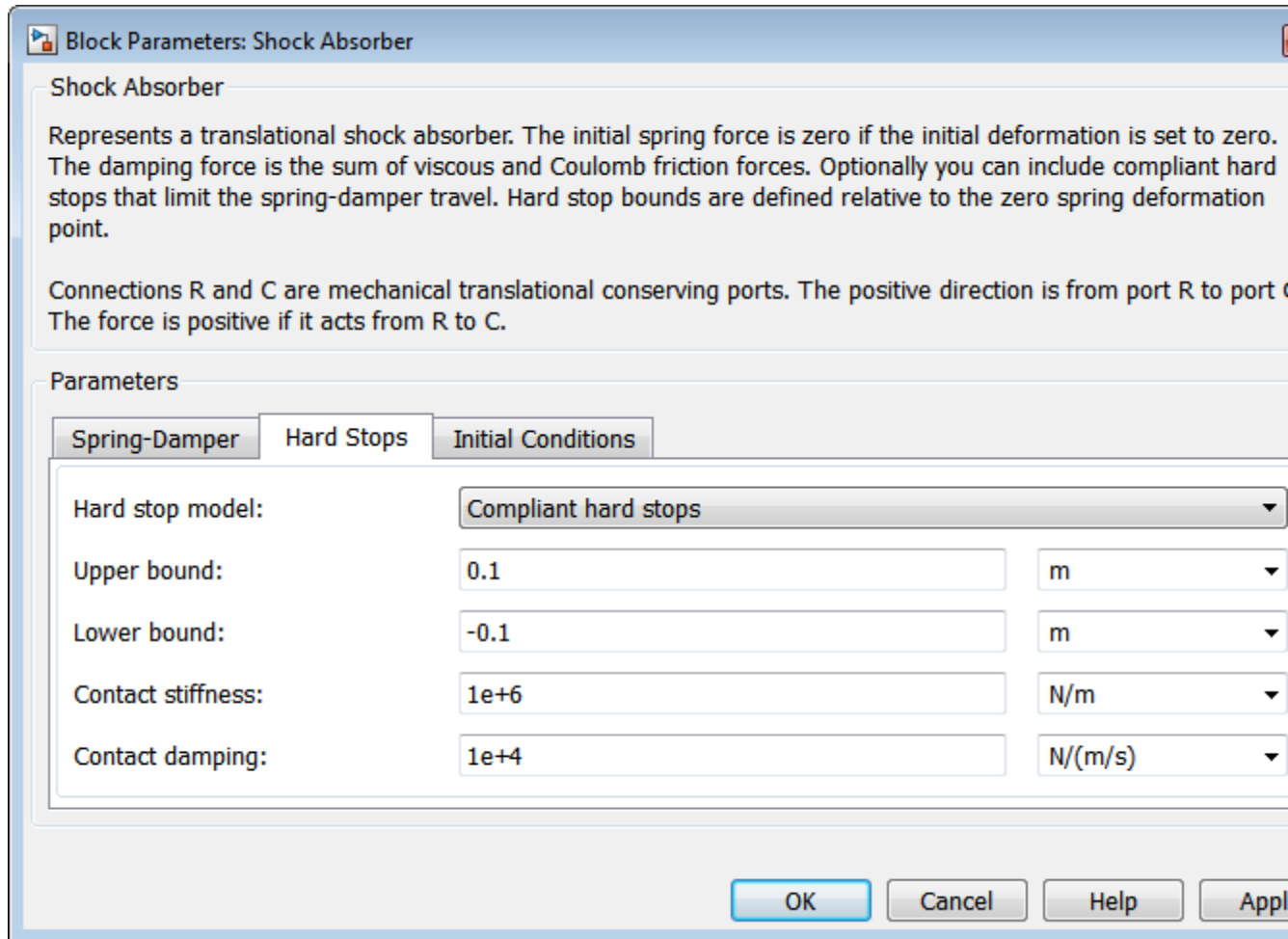
### Hard stop model

Select hard stop model. Selecting No hard-stops Suitable for HIL simulation eliminates the hard stop force contribution. This enhances simulation speed.

# Shock Absorber

The default setting is No hard stops Suitable for HIL simulation.

## Compliant hard stops



**Block Parameters: Shock Absorber**

**Shock Absorber**

Represents a translational shock absorber. The initial spring force is zero if the initial deformation is set to zero. The damping force is the sum of viscous and Coulomb friction forces. Optionally you can include compliant hard stops that limit the spring-damper travel. Hard stop bounds are defined relative to the zero spring deformation point.

Connections R and C are mechanical translational conserving ports. The positive direction is from port R to port C. The force is positive if it acts from R to C.

**Parameters**

Spring-Damper    **Hard Stops**    Initial Conditions

Hard stop model:	Compliant hard stops	
Upper bound:	0.1	m
Lower bound:	-0.1	m
Contact stiffness:	1e+6	N/m
Contact damping:	1e+4	N/(m/s)

OK    Cancel    Help    Appl

## Upper bound

Upper hard-stop position,  $UB$ . Positive displacement beyond the upper bound activates the hard stop contact force. The value must be greater than the lower bound.

The default value is 0.1. The default unit is m.

## Lower bound

Lower hard-stop position,  $LB$ . Negative displacement beyond the lower bound activates the hard stop contact force. The value must be smaller than the upper bound.

The default value is -0.1. The default unit is m.

## Contact stiffness

Enter the value of the hard stop stiffness constant,  $k_{HS}$ . The value must be greater than zero.

The default value is 1e6. The default unit is N/m.

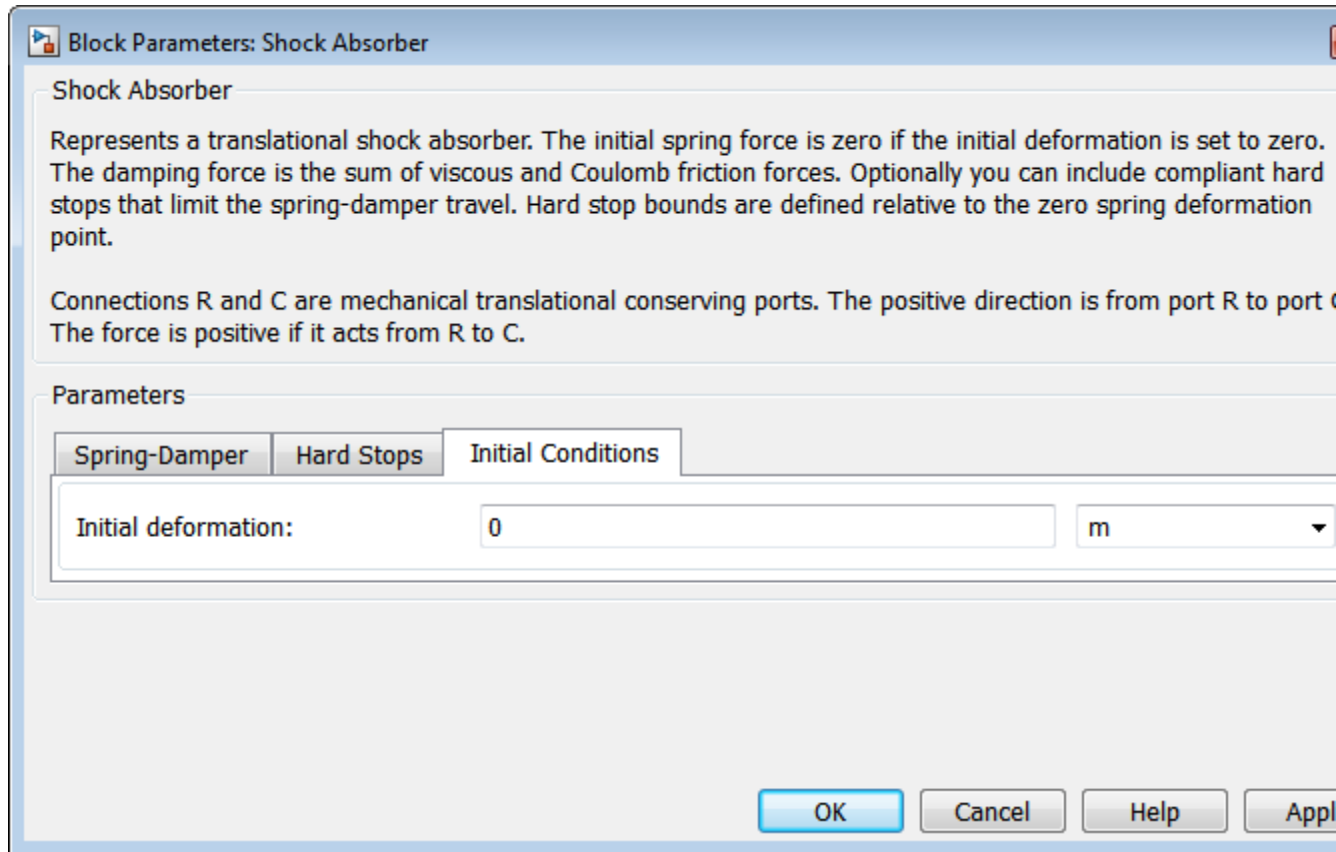
## Contact damping

Enter the value of the hard stop damping constant,  $b_{HS}$ . The value must be greater than zero.

The default value is 1e4. The default unit is N/(m/s).

# Shock Absorber

## Initial Conditions



### Initial deformation

Enter the initial value of the spring deformation.

The default value is 0. The default unit is m.



**Ports**

Port	Description
C	Translational conserving port
R	Translational Conserving port

**See Also**

Nonlinear Translational Spring, Variable Translational Spring, Nonlinear Translational Damper, Variable Translational Damper

# Simple Gear

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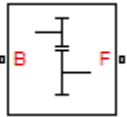
## Purpose

Simple gear of base and follower wheels with adjustable gear ratio and friction losses

## Library

Gears

## Description

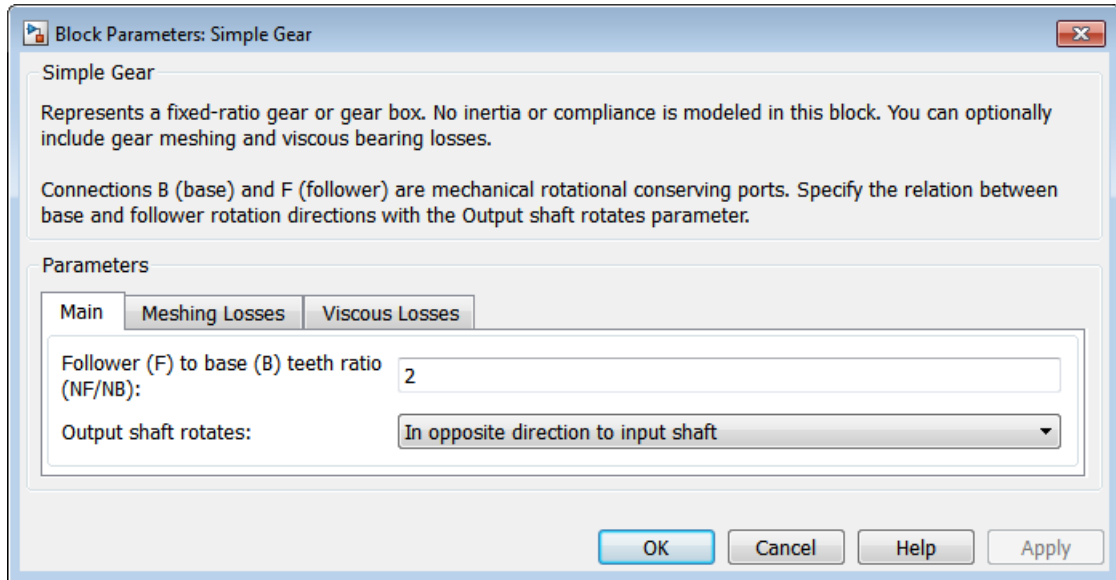


The Simple Gear block represents a gearbox that constrains the two connected driveline axes, base (B) and follower (F), to corotate with a fixed ratio that you specify. You can choose whether the follower axis rotates in the same or opposite direction as the base axis. If they rotate in the same direction,  $\omega_F$  and  $\omega_B$  have the same sign. If they rotate in opposite directions,  $\omega_F$  and  $\omega_B$  have opposite signs. For model details, see “Simple Gear Model” on page 1-296.

## Ports

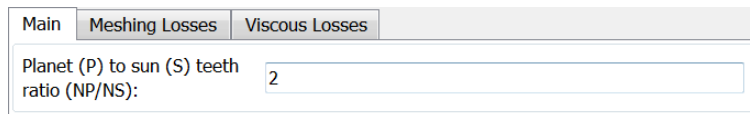
B and F are rotational conserving ports representing, respectively, the base and follower gear wheels.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



### Follower (F) to base (B) teeth ratio (NF/NB)

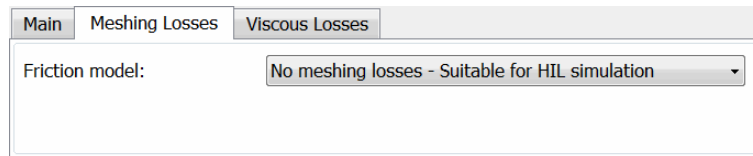
Fixed ratio  $g_{FB}$  of the follower axis to the base axis. The gear ratio must be strictly positive. The default is 2.

# Simple Gear

## Output shaft rotates

Choose the direction of motion of the follower (output) driveshaft relative to the motion of the base (input) driveshaft. The default is In opposite direction to input shaft.

## Meshing Losses



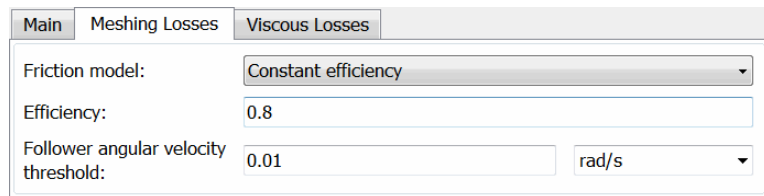
The screenshot shows a software interface with three tabs: 'Main', 'Meshing Losses', and 'Viscous Losses'. The 'Meshing Losses' tab is active. Below the tabs, there is a label 'Friction model:' followed by a dropdown menu. The dropdown menu is open, showing the selected option: 'No meshing losses - Suitable for HIL simulation'.

## Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

## Constant Efficiency



The screenshot shows the same software interface as above, but with the 'Constant efficiency' option selected in the 'Friction model' dropdown. Below the dropdown, there are three input fields: 'Efficiency:' with the value '0.8', 'Follower angular velocity threshold:' with the value '0.01', and a unit dropdown menu set to 'rad/s'.

## Efficiency

Torque transfer efficiency  $\eta$  for follower-base gear wheel pair meshing.

- **Load-dependent efficiency** — Transfer of torque between gear wheel pairs is reduced by variable efficiency  $0 < \eta < 1$ , a function of load across the gear. If you select this option, the panel changes from its default.

## Load-Dependent Efficiency

Main	Meshing Losses	Viscous Losses
Friction model:	Load-dependent efficiency	
Input shaft torque at no load:	0.1	N*m
Nominal output torque:	5	N*m
Efficiency at nominal output torque:	0.8	
Follower angular velocity threshold:	0.01	rad/s

### Input shaft torque at no load

Input idle stroke torque  $\tau_{idle}$  at zero load.

From the drop-down list, choose units. The default is newton-meters (N\*m).

### Nominal output torque

Output torque  $\tau_F$  at which to normalize load-dependent efficiency. Determines load-loss constant  $k$ .

From the drop-down list, choose units. The default is newton-meters (N\*m).

### Efficiency at nominal output torque

Torque transfer efficiency  $\eta$ , evaluated at the nominal output torque.

### Follower angular velocity threshold

Absolute angular velocity threshold  $\omega_{th}$  above which full efficiency loss is applied, for follower velocity  $\omega_F$ .

# Simple Gear

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses	
Viscous friction coefficients at base (B) and follower (F):			
[ 0 0 ]		N*m/(rad/s)	▼

### Viscous friction coefficients at base (B) and follower (F)

Vector of viscous friction coefficients  $[\mu_B \mu_F]$  for the base and follower gear wheels, respectively. The default is [0 0].

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Simple Gear Model

### Ideal Gear Constraint and Gear Ratio

Simple Gear imposes one kinematic constraint on the two connected axes:

$$r_F \omega_F = r_B \omega_B .$$

The follower-base gear ratio  $g_{FB} = r_F/r_B = N_F/N_B$ .  $N$  is the number of teeth on each gear. The two degrees of freedom reduce to one independent degree of freedom.

The torque transfer is:

$$g_{FB} \tau_B + \tau_F - \tau_{\text{loss}} = 0 ,$$

with  $\tau_{\text{loss}} = 0$  in the ideal case.

### Nonideal Gear Constraint and Losses

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . For general considerations on nonideal gear modeling, see “Model Gears with Losses”.

In a nonideal gear pair (B,F), the angular velocity, gear radii, and gear teeth constraints are unchanged. But the transferred torque and power are reduced by:

- Coulomb friction between teeth surfaces on gears B and F, characterized by efficiency  $\eta$
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients  $\mu$

$$\tau_{\text{loss}} = \tau_{\text{Coul}} \tanh(4\omega_{\text{out}}/\omega_{\text{th}}) + \mu\omega_{\text{out}}, \tau_{\text{Coul}} = |\tau_{\text{F}}| (1 - \eta).$$

The hyperbolic tangent regularizes the sign change in the Coulomb friction torque when the angular velocity changes sign.

<b>Power Flow</b>	<b>Power Loss Condition</b>	<b>Output Driveshaft</b>
Forward	$\omega_{\text{B}}\tau_{\text{B}} > \omega_{\text{F}}\tau_{\text{F}}$	<b>out</b> Follower, $\omega_{\text{F}}$
Reverse	$\omega_{\text{B}}\tau_{\text{B}} < \omega_{\text{F}}\tau_{\text{F}}$	Base, $\omega_{\text{B}}$

### Constant Efficiency

In the constant efficiency case,  $\eta$  is constant, independent of load or power transferred.

### Load-Dependent Efficiency

In the load-dependent efficiency case,  $\eta$  depends on the load or power transferred across the gears. For either power flow,  $\tau_{\text{Coul}} = g_{\text{FB}}\tau_{\text{idle}} + k\tau_{\text{F}}$ .  $k$  is a proportionality constant.  $\eta$  is related to  $\tau_{\text{Coul}}$  in the standard, preceding form but becomes dependent on load:

$$\eta = \tau_{\text{F}} / [g_{\text{FB}}\tau_{\text{idle}} + (k + 1)\tau_{\text{F}}].$$

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

# Simple Gear

---

## Examples

The `sdl_simple_gear` example model gives a basic example of a simple gear.

The `sdl_backlash` example model illustrates a gear with backlash.

The `sdl_gearbox_efficiency` example model measures the efficiency of a nonideal simple gear by comparing output to input power.

## See Also

Variable Ratio Transmission

“Represent and Transfer Driveline Motion and Torque”



# Simple Gear with Variable Efficiency

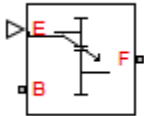
## Purpose

Gear set with parallel-axis rotation and variable meshing efficiency

## Library

Gears

## Description



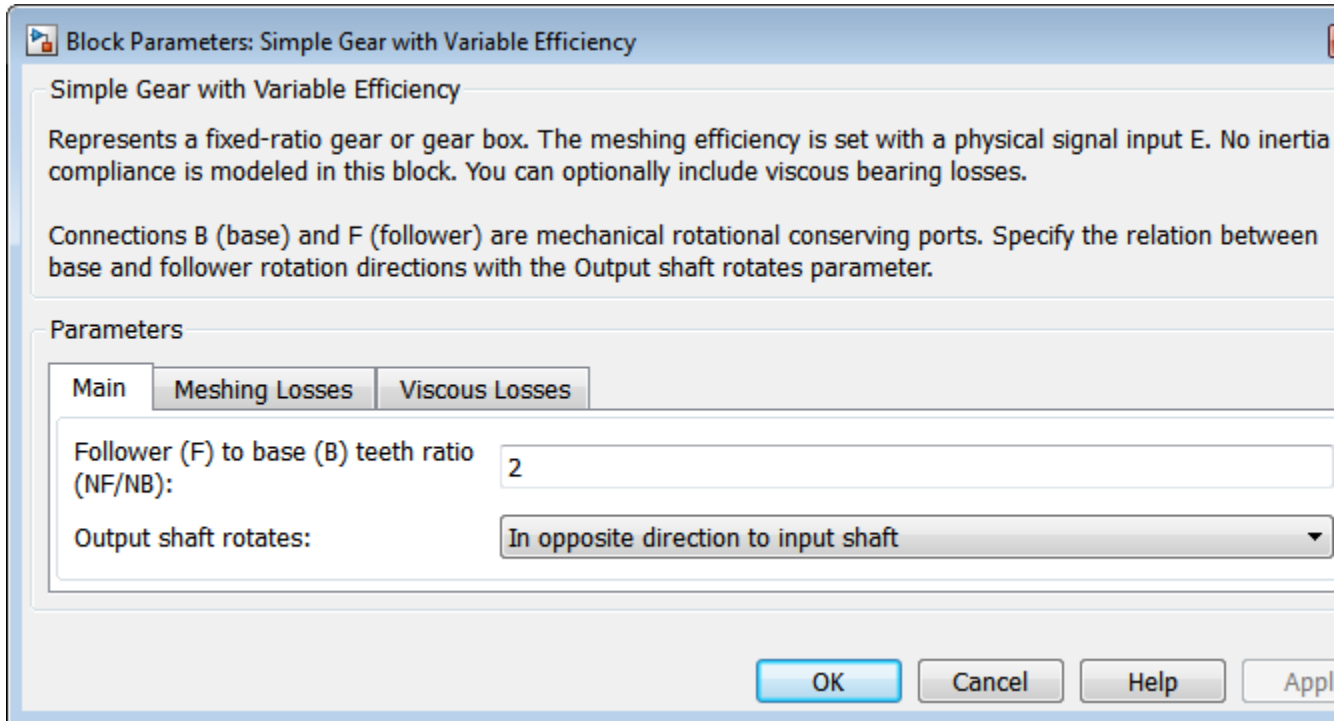
The block represents a simple gear train with variable meshing efficiency. The gear train transmits torque at a specified ratio between base and follower shafts arranged in a parallel configuration. Shaft rotation can occur in equal or opposite directions. Gear losses are optional. They include meshing and viscous bearing losses. To specify the variable meshing efficiency, the block contains a physical signal port that you can use to input a general time-varying signal. Inertia and compliance effects are ignored.

# Simple Gear with Variable Efficiency

## Dialog Box and Parameters

The dialog box contains three tabs, **Main**, **Meshing Losses**, and **Viscous Losses**.

### Main



### Follower (F) to base (B) teeth ratio (NF/NB)

Enter the gear ratio. This is the fraction of follower over base gear teeth numbers,  $NF/NB$ . The ratio must be positive. The default value is 2.

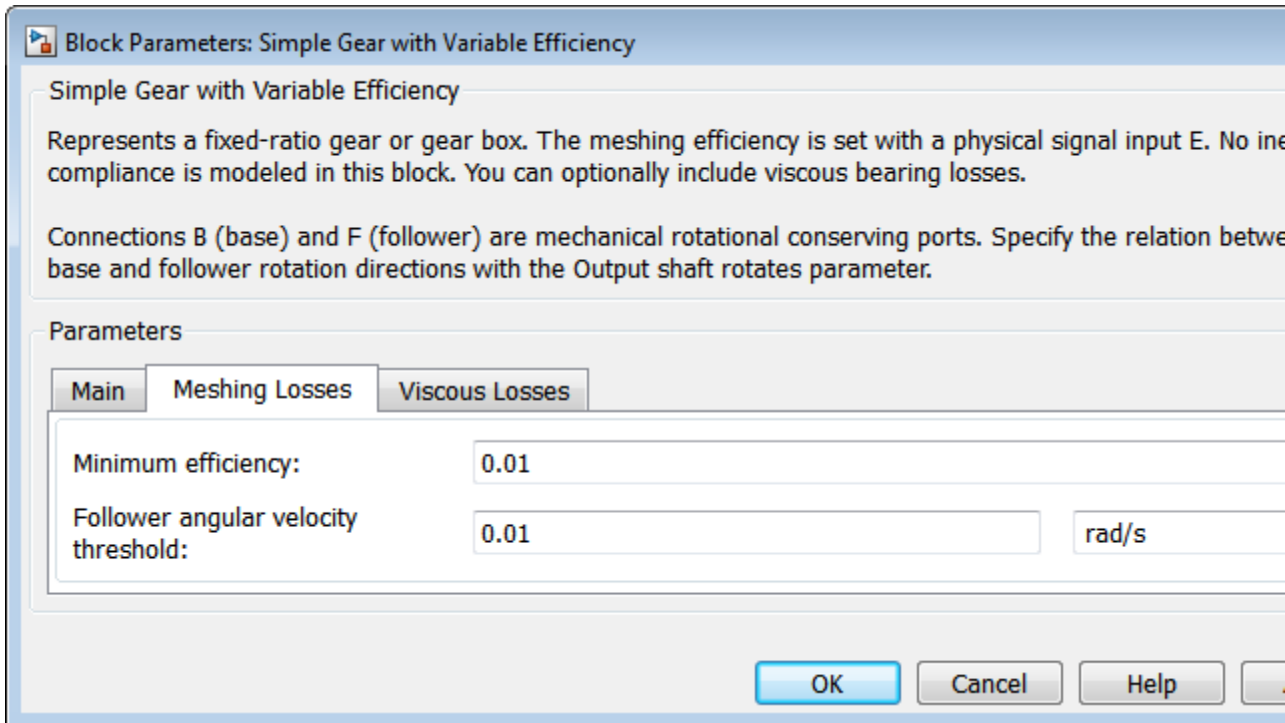
### Output shaft rotates

Select the relative rotation between shafts. This is the rotation direction of the output shaft with respect to the input shaft.

# Simple Gear with Variable Efficiency

Options include equal or opposite directions. The default setting is In opposite direction to input shaft.

## Meshing Losses



The screenshot shows a software dialog box titled "Block Parameters: Simple Gear with Variable Efficiency". The dialog has a description of the block and a "Parameters" section with three tabs: "Main", "Meshing Losses", and "Viscous Losses". The "Meshing Losses" tab is active, showing two input fields: "Minimum efficiency:" with a value of 0.01, and "Follower angular velocity threshold:" with a value of 0.01 and a unit of rad/s. At the bottom right, there are buttons for "OK", "Cancel", and "Help".

Block Parameters: Simple Gear with Variable Efficiency

Simple Gear with Variable Efficiency

Represents a fixed-ratio gear or gear box. The meshing efficiency is set with a physical signal input E. No in compliance is modeled in this block. You can optionally include viscous bearing losses.

Connections B (base) and F (follower) are mechanical rotational conserving ports. Specify the relation between base and follower rotation directions with the Output shaft rotates parameter.

Parameters

Main Meshing Losses Viscous Losses

Minimum efficiency: 0.01

Follower angular velocity threshold: 0.01 rad/s

OK Cancel Help

### Minimum Efficiency

Enter the smallest efficiency value allowed for the gear. The efficiency is the power ratio between output and input shafts. The physical signal input saturates for values below the minimum efficiency or above 1. The minimum efficiency must be positive. The default value is 0.01.

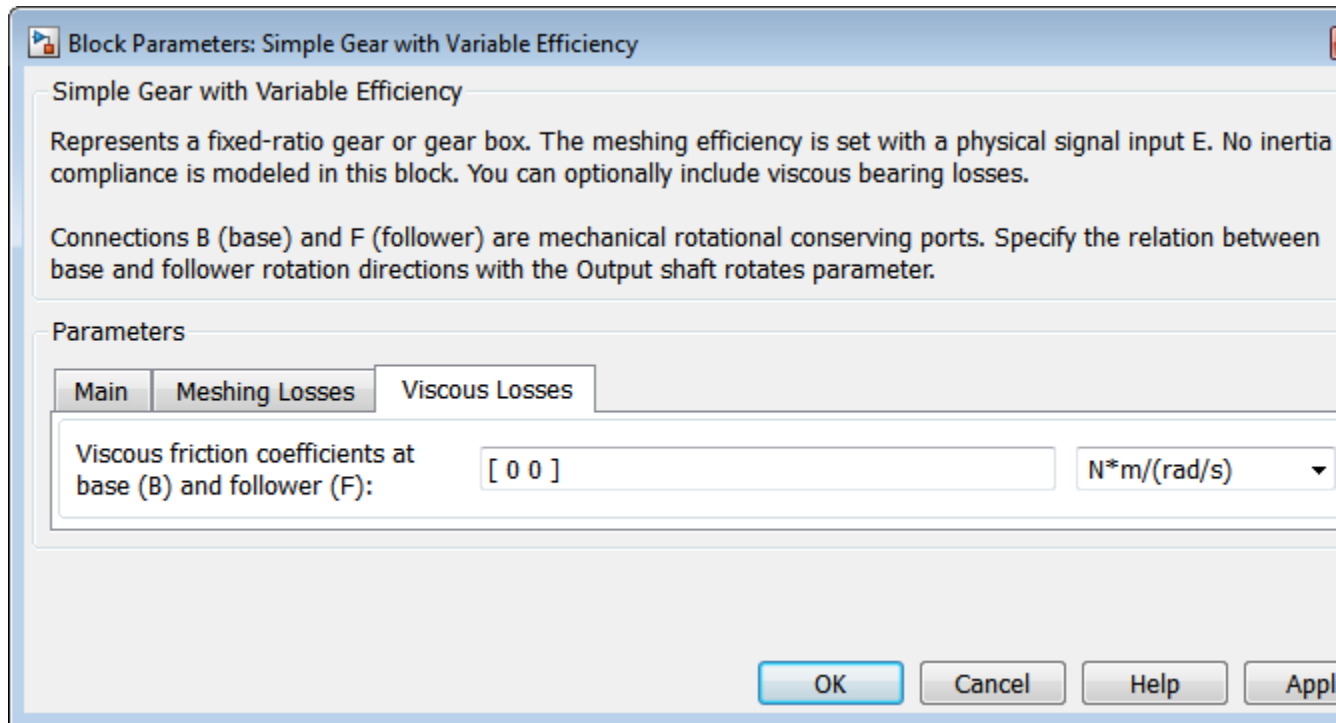
# Simple Gear with Variable Efficiency

## Follower angular velocity threshold

Enter the relative angular velocity above which full efficiency losses are included. Values below this threshold mark an efficiency transition region where the driving shaft becomes the driven shaft and vice-versa. The follower angular velocity threshold must be positive. Select a physical unit.

The default value is 0.01. The default unit is rad/s.

## Viscous Losses



## Simple Gear Model

### Viscous friction coefficients at base (B) and follower (F)

Enter a two-element vector with the viscous friction coefficients of the base and follower gears. Coefficients must be positive. The default vector is [0 0]. The default unit is N\*m/(rad/s).

### Ideal Gear Constraint and Gear Ratio

Simple Gear imposes one kinematic constraint on the two connected axes:

$$r_F \omega_F = r_B \omega_B .$$

The follower-base gear ratio  $g_{FB} = r_F/r_B = N_F/N_B$ .  $N$  is the number of teeth on each gear. The two degrees of freedom reduce to one independent degree of freedom.

The torque transfer is:

$$g_{FB} \tau_B + \tau_F - \tau_{\text{loss}} = 0 ,$$

with  $\tau_{\text{loss}} = 0$  in the ideal case.

### Nonideal Gear Constraint and Losses

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . For general considerations on nonideal gear modeling, see “Model Gears with Losses”.

In a nonideal gear pair (B,F), the angular velocity, gear radii, and gear teeth constraints are unchanged. But the transferred torque and power are reduced by:

- Coulomb friction between teeth surfaces on gears B and F, characterized by efficiency  $\eta$
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients  $\mu$

$$\tau_{\text{loss}} = \tau_{\text{Coul}} \tanh(4\omega_{\text{out}}/\omega_{\text{th}}) + \mu\omega_{\text{out}} , \tau_{\text{Coul}} = |\tau_F| (1 - \eta) .$$

# Simple Gear with Variable Efficiency

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The hyperbolic tangent regularizes the sign change in the Coulomb friction torque when the angular velocity changes sign.

<b>Power Flow</b>	<b>Power Loss Condition</b>	<b>Output Driveshaft out</b>
Forward	$\omega_B \tau_B > \omega_F \tau_F$	Follower, $\omega_F$
Reverse	$\omega_B \tau_B < \omega_F \tau_F$	Base, $\omega_B$

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## Ports

<b>Port</b>	<b>Description</b>
B	Rotational conserving port representing the base shaft
F	Rotational Conserving port representing the follower shaft

## See Also

Simple Gear

Variable Ratio Transmission

“Represent and Transfer Driveline Motion and Torque”

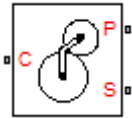
## Purpose

Planetary gear set of carrier, planet, and sun wheels with adjustable gear ratio and friction losses

## Library

Gears/Planetary Subcomponents

## Description



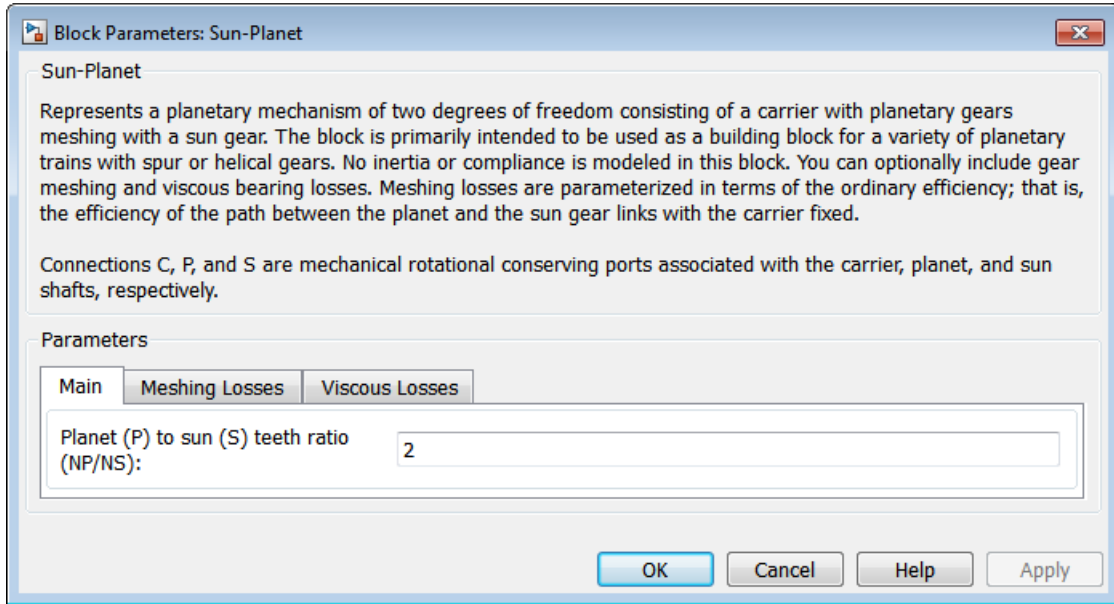
The Sun-Planet gear block represents a set of carrier, planet, and sun gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and sun corotate with a fixed gear ratio that you specify and in the same direction with respect to the carrier. A sun-planet and a ring-planet gear are basic elements of a planetary gear set. For model details, see “Sun-Planet Gear Model” on page 1-308.

## Ports

C, P, and S are rotational conserving ports representing, respectively, the carrier, planet, and sun gear wheels.

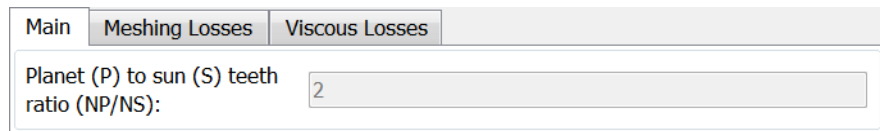
# Sun-Planet

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



### Ring (R) to sun (S) teeth ratio (NP/NS)

Ratio  $g_{RS}$  of the ring gear wheel radius to the sun gear wheel radius. This gear ratio must be strictly greater than 1. The default value is 2.



## Meshing Losses

Main	Meshing Losses	Viscous Losses
Friction model: <span style="border: 1px solid gray; padding: 2px;">No meshing losses - Suitable for HIL simulation</span>		

### Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

### Constant Efficiency

Main	Meshing Losses	Viscous Losses
Friction model: <span style="border: 1px solid gray; padding: 2px;">Constant efficiency</span>		
Ordinary efficiency: <input style="width: 100%;" type="text" value="0.98"/>		
Sun-carrier velocity threshold: <input style="width: 80%;" type="text" value="0.01"/> <span style="border: 1px solid gray; padding: 2px;">rad/s</span>		

### Ordinary efficiency

Torque transfer efficiency  $\eta_{SP}$  for sun-planet gear wheel pair meshing.

### Sun-carrier velocity threshold

Absolute angular velocity threshold  $\omega_{th}$  above which full efficiency loss is applied, for sun-carrier velocity  $\omega_S$ .

# Sun-Planet

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses	
Sun-carrier viscous friction coefficient:		<input type="text" value="0"/>	N*m/(rad/s) ▾

### Sun-carrier viscous friction coefficient

Viscous friction coefficient  $\mu_S$  for the sun-carrier gear motion. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Sun-Planet Gear Model

### Ideal Gear Constraints and Gear Ratios

Sun-Planet imposes one kinematic and one geometric constraint on the three connected axes:

$$r_C \omega_C = r_S \omega_S + r_P \omega_P, \quad r_C = r_P + r_S.$$

The planet-sun gear ratio  $g_{PS} = r_P/r_S = N_P/N_S$ .  $N$  is the number of teeth on each gear. In terms of this ratio, the key kinematic constraint is:

$$\omega_S = -g_{PS} \omega_P + (1 + g_{PS}) \omega_C.$$

The three degrees of freedom reduce to two independent degrees of freedom. The gear pair is (1,2) = (S,P).

### Warning

**The planet-sun gear ratio  $g_{PS}$  must be strictly greater than one.**

The torque transfer is:

$$g_{PS}\tau_S + \tau_P - \tau_{\text{loss}} = 0 ,$$

with  $\tau_{\text{loss}} = 0$  in the ideal case.

## **Nonideal Gear Constraints and Losses**

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . See “Model Gears with Losses”.

### **Limitations**

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

### **See Also**

Compound Planetary Gear, Planet-Planet, Planetary Gear, Ring-Planet, Sun-Planet Bevel, Sun-Planet Worm Gear

“Represent and Transfer Driveline Motion and Torque”

# Sun-Planet Bevel

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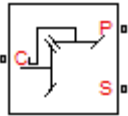
## Purpose

Planetary gear set of carrier, beveled planet, and sun wheels with adjustable gear ratio, assembly orientation, and friction losses

## Library

Gears/Planetary Subcomponents

## Description

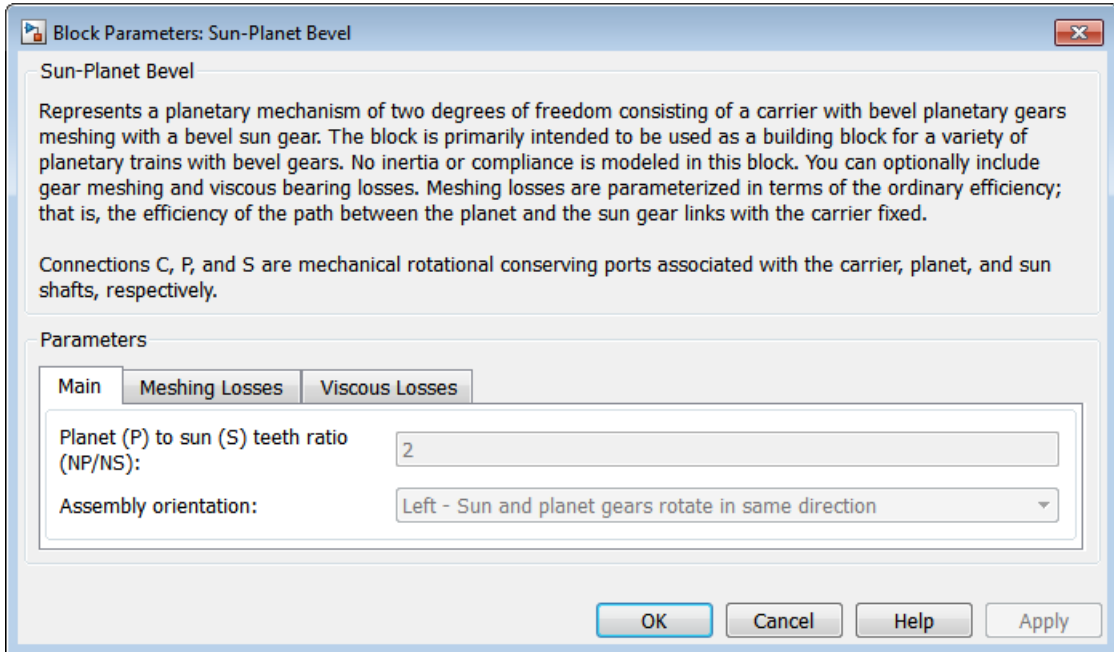


The Sun-Planet Bevel gear block represents a set of carrier, planet, and sun gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and sun corotate with a fixed gear ratio and in a direction that you specify. You control the direction by setting the assembly orientation, left or right. A sun-planet and a ring-planet gear are basic elements of a planetary gear set. For model details, see “Sun-Planet Bevel Gear Model” on page 1-314.

## Ports

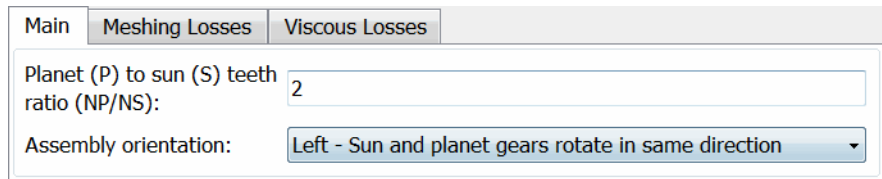
C, P, and S are rotational conserving ports representing, respectively, the carrier, planet, and sun gear wheels.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



# Sun-Planet Bevel

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## Planet (P) to sun (S) teeth ratio (NP/NS)

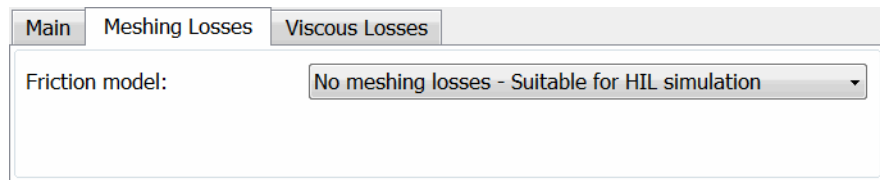
Ratio  $g_{PS}$  of the planet gear wheel radius to the sun gear wheel radius. This gear ratio must be strictly greater than 1. The default value is 2.

## Assembly orientation

Relative orientation of sun and planet gears, controlling their corotation direction. Left or right orientation imply, respectively, that the gears corotate in the same or opposite direction.

The default is Left Sun and planet gears rotate in same direction.

## Meshing Losses



Friction model: No meshing losses - Suitable for HIL simulation

## Friction model

Select how to implement friction losses from nonideal meshing of gear teeth. The default is No meshing losses.

- No meshing losses Suitable for HIL simulation — Gear meshing is ideal.
- Constant efficiency — Transfer of torque between gear wheel pairs is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

## Constant Efficiency

Main	Meshing Losses	Viscous Losses
Friction model: <input type="text" value="Constant efficiency"/>		
Ordinary efficiency: <input type="text" value="0.9"/>		
Sun-carrier velocity threshold: <input type="text" value="0.01"/> <input type="text" value="rad/s"/>		

### Ordinary efficiency

Torque transfer efficiency  $\eta_{SP}$  for sun-planet gear wheel pair meshing.

### Sun-carrier velocity threshold

Absolute angular velocity threshold  $\omega_{th}$  above which full efficiency loss is applied, for sun-carrier velocity  $\omega_S$ .

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Losses

Main	Meshing Losses	Viscous Losses
Sun-carrier viscous friction coefficient: <input type="text" value="0"/> <input type="text" value="N*m/(rad/s)"/>		

### Sun-carrier viscous friction coefficient

Viscous friction coefficient  $\mu_S$  for the sun-carrier gear motion. The default is 0.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

# Sun-Planet Bevel

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## Sun-Planet Bevel Gear Model

### Ideal Gear Constraints and Gear Ratios

Sun-Planet Bevel imposes one kinematic and one geometric constraint on the three connected axes:

$$r_C \omega_C = r_S \omega_S \pm r_P \omega_P, \quad r_C = r_S \pm r_P.$$

The planet-sun gear ratio  $g_{PS} = r_P/r_S = N_P/N_S$ .  $N$  is the number of teeth on each gear. In terms of this ratio, the key kinematic constraint is:

$$\omega_S = \mp g_{PS} \omega_P + (1 \pm g_{PS}) \omega_C.$$

The three degrees of freedom reduce to two independent degrees of freedom. The upper or lower sign applies, respectively, to left-oriented or right-oriented bevel assembly. The gear pair is (1,2) = (S,P).

### Warning

**The planet-sun gear ratio  $g_{PS}$  must be strictly greater than one.**

The torque transfer is:

$$g_{PS} \tau_S + \tau_P - \tau_{\text{loss}} = 0,$$

with  $\tau_{\text{loss}} = 0$  in the ideal case.

### Nonideal Gear Constraints and Losses

In the nonideal case,  $\tau_{\text{loss}} \neq 0$ . See “Model Gears with Losses”.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## Examples

These SimDriveline example models use the Sun-Planet Bevel gear to create custom gear sets:

- `sdl_limited_slip_differential`



- `sdl_custom_planetary_gear`

### **See Also**

Differential, Planet-Planet, Planetary Gear, Ring-Planet, Sun-Planet, Sun-Planet Worm Gear

“Represent and Transfer Driveline Motion and Torque”

# Sun-Planet Worm Gear

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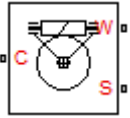
## Purpose

Planetary gear set of carrier, worm planet, and sun wheels with adjustable gear ratio, worm thread type, and friction losses

## Library

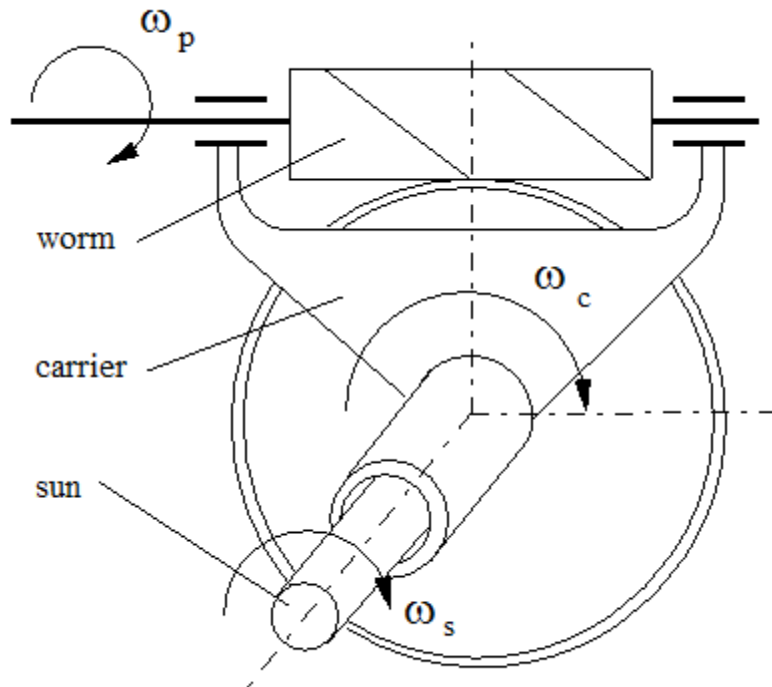
Gears/Planetary Subcomponents

## Description



The Sun-Planet Worm Gear block represents a two-degree-of-freedom planetary gear built from carrier, sun and planet gears. By type, the sun and planet gears are crossed helical spur gears arranged as a worm-gear transmission, in which the planet gear is a worm. Such transmissions are used in the Torsen type 1 differential. When transmitting power, the sun gear can be independently rotated by the worm (planet) gear, or by the carrier, or both.

# Sun-Planet Worm Gear



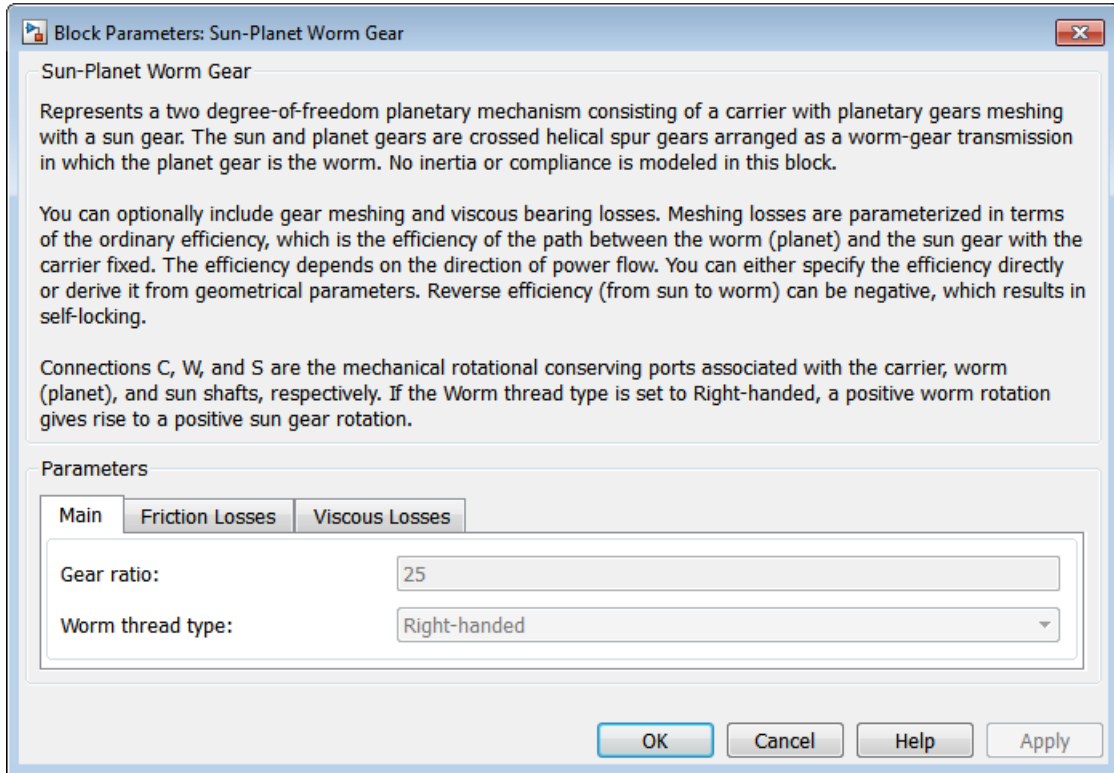
You specify a fixed gear ratio, which is determined as the ratio of the worm angular velocity to the sun gear angular velocity. You control the direction by setting the worm thread type, left-handed or right-handed. Rotation of the right-handed worm in positive direction causes the sun gear to rotate in positive direction too. The positive directions of the sun gear and the carrier are the same.

## Ports

C, W, and S are rotational conserving ports. They represent the carrier, worm (planet), and sun shafts, respectively.

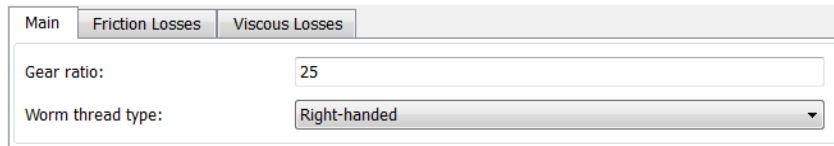
# Sun-Planet Worm Gear

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



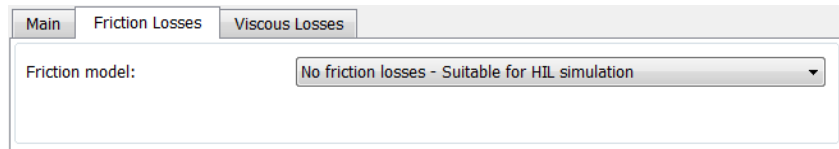
## Gear ratio

Gear or transmission ratio  $R_{WG}$  determined as the ratio of the worm angular velocity to the gear angular velocity. The default is 25.

## Worm thread type

Choose the directional sense of gear rotation corresponding to positive worm rotation. The default is **Right-handed**. If you select **Left-handed**, rotation of the worm in the generally-assigned positive direction results in the gear rotation in negative direction.

## Friction Losses



Friction model: No friction losses - Suitable for HIL simulation

## Friction model

Select how to implement friction losses from nonideal meshing of gear threads. The default is **No friction losses**.

- **No friction losses** Suitable for HIL simulation — Gear meshing is ideal.
- **Constant efficiency** — Transfer of torque and force between worm and gear is reduced by friction. If you select this option, the panel expands.

# Sun-Planet Worm Gear

## Constant Efficiency

Main	Friction Losses	Viscous Losses
Friction model:	Constant efficiency	
Friction parameterization:	Friction coefficient and geometrical parameters	
Normal pressure angle:	17.5	deg
Lead angle:	20	deg
Friction coefficient:	0.08	
Angular velocity threshold:	0.01	rad/s

### Friction parametrization

Select how to characterize the friction between gear threads. The default is Friction coefficient and geometrical parameters.

- Friction coefficient and geometrical parameters — Friction is determined by contact friction between surfaces.

#### Normal pressure angle

The thread pressure angle  $\alpha$  in the normal plane. Must be greater than zero and less than 90 degrees. The default is 17.5.

From the drop-down list, choose units. The default is degrees (deg).

#### Lead angle

Thread helix angle  $\lambda = \arctan[L/(\pi d)]$ .  $L$  is the worm lead,  $d$  is the worm pitch diameter. Must be greater than zero. The default is 20.

From the drop-down list, choose units. The default is degrees (deg).

## Friction coefficient

Dimensionless coefficient of normal friction in the thread. Must be greater than zero. The default is 0.08.

- **Efficiencies** — Friction is determined by constant efficiencies  $0 < \eta < 1$ . If you select this option, the panel changes from its default.

## Efficiencies

Parameter	Value
Friction model	Constant efficiency
Friction parameterization	Efficiencies
Worm-gear efficiency	0.74
Gear-worm efficiency	0.65
Angular velocity threshold	0.01
Angular velocity threshold units	rad/s

## Worm-gear efficiency

Efficiency  $\eta_{WG}$  of the power transfer from worm to gear.

## Gear-worm efficiency

Efficiency  $\eta_{GW}$  of the power transfer from gear to worm.

## Angular velocity threshold

Absolute angular velocity threshold above which full efficiency loss is applied. Must be greater than zero. The default is 0.01.

From the drop-down list, choose units. The default is radians/second (rad/s).

# Sun-Planet Worm Gear

## Viscous Losses

Worm-carrier and sun-carrier viscous friction coefficients: [ 0 0 ] N\*m/(rad/s)

### Worm-carrier and sun-carrier viscous friction coefficients

Vector of viscous friction coefficients  $[\mu_{WC} \mu_{SC}]$ , for the worm-carrier and sun-carrier shafts, respectively. The default is [ 0 0 ].

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Sun-Planet Worm Gear Model

### Model Variables

$R_{WG}$	Gear, or transmission, ratio determined as the ratio of the worm angular velocity to the gear angular velocity. The ratio is positive for the right-hand worm and negative for the left-hand worm.
$\omega_S$	Angular velocity of the sun gear
$\omega_P$	Planet (that is, worm) angular velocity
$\omega_C$	Carrier angular velocity
$\omega_{SC}$	Angular velocity of the sun with respect to the carrier
$a$	Normal pressure angle
$\lambda$	Worm lead angle
$L$	Worm lead
$d$	Worm pitch diameter
$\tau_S$	Torque applied to the sun shaft
$\tau_P$	Torque applied to the planet shaft



$\tau_C$	Torque applied to the carrier shaft
$\tau_{\text{loss}}$	Torque loss due to meshing friction. The loss depends on the device efficiency and the power flow direction. To avoid abrupt change of the friction torque at $\omega_S = 0$ , the friction torque is introduced via the hyperbolic function.
$\tau_{\text{instfr}}$	Instantaneous value of the friction torque added to the model to simulate friction losses
$\tau_{\text{fr}}$	Steady-state value of the friction torque
$k$	Friction coefficient
$\eta_{\text{WG}}$	Efficiency for worm-gear power transfer
$\eta_{\text{GW}}$	Efficiency for gear-worm power transfer
$\omega_{\text{th}}$	Absolute angular velocity threshold
$\mu_{\text{SC}}$	Sun-carrier viscous friction coefficient
$\mu_{\text{WC}}$	Worm-carrier viscous friction coefficient

## Ideal Gear Constraints and Gear Ratio

Sun-planet worm gear imposes one kinematic constraint on the three connected axes:

$$\omega_S = \omega_P/R_{\text{WG}} + \omega_C .$$

The gear has two independent degrees of freedom. The gear pair is  $(1,2) = (S,P)$ .

The torque transfer is:

$$R_{\text{WG}}\tau_P + \tau_S - \tau_{\text{loss}} = 0 ,$$

$$\tau_C = - \tau_S,$$

with  $\tau_{\text{loss}} = 0$  in the ideal case.

# Sun-Planet Worm Gear

## Nonideal Gear Constraints

In a nonideal gear, the angular velocity and geometric constraints are unchanged. But the transferred torque and power are reduced by:

- Coulomb friction between thread surfaces on W and G, characterized by friction coefficient  $k$  or constant efficiencies  $[\eta_{WG}, \eta_{GW}]$
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients  $\mu_{SC}$  and  $\mu_{WC}$

The torque transfer for nonideal gear has the general form:

$$\tau_S = -R_{WG}(\tau_P - \mu_{WC}\omega_P) + \tau_{\text{instfr}} ,$$

$$\tau_{\text{instfr}} = \tau_{\text{fr}} \tanh(4\omega_{SC}/\omega_{\text{th}}) + \mu_{SC}\omega_{SC} .$$

The hyperbolic tangent regularizes the sign change in the friction torque when the sun gear velocity changes sign.

Condition	Friction Torque $\tau_{\text{fr}}$
$\omega_{SC}\tau_S > 0$	$ \tau_S  \cdot (1 - \eta_{GW})$
$\omega_{SC}\tau_S \leq 0$	$ \tau_S  \cdot (1 - \eta_{WG})/\eta_{WG}$

Because the transmission incorporates a worm gear, the efficiencies are different for the direct and reverse power transfer. The following table shows the value of the efficiency for all combinations of the power transfer.

Driving shaft	Driven shaft		
	Planet	Sun	Carrier
Planet	n/a	$\eta_{WG}$	$\eta_{WG}$
Sun	$\eta_{GW}$	n/a	No loss
Carrier	$\eta_{GW}$	No loss	n/a

## Geometric Surface Contact Friction

In the contact friction case,  $\eta_{WG}$  and  $\eta_{GW}$  are determined by:

- The worm-gear threading geometry, specified by lead angle  $\lambda$  and normal pressure angle  $a$ .
- The surface contact friction coefficient  $k$ .

$$\eta_{WG} = (\cos a - k \tan \lambda) / (\cos a + k / \tan \lambda) ,$$

$$\eta_{GW} = (\cos a - k / \tan \lambda) / (\cos a + k \tan a) .$$

## Constant Efficiencies

In the constant efficiency case, you specify  $\eta_{WG}$  and  $\eta_{GW}$ , independently of geometric details.

## Self-Locking and Negative Efficiency

If you set efficiency for the reverse power flow to a negative value, the train exhibits *self-locking*. Power can not be transmitted from sun gear to worm and from carrier to worm unless some torque is applied to the worm to release the train. In this case, the absolute value of the efficiency specifies the ratio at which the train is released. The smaller the train lead angle, the smaller the reverse efficiency.

## Meshing Efficiency

The efficiencies  $\eta$  of meshing between worm and gear are fully active only if the absolute value of the gear angular velocity is greater than the velocity tolerance.

If the velocity is less than the tolerance, the actual efficiency is automatically regularized to unity at zero velocity.

## Viscous Friction Force

The viscous friction coefficients of the worm-carrier and sun-carrier bearings control the viscous friction torque experienced by the carrier from lubricated, nonideal gear threads. For details, see the “Nonideal Gear Constraints” on page 1-324 section.

# Sun-Planet Worm Gear

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## **Limitations**

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## **Example**

The `sdl_torsen_differential` example model uses a Sun-Planet Worm gear to model a Torsen differential gear set.

## **See Also**

Sun-Planet, Sun-Planet Bevel, Worm Gear

“Represent and Transfer Driveline Motion and Torque”

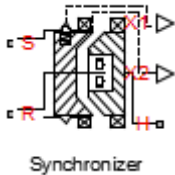
## Purpose

Cone clutch, dog clutch, and translational detent assembled to provide smooth gear engagement

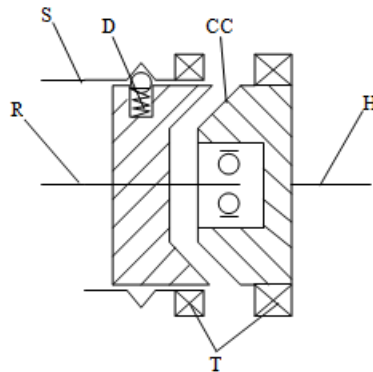
## Library

Clutches

## Description

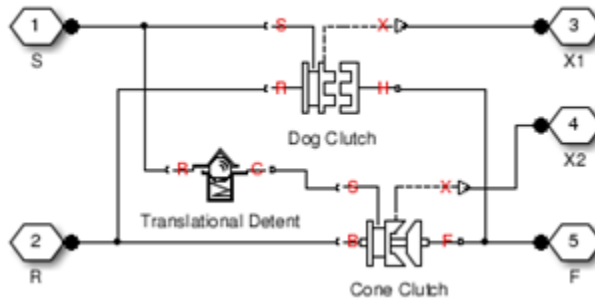


The block represents a synchronizer that contains a dog clutch, a cone clutch, and a translational detent. The shift linkage first translates to engage the cone clutch. Frictional torque causes the shift linkage and cone clutch shaft to rotate at equal speed. When the force acting on the shift linkage exceeds the detent force, the dog clutch can engage.



The schematic illustrates a synchronizer in the disengaged state. In this state, the ring (R) and hub (H) shafts can spin independently at different speeds. To synchronize ring and hub shaft speeds, the shift linkage (S) translates toward the hub shaft to engage the cone clutch. The friction surfaces of the cone clutch produce a frictional torque that equalizes the rotational speeds of the ring and hub shafts. The dog clutch teeth (T) can engage when the translational force acting on the shift linkage exceeds the peak detent force. The peak detent force should allow sufficient time and normal force to equalize ring and hub shaft speeds so that the dog clutch can engage.

# Synchronizer



The model implements Dog Clutch, Translational Detent and a modified version of Cone Clutch blocks. Refer to each block reference page for more information on the corresponding block function. You can use a similar approach to model customized versions of the synchronizer. One example is the Gearbox (Detailed) subsystem in the model `sdl_manual_transmission`.

Connections R and H are mechanical rotational conserving ports that represent the ring (R) and hub (H), respectively. Connection S is a mechanical translational conserving port that represents the ring shifter handle.

Connections X1 and X2 are physical signal outputs that report the shift linkage positions of the dog clutch and cone clutch, respectively. The shift linkage positions are zero when the clutch is fully disengaged. When the dog clutch is fully engaged, the dog clutch shift linkage position has a magnitude equal to the sum of the dog clutch ring-hub gap and the tooth height. When cone clutch is fully engaged, the cone clutch shift linkage position has a magnitude equal to the cone clutch ring-hub gap.

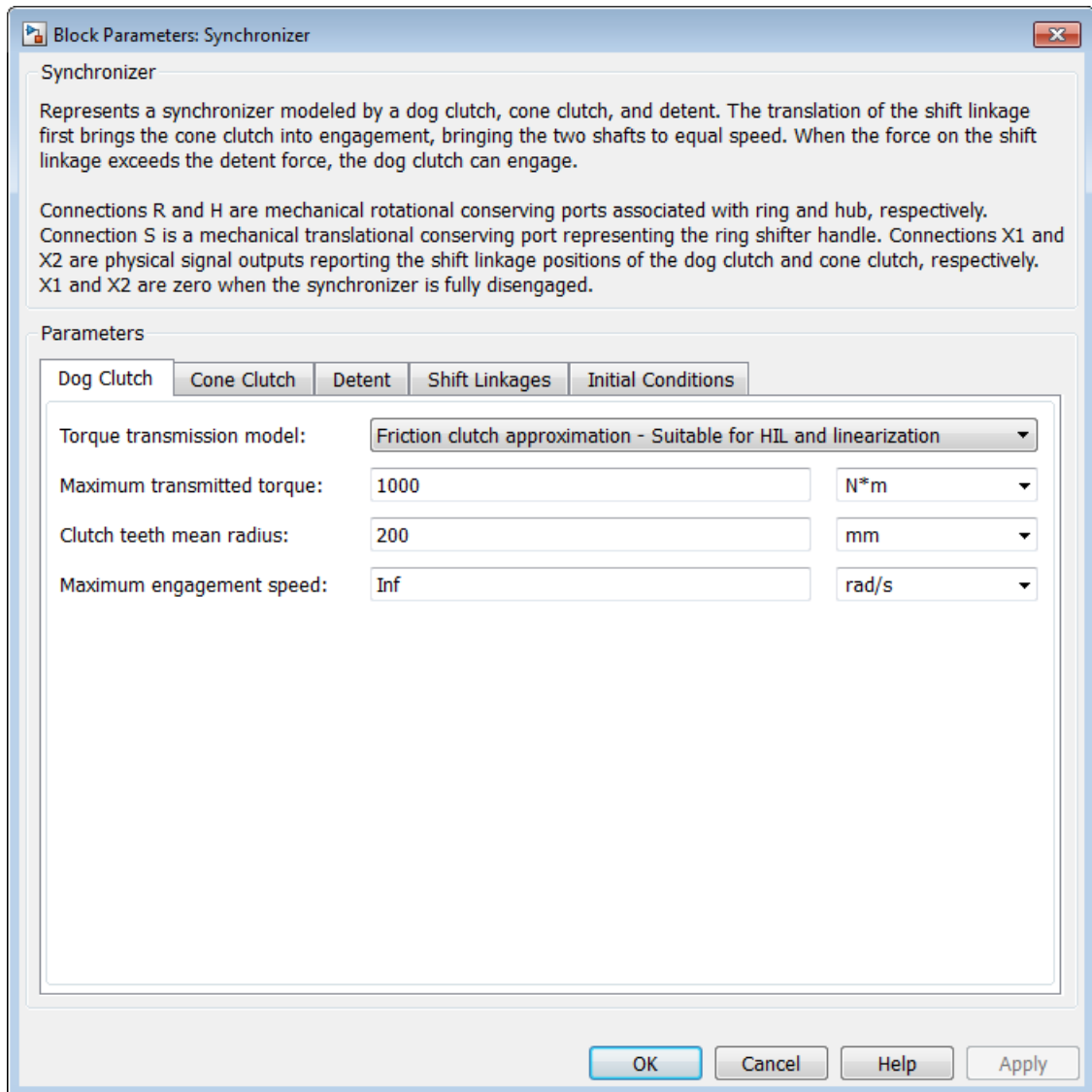
## Assumptions and Limitations

- The model does not account for inertia effects. You can add a Simscape Inertia block at each port to add inertia to the synchronizer model.

## Dialog Box and Parameters

The dialog box contains five tabs. Use the tabs to specify the parameters for the synchronizer components and initial conditions.

### Dog Clutch



## **Torque Transmission Model**

Model used to represent torque transmission. The default setting is Friction clutch approximation - Suitable for HIL and linearization.

Each model provides the following common parameters:

### **Clutch teeth mean radius**

Mean radius of the clutch teeth. The parameter must be greater than zero. The default value is 200 mm.

### **Maximum engagement speed**

The relative angular speed between the ring and hub shafts above which the dog clutch cannot engage. The value is specific to the specific gearbox or transmission and must be minimized to avoid high dynamic impact during engagement. The default value is `inf` rad/s

## **Friction clutch approximation - Suitable for HIL and linearization**

### **Maximum transmitted torque**

Maximum expected torque across the clutch. The default value is 1000 Nm.

## **Dynamic with Backlash**

### **Number of teeth**

Number of teeth in the clutch. The parameter must be greater than or equal to one. The default value is 6.

### **Rotational backlash**

Angular play of the ring with respect to the hub. Greater backlash values correspond to wider tooth gaps that makes engagement easier. The parameter must be greater than zero. The default value is 10 deg.



**Torsional stiffness**

Stiffness of the tooth contact. The model assumes deformation due to tooth interaction is elastic. The angular deformation at the contact surface is proportional to the applied torque and the reciprocal of the contact stiffness. The parameter must be greater than zero. The default value is  $1e+7$  N\*m/rad.

**Torsional damping**

Damping at the tooth contact surface. The value of the parameter is inversely proportional to the number of oscillations following impact. The parameter must be greater than zero. The default value is  $2$  N\*m/(rad/s).

**Tooth-tooth friction coefficient**

Friction coefficient for the translational relative motion between the hub and ring teeth. The value of the parameter depends on the tooth material, lubrication, and quality of contacting surfaces. The parameter must be greater than zero. The default value is  $0.05$ .

# Synchronizer

## Cone Clutch

**Block Parameters: Synchronizer**

Synchronizer

Represents a synchronizer modeled by a dog clutch, cone clutch, and detent. The translation of the shift linkage first brings the cone clutch into engagement, bringing the two shafts to equal speed. When the force on the shift linkage exceeds the detent force, the dog clutch can engage.

Connections R and H are mechanical rotational conserving ports associated with ring and hub, respectively. Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutch and cone clutch, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

Parameters

**Dog Clutch** | **Cone Clutch** | Detent | Shift Linkages | Initial Conditions

Contact surface maximum diameter:	<input type="text" value="150"/>	<input type="text" value="mm"/>
Contact surface minimum diameter:	<input type="text" value="100"/>	<input type="text" value="mm"/>
Cone half angle:	<input type="text" value="12"/>	<input type="text" value="deg"/>
Kinetic friction coefficient:	<input type="text" value="0.3"/>	
Static friction coefficient:	<input type="text" value="0.35"/>	
Velocity tolerance:	<input type="text" value="0.001"/>	<input type="text" value="rad/s"/>
Threshold force:	<input type="text" value="1"/>	<input type="text" value="N"/>

OK Cancel Help Apply

**Contact surface maximum diameter**

Outer diameter of the friction material lining that lines the cone clutch disks. The parameter must be greater than zero. The default value is 150 mm.

**Contact surface minimum diameter**

Inner diameter of the friction material lining that lines the cone clutch disks. The parameter must be greater than zero, but smaller than the value of **Contact surface maximum diameter**.

**Cone half angle**

Half angle of the clutch cone. The parameter must be greater than zero and less than or equal to 90°. The default value is 12 deg.

**Kinetic friction coefficient**

Kinetic friction coefficient between the friction surfaces of the cone clutch disks. The parameter must be greater than zero. The default value is 0.3.

**Static friction coefficient**

Static friction coefficient between the friction surfaces of the cone clutch disks. The parameter must be greater than the kinetic friction coefficient. The default value is 0.35.

**Velocity tolerance**

Relative velocity between the cone clutch disks below which the disk friction surfaces lock. Surface locking requires that the torque across the R and H rotational ports be smaller than the product of the effective radius, the static friction coefficient, and the applied normal force. The default value is 0.001 rad/s.

**Threshold force**

Minimum value of the applied normal force for frictional torque transmittal to occur. If the value of the parameter exceeds the value of the applied normal force, no frictional torque is transmitted.

# Synchronizer

## Detent

**Block Parameters: Synchronizer**

Synchronizer

Represents a synchronizer modeled by a dog clutch, cone clutch, and detent. The translation of the shift linkage first brings the cone clutch into engagement, bringing the two shafts to equal speed. When the force on the shift linkage exceeds the detent force, the dog clutch can engage.

Connections R and H are mechanical rotational conserving ports associated with ring and hub, respectively. Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutch and cone clutch, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

Parameters

Dog Clutch   Cone Clutch   **Detent**   Shift Linkages   Initial Conditions

Peak force:	<input type="text" value="500"/>	<input type="text" value="N"/>
Notch width:	<input type="text" value="3"/>	<input type="text" value="mm"/>
Viscous friction coefficient:	<input type="text" value="0.1"/>	<input type="text" value="N/(m/s)"/>
Friction to peak force ratio:	<input type="text" value="0.01"/>	
Friction velocity threshold:	<input type="text" value="0.05"/>	<input type="text" value="m/s"/>

OK   Cancel   Help   Apply

**Peak force**

Peak shear force of the detent. The default value is 500 N.

**Notch width**

Width of the region where the detent exhibits shear force. The default value is 3 mm.

**Viscous friction coefficient**

Viscous friction coefficient at the contact surface of the detent. The parameter must be greater than or equal to zero. The default value is 0.1 N/(m/s).

**Friction to peak force ratio**

Ratio of the kinetic friction to the peak shear force of the detent. The parameter is used to set the value of the kinetic friction. The parameter must be greater than or equal to zero. The default value is 0.01.

**Friction velocity threshold**

Velocity required for peak kinetic friction at the contact surface of the detent. The parameter ensures the force is continuous when the travel direction changes, increasing the numerical stability of the simulation. The parameter must be greater than zero. The default value is 0.05 m/s.

# Synchronizer

## Shift Linkages

**Block Parameters: Synchronizer**

**Synchronizer**

Represents a synchronizer modeled by a dog clutch, cone clutch, and detent. The translation of the shift linkage first brings the cone clutch into engagement, bringing the two shafts to equal speed. When the force on the shift linkage exceeds the detent force, the dog clutch can engage.

Connections R and H are mechanical rotational conserving ports associated with ring and hub, respectively. Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog clutch and cone clutch, respectively. X1 and X2 are zero when the synchronizer is fully disengaged.

**Parameters**

Dog Clutch	Cone Clutch	Detent	Shift Linkages	Initial Conditions
Linkage travel direction: Positive shift linkage displacement engages clutch				
Ring-hub clearance when dog clutch disengaged: 5 mm				
Tooth overlap to engage: 3 mm				
Tooth height: 10 mm				
Ring-hub clearance when cone clutch disengaged: 3 mm				
Hard stop at back of shift linkage: Hard stop when fully disengaged				
Dog clutch ring stop stiffness: 1e+6 N/m				
Cone clutch ring stop stiffness: 1e+6 N/m				
Dog clutch ring stop damping: 1000 N/(m/s)				
Cone clutch ring stop damping: 1000 N/(m/s)				
Viscous friction coefficient: 100 N/(m/s)				

OK Cancel Help Apply

## **Linkage travel direction**

Direction the shift linkage must translate along in order to engage the clutch. The default setting is Positive shift linkage displacement engages clutch

## **Ring-hub clearance when dog clutch disengaged**

Distance between ring and dog clutch hub with the ring fully retracted. The parameter specifies the maximum travel distance if engagement requirements are not met. The parameter must be greater than the value of **Ring-hub clearance when cone clutch disengaged**. The default value is 5 mm.

## **Tooth overlap to engage**

Minimum value of the tooth overlap beyond which the dog clutch is considered engaged. The parameter must be greater than zero. The default value is 3 mm.

## **Tooth height**

Height of dog clutch teeth. The parameter must be greater than zero. The default value is 10 mm.

## **Ring-hub clearance when cone clutch disengaged**

Distance between the ring and cone clutch hub when the ring is fully retracted. The cone clutch can begin to engage once the slider has translated this distance. The ring can travel only this distance if the engagement requirements are not satisfied. The parameter must be greater than zero. The default value is 3 mm.

## **Hard stop at back of shift linkage**

Hard stop that prevents the shift linkage from traveling beyond the fully disengaged position. Choices include:

- No hard stop when fully disengaged
- Hard stop when fully disengaged

The default setting is Hard stop when fully disengaged.

## **Dog clutch ring stop stiffness**

Stiffness of the hard stops on both sides of the dog clutch ring.

The model assumes the ring and stops behave elastically. Contact deformation is proportional to the applied force and the reciprocal of the contact stiffness. The value of the stiffness must be assigned with reference to the parameter **Tooth overlap to engage**. Too low a stiffness could cause the deformation to exceed the required overlap and initiate a false engagement. The parameter must be greater than zero. The default value is  $1e+6$  N/m.

## **Cone clutch ring stop stiffness**

Stiffness of the hard stops on both sides of the cone clutch ring.

The model assumes the ring and stops behave elastically. Contact deformation is proportional to the applied force and the reciprocal of the contact stiffness. The parameter must be greater than zero. The default value is  $1e+6$  N/m.

## **Dog clutch ring stop damping**

Translational contact damping between the dog clutch ring and the hub. The value of the damping is inversely proportional to the number of oscillations that occur after impact. The parameter must be greater than zero. The default value is  $1e+3$  N/(m/s).

## **Cone clutch ring stop damping**

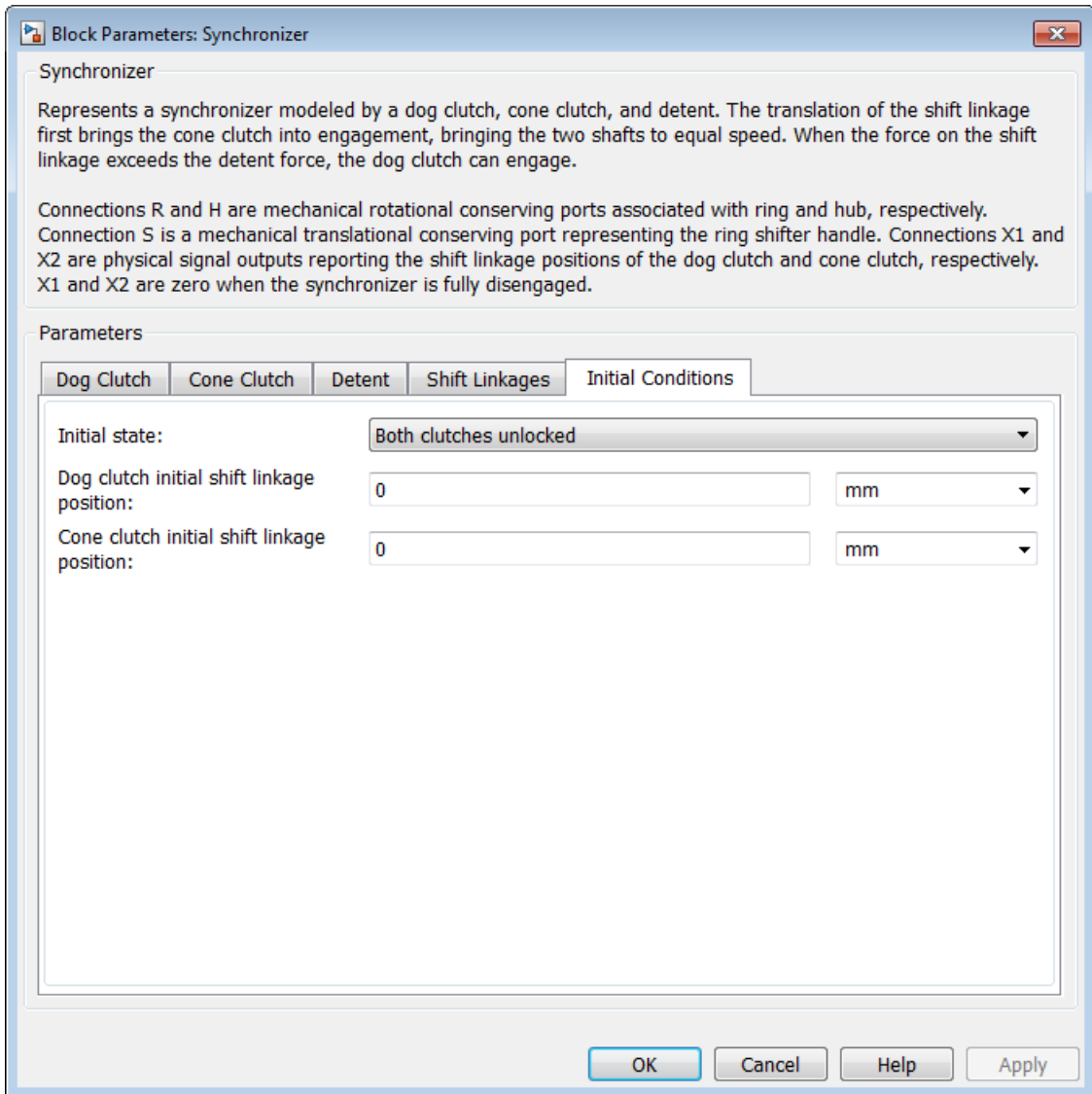
Translational contact damping between the cone clutch ring and the hub. The value of damping is inversely proportional to the number of oscillations that occur after impact. The parameter must be greater than zero. The default value is  $1e+3$  N/(m/s).

## **Viscous friction coefficient**

Viscous friction coefficient for the relative translational motion between the hub and the ring. The value of the parameter depends on lubrication state and quality of contacting surfaces. The coefficient must be greater than or equal to zero. The default value is  $100$  N/(m/s).



## Initial Conditions



# Synchronizer

---

## Initial state

Beginning configuration of cone and dog clutches. Choices include:

- Both clutches unlocked—Cone and dog clutches transmit zero torque between the ring and hub shafts.
- Cone clutch locked—Cone clutch transmits torque between the ring and hub shafts.
- Both clutches locked—Cone and dog clutches transmit torque between the ring and hub shafts.

The default value is Both clutches unlocked.

## Dog clutch initial shift linkage position

Initial position of the shift linkage section that attaches to the dog clutch. The value of the parameter has the following restrictions:

Linkage Travel Direction	Dog Clutch State	Parameter Restriction
Positive shift linkage displacement engages clutch	Initially engaged	Parameter must be greater than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>
	Initially disengaged	Parameter must be smaller than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and

Linkage Travel Direction	Dog Clutch State	Parameter Restriction
		<b>Tooth overlap to engage</b>
Negative shift linkage displacement engages clutch	Initially engaged	Negative of the parameter must be greater than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>
	Initially disengaged	Negative of the parameter must be smaller than the sum of parameters <b>Ring-hub clearance when dog clutch disengaged</b> and <b>Tooth overlap to engage</b>

The default value is 0 mm.

### **Cone clutch initial shift linkage position**

Initial position of the shift linkage section that attaches to the cone clutch. The value of the parameter has the following restrictions:

# Synchronizer

<b>Linkage Travel Direction</b>	<b>Dog Clutch State</b>	<b>Parameter Restriction</b>
Positive shift linkage displacement engages clutch	Initially engaged	Parameter must be greater than the value of <b>Ring-hub clearance when cone clutch disengaged</b>
	Initially disengaged	Parameter must be smaller than the value of <b>Ring-hub clearance when cone clutch disengaged</b>
Negative shift linkage displacement engages clutch	Initially engaged	Negative of the parameter must be greater than the value of <b>Ring-hub clearance when dog cone disengaged</b>
	Initially disengaged	Negative of the parameter must be smaller than the value of <b>Ring-hub clearance when dog cone disengaged</b>

The default value is 0 mm.

## **Initial dog clutch ring-hub offset angle**

Initial angle between the ring and hub portions of the dog clutch.  
To edit the parameter, you must set **Dog Clutch > Torque**

**transmission model** to Dynamic with backlash. If the clutch is disengaged, the initial angle must lie in the range  $-\pi/N \leq \theta \leq +\pi/N$ , where  $N$  denotes the number of teeth in the dog clutch. If the clutch is engaged, the initial angle must be in the range  $-\delta/2 \leq \theta \leq +\delta/2$ , where  $\delta$  is the backlash angle. The default value is 0 deg.

## Ports

Port	Description
S	Conserving translational port that represents the shift linkage
R	Conserving rotational port that represents the ring shaft
H	Conserving rotational port that represents the hub shaft
X1	Physical signal output port that measures the magnitude of the dog clutch translation
X2	Physical signal output port that measures the magnitude of the cone clutch translation

## See Also

Cone Clutch, Dog Clutch, Translational Detent

# Tire (Friction Parameterized)

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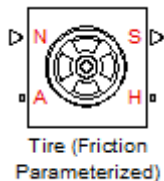
## Purpose

Model a tire using kinetic and static friction coefficients

## Library

Tires & Vehicles

## Description



The block represents a model of a tire that is parameterized with static and dynamic friction coefficients. When the static friction limit is exceeded, the tire loses traction and begins to slip. The kinetic friction coefficient determines the ability of the tire to transmit torque when slipping. The kinetic friction coefficient is either fixed or is a function of the current slip. The tire can regain traction when the magnitude of the relative velocity between the tire and ground is less than the traction velocity tolerance parameter.

Connection A is a mechanical rotational conserving port associated with the axle of the tire. Connection H is a mechanical translational conserving port associated with the hub of the tire. The physical signal input N is the normal force acting on the wheel. A force pushing the tire into the ground is considered positive. Output physical signal port S gives the current slip.

By default, the tire does not have inertia, stiffness, or damping. You can specify these additional dynamics using the **Dynamics** tab. These dynamics require additional computation and might make the model less suitable for real-time simulation.

The **Advanced** tab contains parameters that the state machine uses to determine the mode of the tire. The parameter defaults are reasonable in many simulations. Adjust the parameters if the model is not switching from the traction state to the slipping state (or vice-versa) at appropriate times in the simulation, adjust the parameters.

The traction or slip state model for the Tire (Friction Parameterized) block is based on that of the Fundamental Friction Clutch block. For more information on the state machine, see “Clutch State, Transition, and Variable Summary”.

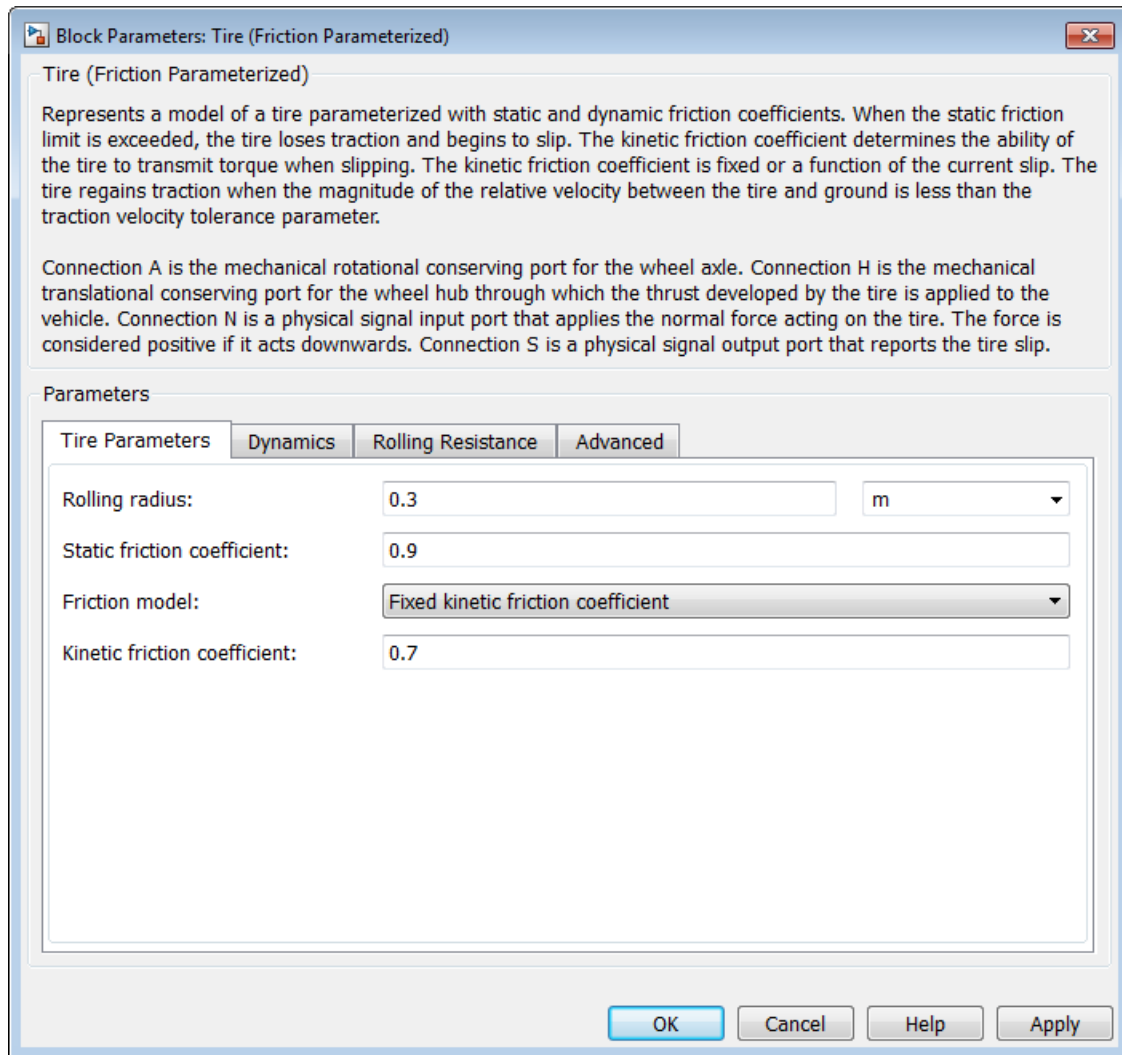
## **Assumptions and Limitations**

The kinetic friction coefficient can vary only as a function of slip during simulation. Therefore, you cannot drive over multiple different surfaces during one simulation.

## **Dialog Box and Parameters**

The dialog box contains one active pane, **Parameters**.

# Tire (Friction Parameterized)



## Tire Parameters

Use the **Tire Parameters** pane to specify characteristics of the tire, such as the rolling radius or static friction coefficient.



## Rolling radius

The effective rolling radius of the tire. The rolling radius must be greater than zero. Default is 0.30 m.

## Static friction coefficient

The ratio of the allowable longitudinal force to the normal force allowed before the tire begins to slip ( $\mu_s$ ). The parameter must be greater than either the kinetic friction coefficient or the largest value in the kinetic friction coefficient vector. Default is 0.9.

## Friction model

Specifies whether the kinetic friction coefficient is constant or a function of slip. The choices are:

- Fixed kinetic friction coefficient — (Default) Uses a constant kinetic function coefficient. Exposes **Kinetic friction coefficient** parameter.
- Table lookup kinetic friction coefficient — Uses a function of slip. Exposes **Tire slip vector** and **Kinetic friction coefficient vector** parameters.

## Kinetic friction coefficient

The ratio of the transmitted longitudinal force to the normal force allowed while the tire is slipping ( $\mu_k$ ). The ratio must be greater than zero. Default 0.7.

The **Kinetic friction coefficient** parameter appears if the **Friction model** parameter is Fixed kinetic friction coefficient.

## Tire slip vector

Vector that contains values of tire slip that correspond to the kinetic friction coefficients in the **Kinetic friction coefficient vector** parameter. The vectors must be the same size. If the **Tire slip vector** parameter contains non-only negative values, the slip versus friction function is assumed to be symmetric about the slip axis. Default vector is [0 0.02 0.06 0.15 0.6 1].

# Tire (Friction Parameterized)

---

The **Tire slip vector** parameter appears if you set the **Friction model** parameter to Table lookup kinetic friction coefficient.

## **Kinetic friction coefficient vector**

Vector that contains kinetic friction coefficients that correspond to the tire slip values specified in the **Tire slip vector** parameter. The vectors must be the same size. Default vector is [0.89 0.88 0.8 0.75 0.7 0.7].

The **Kinetic friction coefficient vector** parameter appears if the **Friction model** parameter is Table lookup kinetic friction coefficient.

## **Interpolation method**

The interpolation method for the lookup table to use for processing the tire slip-kinetic friction coefficient characteristic. The interpolation method choices are:

- Linear (default)
- Cubic
- Spline

The interpolation uses the PS Lookup Table (1-D).Simscape block.

The **Interpolation method** parameter appears if the **Friction model** parameter is Table lookup kinetic friction coefficient.

## **Extrapolation method**

The extrapolation method for the lookup table to use for processing the tire slip-kinetic friction coefficient characteristic.

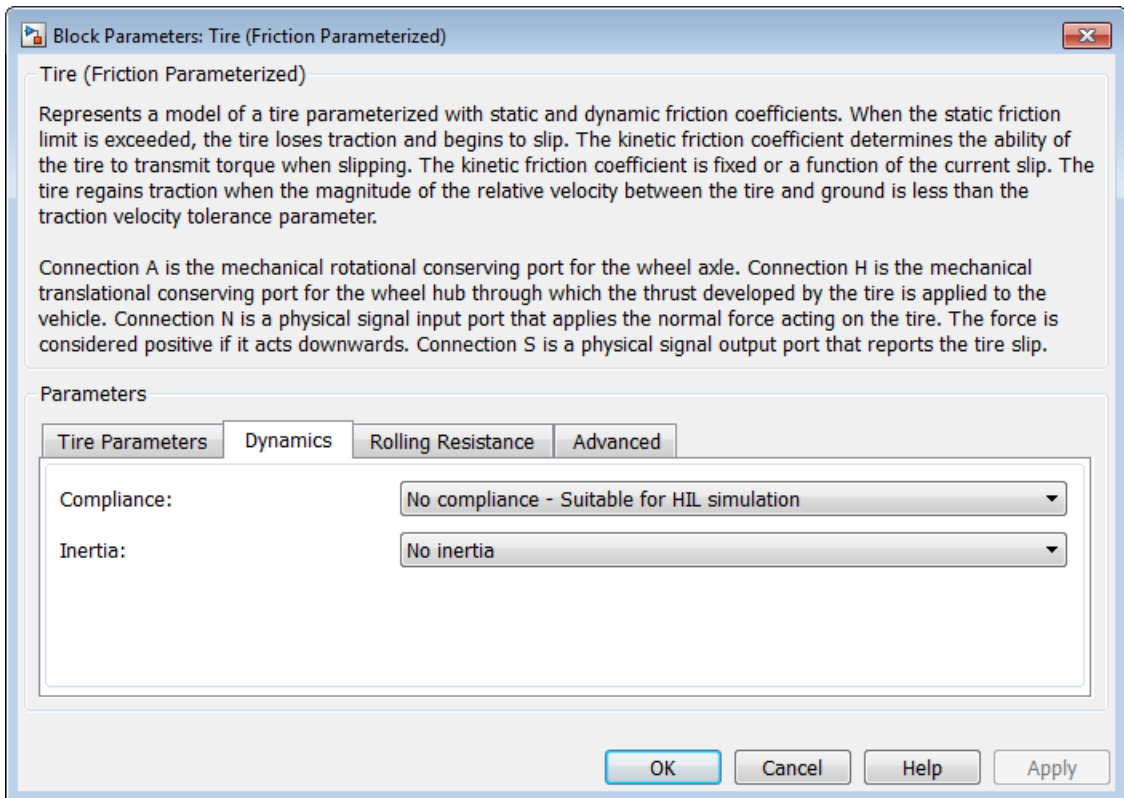
- From last 2 points — (Default) Performs extrapolation with linear method, regardless of the specified interpolation method.
- From last point — Preserves last value if an argument falls outside of the specified range.

# Tire (Friction Parameterized)

The **Extrapolation method** parameter appears if you set the **Friction model** parameter to Table lookup kinetic friction coefficient.

## Dynamics

Select the **Dynamics** pane to specify inertia, stiffness, and damping.



## Compliance

Specifies whether the model includes longitudinal stiffness and damping.

# Tire (Friction Parameterized)

---

- No compliance - Suitable for HIL simulation — (Default) Does not include longitudinal stiffness and damping.
- Specify stiffness and damping — Includes longitudinal stiffness and damping. Exposes the **Longitudinal stiffness** and **Longitudinal damping** parameters.

## Longitudinal stiffness

The longitudinal stiffness of the tire. The parameter must be greater than zero. Default is  $1e+6$  N/m.

The **Longitudinal stiffness** parameter appears if the **Compliance** parameter is Specify stiffness and damping.

## Longitudinal damping

The longitudinal damping of the tire. The parameter must be greater than zero. Default is  $1000$  N/(m/s).

The **Longitudinal damping** parameter appears if the **Compliance** parameter is Specify stiffness and damping.

## Inertia

Specifies whether the model the model includes tire inertia.

- No inertia — (Default) Does not include tire inertia.
- Specify inertia and initial velocity — Includes tire inertia and initial velocity. Exposes the **Tire inertia** and **Initial velocity** parameters.

## Tire inertia

The rotational inertia of the tire. The parameter must be greater than zero. Default is  $1$  kg\*m<sup>2</sup>.

The **Tire inertia** parameter appears if the **Inertia** parameter is Specify inertia and initial velocity.

## Initial velocity

The initial rotational velocity of the tire. Default is  $0$  rad/s.

## Tire (Friction Parameterized)

---

The **Initial velocity** parameter appears if the **Inertia** parameter is Specify inertia and initial velocity.

# Tire (Friction Parameterized)

## Rolling Resistance

**Block Parameters: Tire (Friction Parameterized)**

Tire (Friction Parameterized)

Represents a model of a tire parameterized with static and dynamic friction coefficients. When the static friction limit is exceeded, the tire loses traction and begins to slip. The kinetic friction coefficient determines the ability of the tire to transmit torque when slipping. The kinetic friction coefficient is fixed or a function of the current slip. The tire regains traction when the magnitude of the relative velocity between the tire and ground is less than the traction velocity tolerance parameter.

Connection A is the mechanical rotational conserving port for the wheel axle. Connection H is the mechanical translational conserving port for the wheel hub through which the thrust developed by the tire is applied to the vehicle. Connection N is a physical signal input port that applies the normal force acting on the tire. The force is considered positive if it acts downwards. Connection S is a physical signal output port that reports the tire slip.

Parameters

Tire Parameters	Dynamics	Rolling Resistance	Advanced
Rolling resistance:		Specify rolling resistance	
Resistance model:		Pressure and velocity dependent	
Tire pressure:	250e3		Pa
Alpha:	-0.003		
Beta:	0.97		
Coefficient A:	84e-4		
Coefficient B:	6.2e-4		s/m
Coefficient C:	1.6e-4		s <sup>2</sup> /m <sup>2</sup>
Velocity threshold:	1e-3		m/s

OK Cancel Help Apply

### **Rolling resistance**

Method used to specify the rolling resistance acting on a rotating wheel hub. The default value is `Specify rolling resistance`.

### **No rolling resistance**

Select this option to ignore the effect of rolling resistance on a model.

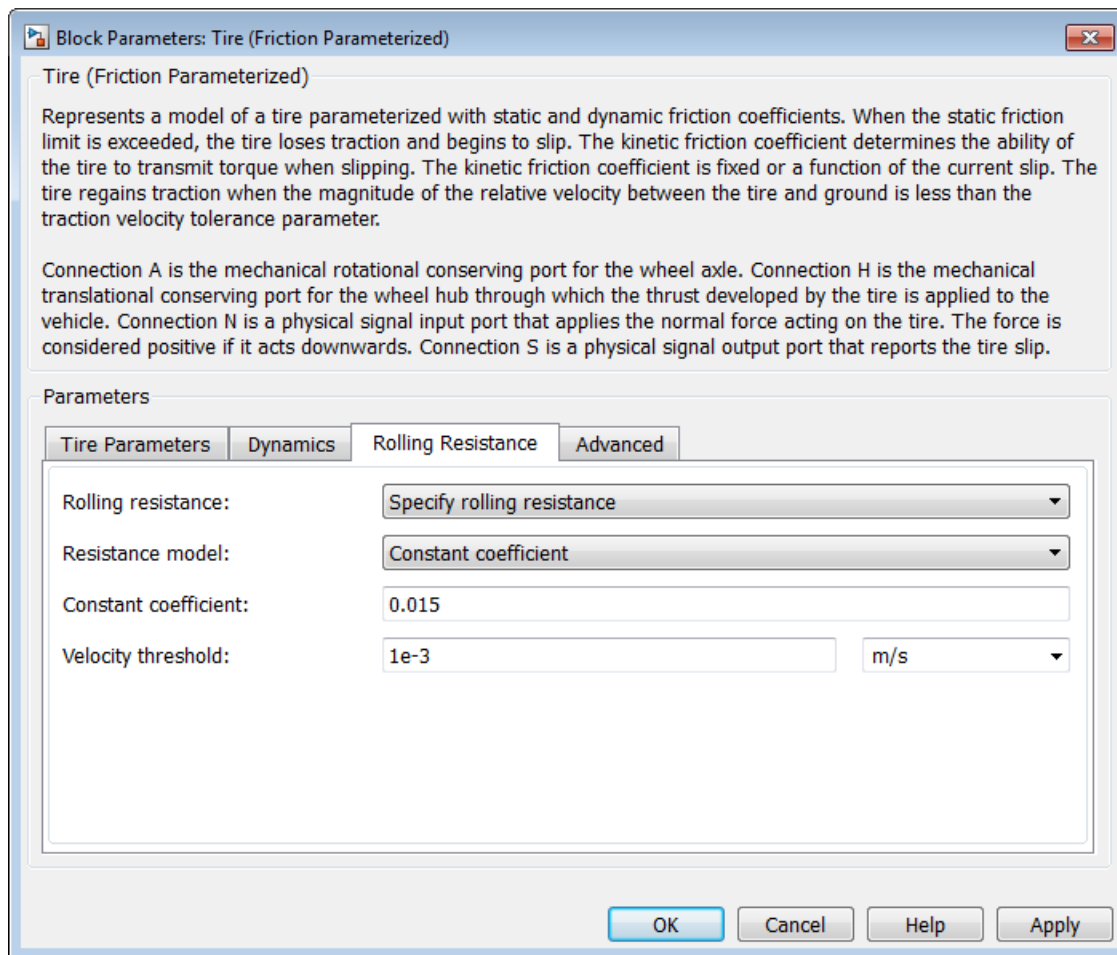
### **Specify rolling resistance**

Select between two rolling resistance models: `Constant coefficient` and `Pressure and velocity dependent`.

The default value is `Pressure and velocity dependent`.

# Tire (Friction Parameterized)

## Constant coefficient



## Constant Coefficient

Coefficient that sets the proportionality between the normal force and the rolling resistance force. The parameter must be greater than zero. The default value is 255.



## Velocity Threshold

Velocity at which the full rolling resistance force is transmitted to the rolling hub. The parameter ensures the force remains continuous during velocity direction changes, which increases the numerical stability of the simulation. The parameter must be greater than zero. The default value is 0.001 m/s.

# Tire (Friction Parameterized)

## Pressure and velocity dependent

**Block Parameters: Tire (Friction Parameterized)**

Tire (Friction Parameterized)

Represents a model of a tire parameterized with static and dynamic friction coefficients. When the static friction limit is exceeded, the tire loses traction and begins to slip. The kinetic friction coefficient determines the ability of the tire to transmit torque when slipping. The kinetic friction coefficient is fixed or a function of the current slip. The tire regains traction when the magnitude of the relative velocity between the tire and ground is less than the traction velocity tolerance parameter.

Connection A is the mechanical rotational conserving port for the wheel axle. Connection H is the mechanical translational conserving port for the wheel hub through which the thrust developed by the tire is applied to the vehicle. Connection N is a physical signal input port that applies the normal force acting on the tire. The force is considered positive if it acts downwards. Connection S is a physical signal output port that reports the tire slip.

Parameters

Tire Parameters | Dynamics | **Rolling Resistance** | Advanced

Rolling resistance: Specify rolling resistance

Resistance model: Pressure and velocity dependent

Tire pressure: 250e3 Pa

Alpha: -0.003

Beta: 0.97

Coefficient A: 84e-4

Coefficient B: 6.2e-4 s/m

Coefficient C: 1.6e-4 s<sup>2</sup>/m<sup>2</sup>

Velocity threshold: 1e-3 m/s

OK Cancel Help Apply

## **Tire pressure**

Inflation pressure of the tire. The parameter must be greater than zero. The default value is  $250e+3$  Pa.

## **Alpha**

Exponent of the tire pressure in the model equation. See “Pressure and Velocity Dependent Model” on page 1-254. The default value is  $-0.003$ .

## **Beta**

Exponent of the normal force model equation. The default value is  $0.97$ .

## **Coefficient A**

Velocity independent force component in the model equation. The parameter must be greater than zero. The default value is  $84e-4$ .

## **Coefficient B**

Velocity dependent force component in the model equation. The parameter must be greater than zero. The default value is  $6.2e-4$  s/m.

## **Coefficient C**

Force component that depends on the square of the velocity term in the model equation. The parameter must be greater than zero. The default value is  $1.6e-4$  s<sup>2</sup>/m<sup>2</sup>.

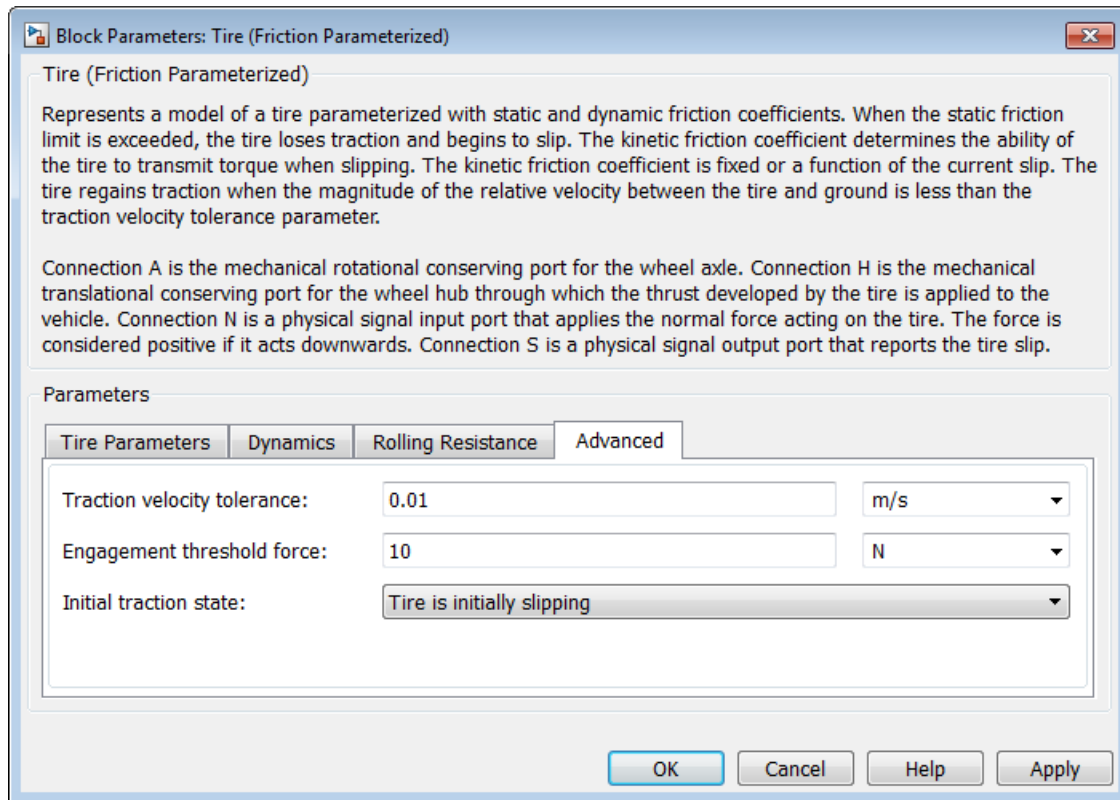
## **Velocity Threshold**

Velocity at which the full rolling resistance force is transmitted to the rolling hub. The parameter ensures the force remains continuous during velocity direction changes, which increases the numerical stability of the simulation. The parameter must be greater than zero. The default value is  $1e-3$  m/s.

## **Advanced**

Use the **Advanced** tab to specify parameters that the state machine uses to determine the mode of the tire.

# Tire (Friction Parameterized)



## Traction velocity tolerance

The magnitude of the relative velocity between the tire and ground at which the tire regains traction. Setting this value too low prevents the tire from entering a state where it has traction. Setting it too high can cause the tire to suddenly change velocity when the tire gains traction, and can result in an unstable simulation. The parameter must be greater than zero. Default is 0.01 m/s.

# Tire (Friction Parameterized)

## Engagement threshold force

Normal force values below the **Engagement threshold force** are not applied to the tire. Setting this value too low can cause the tire to gain and lose traction rapidly. Setting this value too high can give unrealistically low static and dynamic friction forces. The parameter must be greater than zero. Default is 10 N.

## Initial traction state

Specifies whether the tire is initially in traction or slipping.

- Tire is initially slipping — (Default)
- Tire is initially in traction

## Ports

Port	Description
A	Conserving mechanical rotational port associated with the axle of the tire
H	Conserving mechanical translational port associated with the hub of the tire
N	Physical signal input port associated with the normal force on the tire
S	Physical signal output port associated with the relative slip between the tire and road

## Example

The `sdl_4wd_dynamics` example model shows the use of the Tire (Friction Parameterized) block.

## See Also

Fundamental Friction Clutch block. For more information on the state machine, see “Clutch State, Transition, and Variable Summary”

Tire (Simple)

Tire (Magic Formula)

Tire-Road Interaction (Magic Formula)

# Tire (Magic Formula)

---

**Purpose** Tire with longitudinal dynamics and motion approximated by Magic Formula

**Library** Tires & Vehicles

**Description** The Tire (Magic Formula) block models the longitudinal dynamics of a vehicle axle-wheel-tire combination, with road contact represented by the Magic Formula and optional deformation compliance. The convention for the vertical load is positive downward. If the vertical load is zero or negative, the horizontal tire force vanishes. In that case, the tire is just touching the ground or has left the ground.



The longitudinal direction lies along the forward-backward axis of the tire. For model details, see “Tire Model” on page 1-381.

---

**Note** Tire (Magic Formula) is based on the Tire-Road Interaction (Magic Formula) block and the Simscape Wheel and Axle, Translational Spring, and Translational Damper blocks. For more information, see the Tire-Road Interaction (Magic Formula) block reference page about how Tire (Magic Formula) behaves in contact with the road.

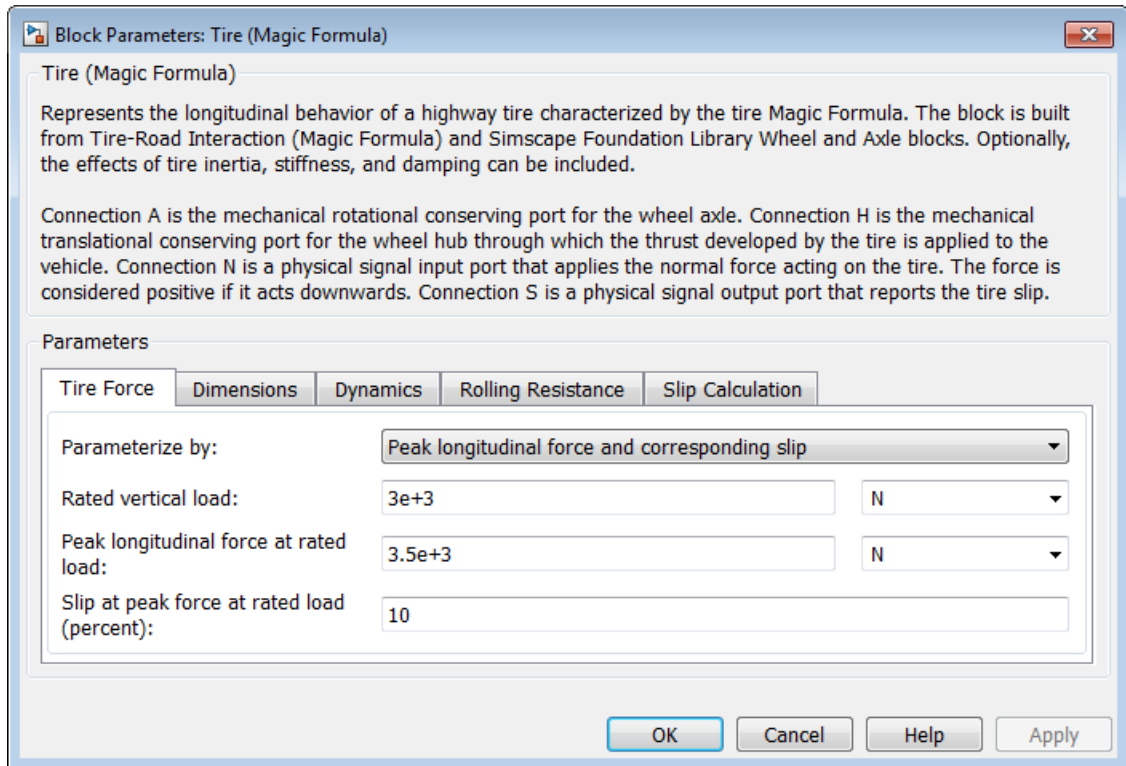
---

## Ports

You specify the downward vertical load  $F_z$  through a physical input signal at port N. The block reports the developed tire slip  $\kappa$ , as a decimal fraction, through a physical signal output at port S.

The wheel axle rotation is represented by the rotational conserving port A. The wheel axle transfer of horizontal thrust reaction to the vehicle is represented by the translational conserving port H.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with four tabs.

### Tire Force

#### Parameterize by

Select how to use the Magic Formula to model the tire-road interaction. The default is Peak longitudinal force and corresponding slip.

# Tire (Magic Formula)

---

- Peak longitudinal force and corresponding slip — Parametrize the Magic Formula with physical characteristics of the tire.

## Peak Force and Corresponding Slip

### Rated vertical load

Rated vertical load force  $F_{z0}$ . The default is 3e+3.

From the drop-down list, choose units. The default is newtons (N).

### Peak longitudinal force at rated load

Maximum longitudinal force  $F_{x0}$  that the tire exerts on the wheel when the vertical load equals its rated value  $F_{z0}$ . The default is 3500.

From the drop-down list, choose units. The default is newtons (N).

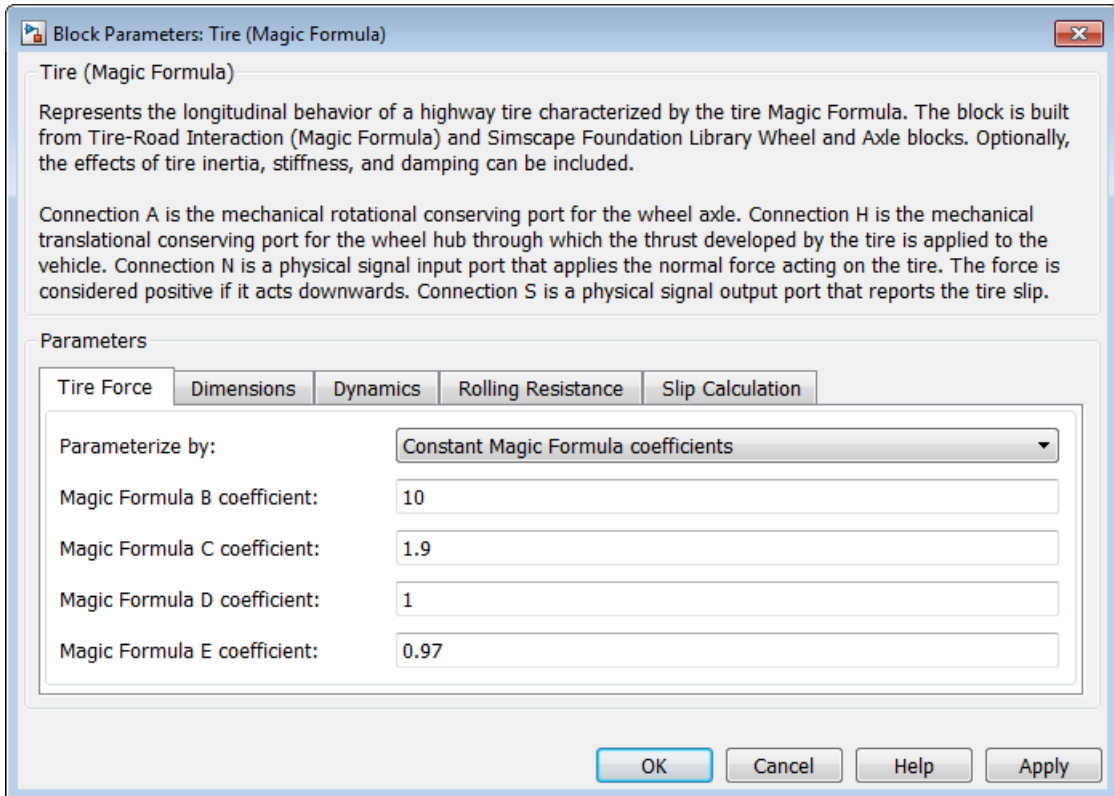
### Slip at peak force at rated load (percent)

The value of the contact slip  $\kappa'_0$ , in percent (%), when the longitudinal force equals its maximum value  $F_{x0}$  and the vertical load equals its rated value  $F_{z0}$ . The default is 10.

- Constant Magic Formula coefficients—Parameterize the Magic Formula directly with its coefficients. If you select this option, the panel changes from its default.



## Constant Magic Formula Coefficients



## Magic formula coefficients

Magic Formula *B*, *C*, *D*, and *E* coefficients. The defaults are:

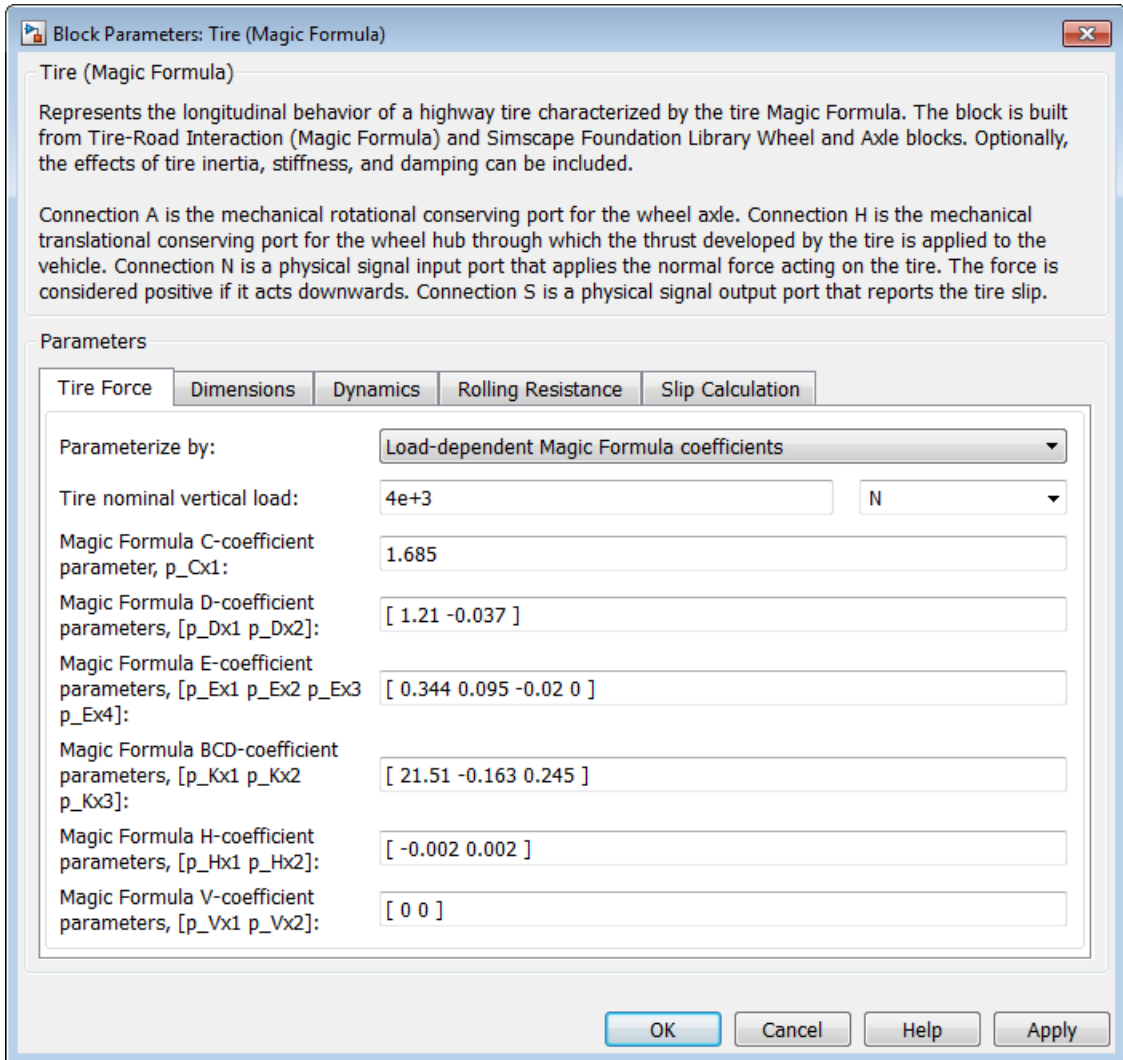
<b>B</b>	<b>C</b>	<b>D</b>	<b>E</b>
10	1.9	1	0.97

## Tire (Magic Formula)

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- Load-dependent Magic Formula coefficients—Parametrize the Magic Formula directly with load-dependent coefficients. If you select this option, the panel changes from its default.

## Load-Dependent Magic Formula Coefficients



# Tire (Magic Formula)

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## Tire nominal vertical load

Nominal normal force  $F_{z0}$  on tire. The default is  $4e+3$ .

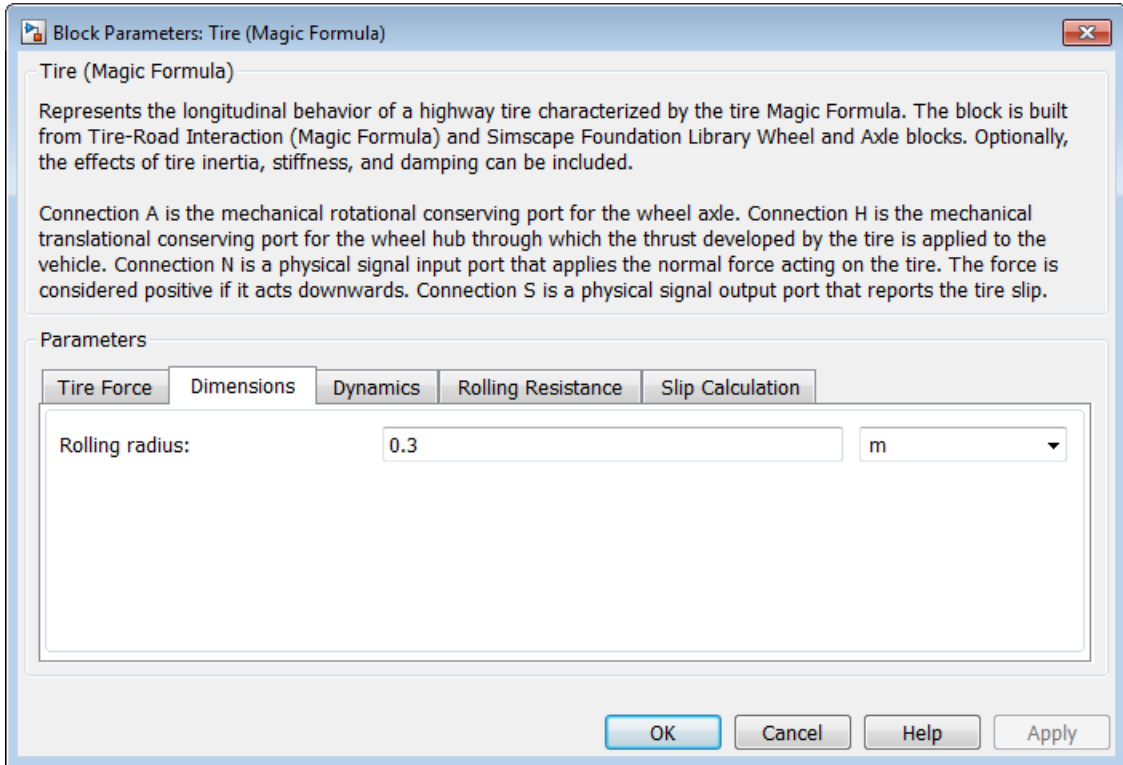
From the drop-down list, choose units. The default is newtons (N).

## Magic formula coefficient parameters

Specify the parameters that define the load-dependent  $C$ ,  $D$ ,  $E$ ,  $K$ ,  $H$ , and  $V$  coefficients as vectors, one for each coefficient. The defaults are:

Coefficient	Parameters	Default Values
C	p_Cx1	1.685
D	[ p_Dx1 p_Dx2 ]	[ 1.21 -0.037 ]
E	[ p_Ex1 p_Ex2 p_Ex3 p_Ex4 ]	[ 0.344 0.095 -0.02 0 ]
K	[ p_Kx1 p_Kx2 p_Kx3 ]	[ 21.51 -0.163 0.245 ]
H	[ p_Hx1 p_Hx2 ]	[ -0.002 0.002 ]
V	[ p_Vx1 p_Vx2 ]	[ 0 0 ]

## Dimensions



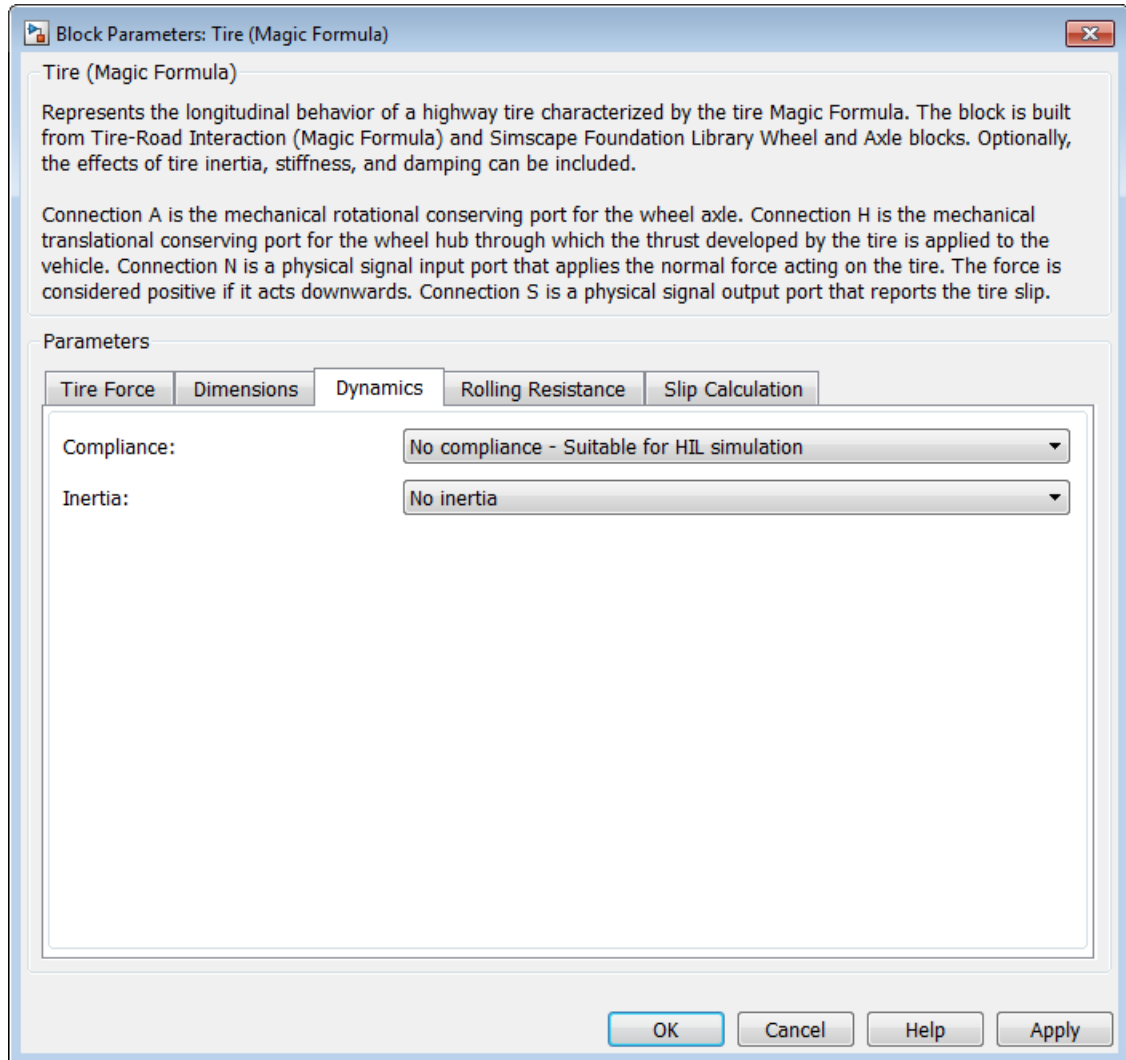
### Rolling radius

Unloaded tire-wheel radius  $r_w$ . The default is 0.3.

From the drop-down list, choose units. The default is meters (m).

# Tire (Magic Formula)

## Dynamics



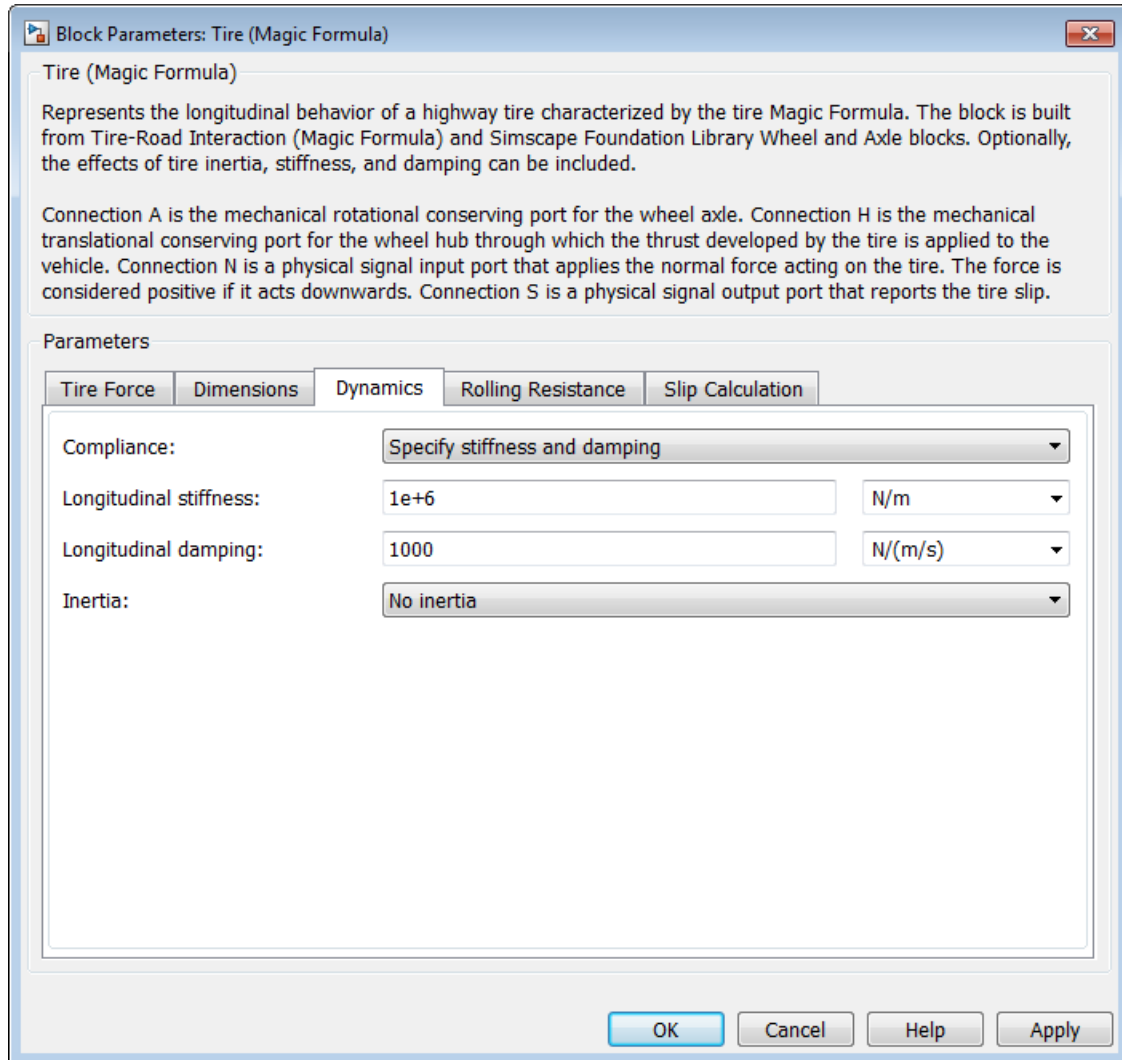
## Compliance

Select how to model the dynamical compliance of the tire. The default is No compliance Suitable for HIL simulation.

- No compliance Suitable for HIL simulation — Tire is modeled with no dynamical compliance.
- Specify stiffness and damping—Tire is modeled as a stiff, dampened spring and deforms under load. If you select this option, the panel changes from its default.

# Tire (Magic Formula)

## Tire Stiffness and Damping





## Longitudinal stiffness

Tire longitudinal stiffness  $C_{FX}$ . The default is 1e+6.

From the drop-down list, choose units. The default is newtons/meter (N/m).

## Longitudinal damping

Tire longitudinal damping  $b_{FX}$ . The default is 1000.

From the drop-down list, choose units. The default is newtons/(meters/second) (N/(m/s)).

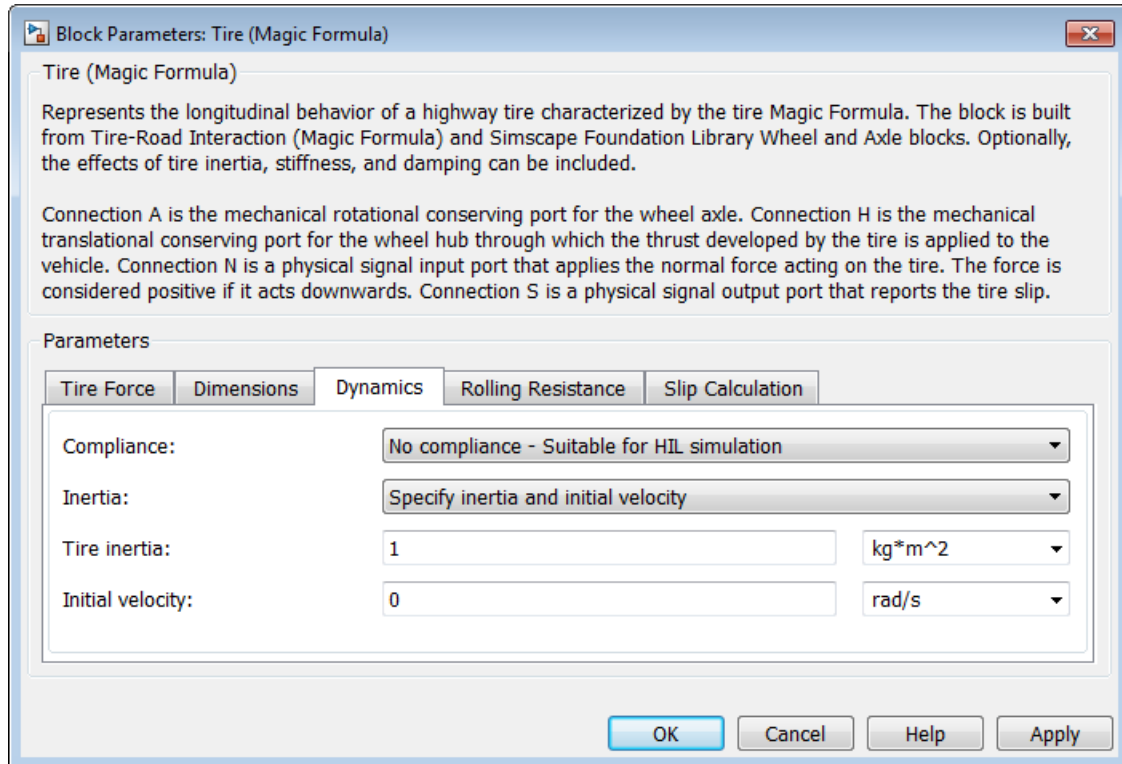
## Inertia

Select how to model the rotational inertia of the tire. The default is No inertia.

- No inertia—Tire is modeled with no inertia.
- Specify inertia and initial velocity—Tire is modeled with rotational inertia. If you select this option, the panel changes from its default.

# Tire (Magic Formula)

## Tire Rotational Inertia and Initial Velocity



### Tire Inertia

Rotational inertia  $I_w$  of the wheel-tire assembly. The default is 1.

From the drop-down list, choose units. The default is kilograms-meters-squared ( $\text{kg}\cdot\text{m}^2$ ).

### Initial velocity

Initial angular velocity  $\Omega(0)$  of the tire. The default is 0.

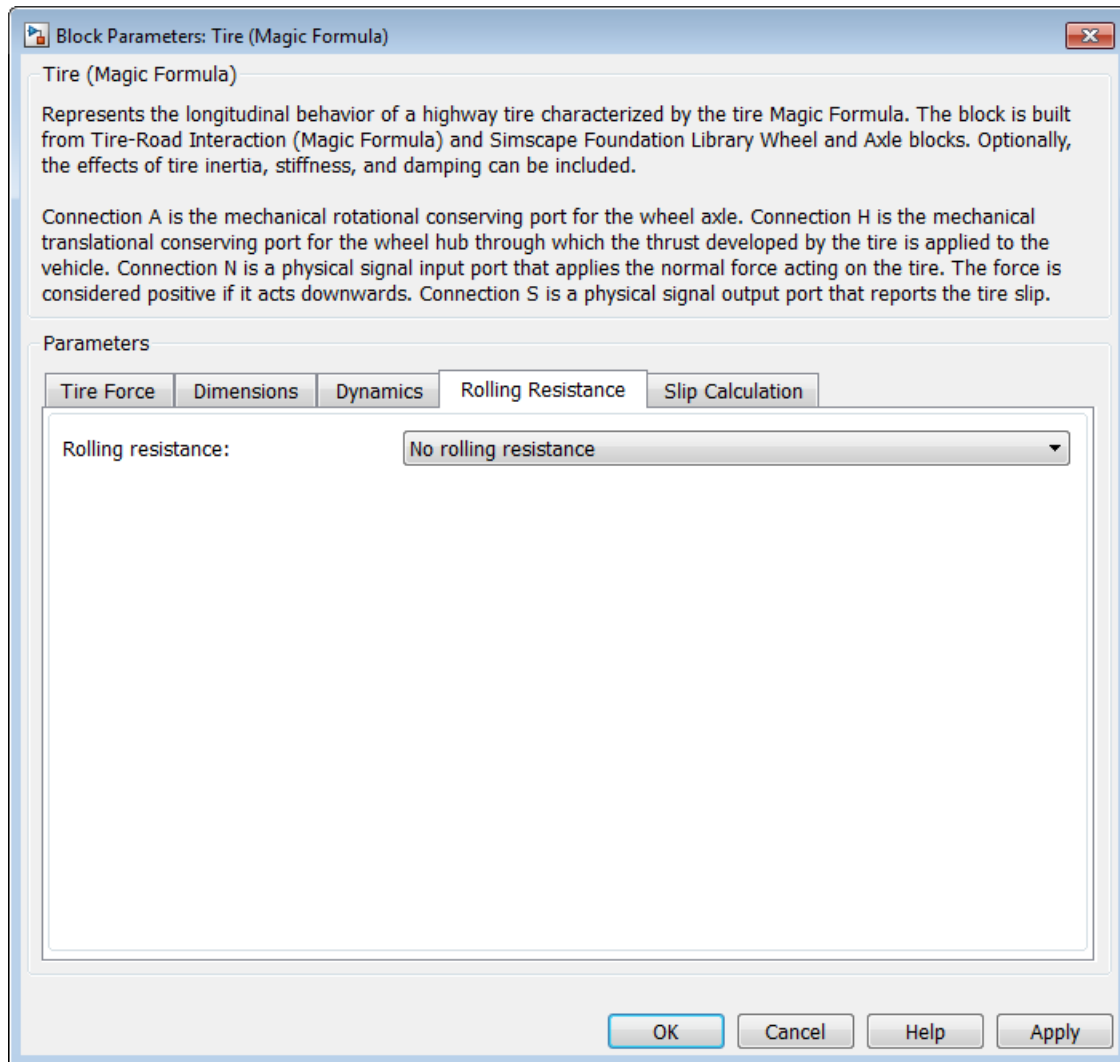
## Tire (Magic Formula)

---

From the drop-down list, choose units. The default is radians/second (rad/s).

# Tire (Magic Formula)

## Rolling Resistance



## **Rolling resistance**

Method used to specify the rolling resistance acting on a rotating wheel hub. The default value is No rolling resistance.

## **No rolling resistance**

Select this option to ignore the effect of rolling resistance on a model.

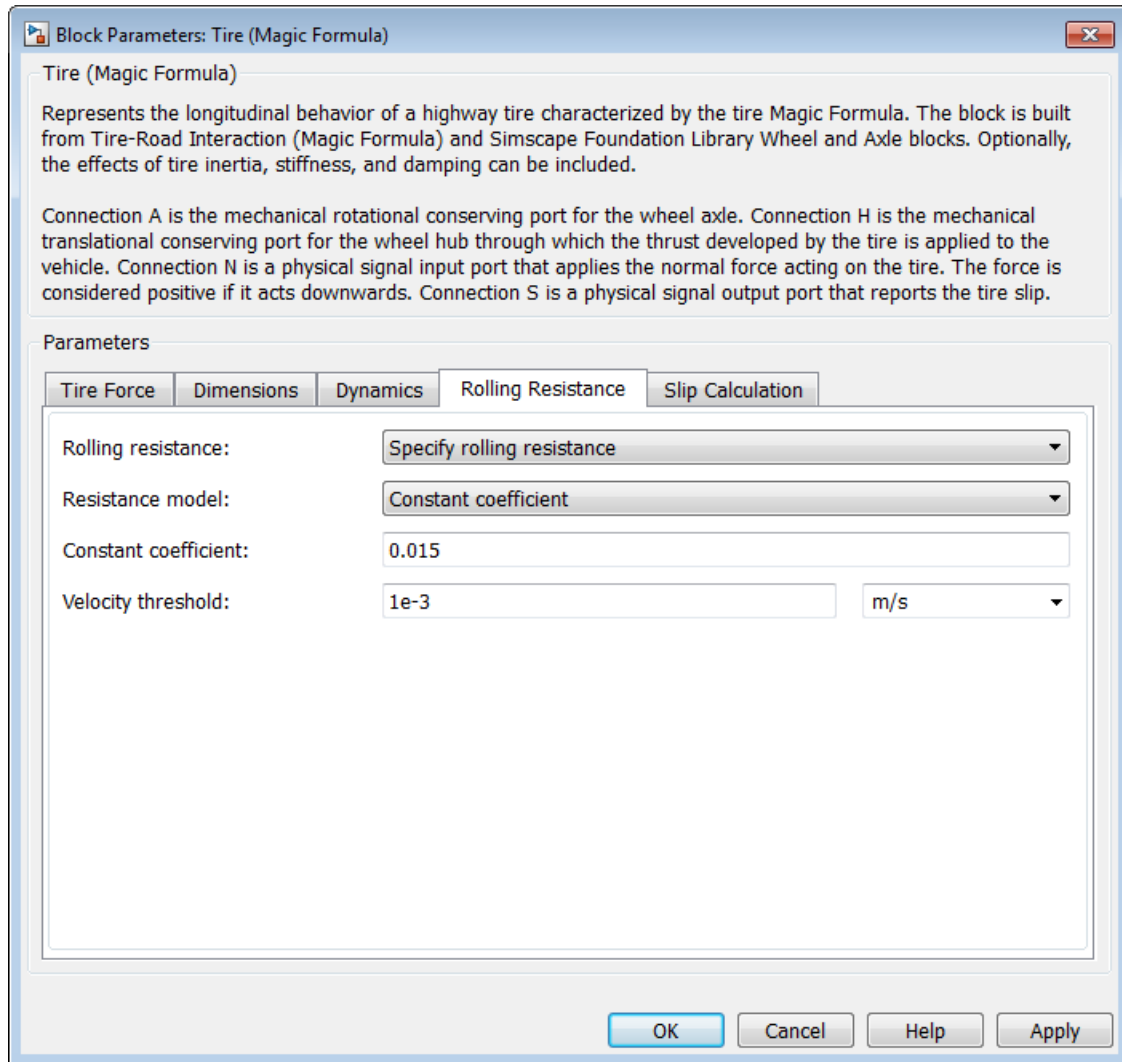
## **Specify rolling resistance**

Select between two rolling resistance models: Constant coefficient and Pressure and velocity dependent.

The default value is Constant coefficient.

# Tire (Magic Formula)

## Constant coefficient



## **Constant Coefficient**

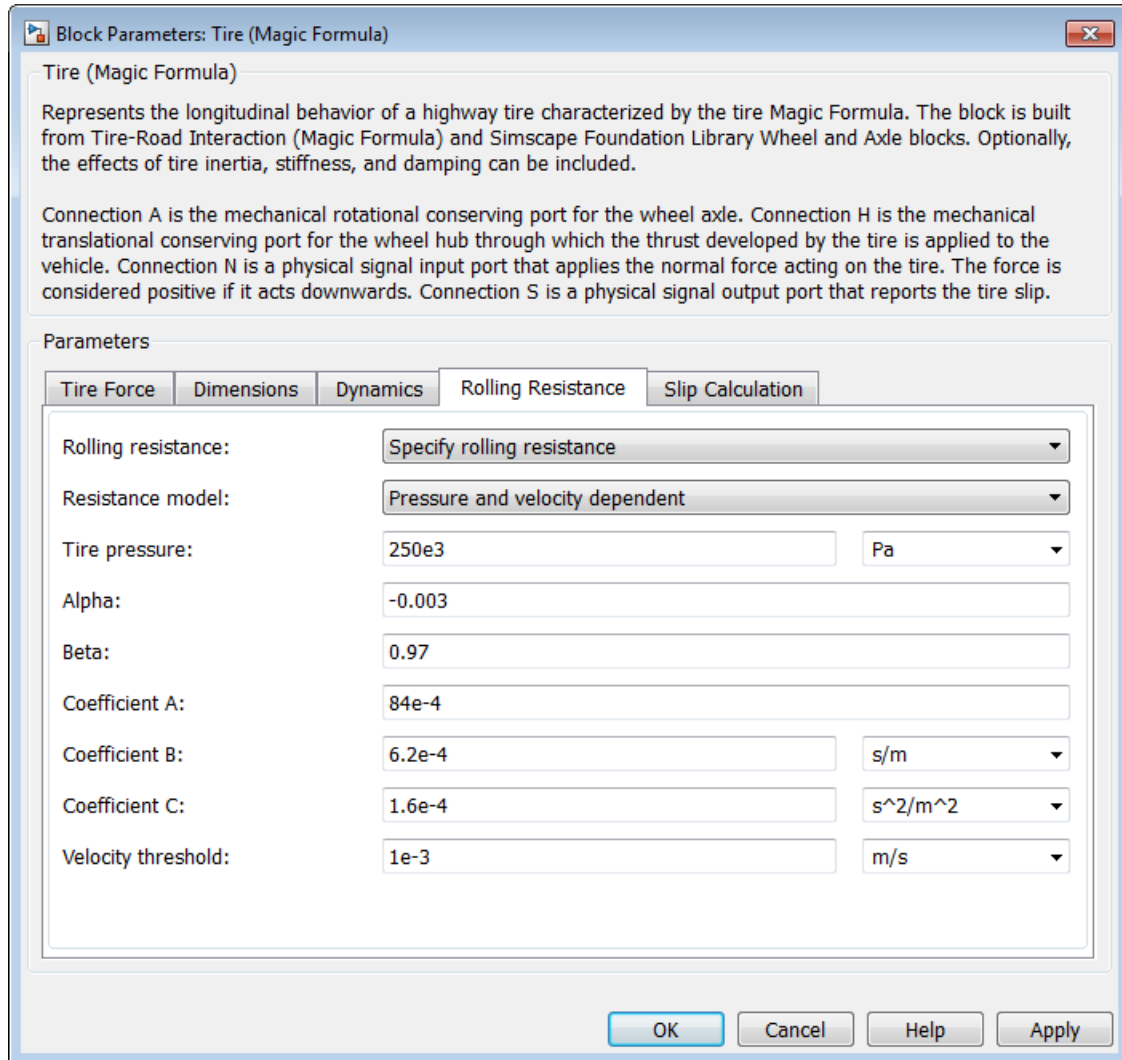
Coefficient that sets the proportionality between the normal force and the rolling resistance force. The parameter must be greater than zero. The default value is 255.

## **Velocity Threshold**

Velocity at which the full rolling resistance force is transmitted to the rolling hub. The parameter ensures the force remains continuous during velocity direction changes, which increases the numerical stability of the simulation. The parameter must be greater than zero. The default value is 0.001 m/s.

# Tire (Magic Formula)

## Pressure and velocity dependent





**Tire pressure**

Inflation pressure of the tire. The parameter must be greater than zero. The default value is 250e+3 Pa.

**Alpha**

Exponent of the tire pressure in the model equation. See “Pressure and Velocity Dependent Model” on page 1-254. The default value is -0.003.

**Beta**

Exponent of the normal force model equation. The default value is 0.97.

**Coefficient A**

Velocity independent force component in the model equation. The parameter must be greater than zero. The default value is 84e-4.

**Coefficient B**

Velocity dependent force component in the model equation. The parameter must be greater than zero. The default value is 6.2e-4 s/m.

**Coefficient C**

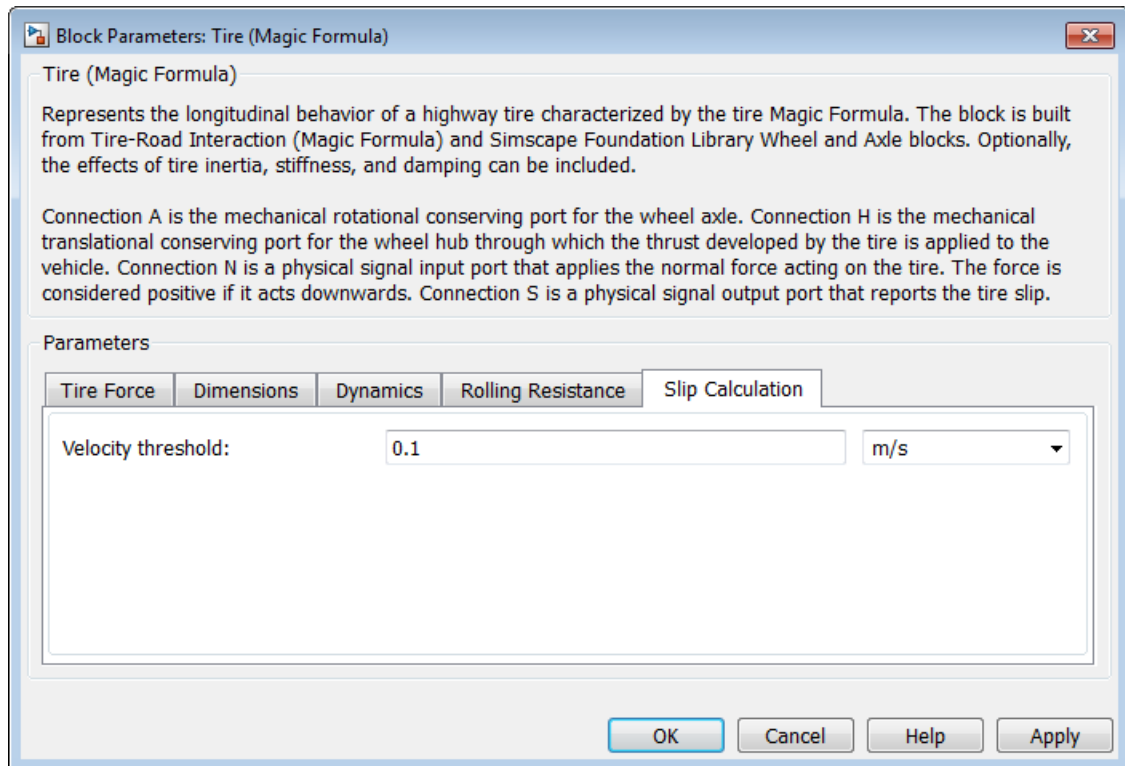
Force component that depends on the square of the velocity term in the model equation. The parameter must be greater than zero. The default value is 1.6e-4 s<sup>2</sup>/m<sup>2</sup>.

**Velocity Threshold**

Velocity at which the full rolling resistance force is transmitted to the rolling hub. The parameter ensures the force remains continuous during velocity direction changes, which increases the numerical stability of the simulation. The parameter must be greater than zero. The default value is 1e-3 m/s.

# Tire (Magic Formula)

## Slip Calculation



### Velocity threshold

The wheel hub velocity  $V_{th}$  below which the slip calculation is modified to avoid singular evolution at zero velocity. Must be positive. The default is 0.1.

From the drop-down list, choose units. The default is meters per second (m/s).

## Tire Model

The Tire block models the tire as a rigid wheel-tire combination in contact with the road and subject to slip. When torque is applied to the wheel axle, the tire pushes on the ground (while subject to contact friction) and transfers the resulting reaction as a force back on the wheel. This action pushes the wheel forward or backward. If you include the optional tire compliance, the tire also flexibly deforms under load.

The figure displays the forces on the tire. The table defines the tire model variables.



### Tire Motion and Forces

### Tire Model Variables

Symbol	Description and Unit
$r_w$	Wheel radius
$V_x$	Wheel hub longitudinal velocity
$u$	Tire longitudinal deformation

# Tire (Magic Formula)

## Tire Model Variables (Continued)

Symbol	Description and Unit
$\Omega$	Wheel angular velocity
$\Omega'$	Contact point angular velocity = $\Omega$ if $u = 0$
$r_w \Omega'$	Tire tread longitudinal velocity
$V_{sx} = r_w \Omega - V_x$	Wheel slip velocity
$V'_{sx} = r_w \Omega' - V_x$	Contact slip velocity = $V_{sx}$ if $u = 0$
$\kappa = V_{sx} /  V_x $	Wheel slip
$\kappa' = V'_{sx} /  V_x $	Contact slip = $\kappa$ if $u = 0$
$V_{th}$	Wheel hub threshold velocity
$F_z$	Vertical load on tire
$F_x$	Longitudinal force exerted on the tire at the contact point.
$C_{Fx} = (\partial F_x / \partial u)_0$	Tire longitudinal stiffness under deformation
$b_{Fx} = (\partial F_x / \partial \dot{u})_0$	Tire longitudinal damping under deformation
$I_w$	Wheel-tire inertia; effective mass = $I_w / r_w^2$
$\tau_{drive}$	Torque applied by the axle to the wheel

## Tire Kinematics and Response

### Roll and Slip

If the tire did not slip, it would roll and translate as  $V_x = r_w \Omega$ . But the tire actually does slip and develops a longitudinal force  $F_x$  only in response to slip.

The wheel slip velocity is  $V_{sx} = r_w \Omega - V_x$ . The *wheel slip* is  $\kappa = V_{sx} / |V_x|$ . For a locked, sliding wheel,  $\kappa = -1$ . For perfect rolling,  $\kappa = 0$ .

## Slip at Low Speed

For low speeds,  $|V_x| \leq |V_{th}|$ , the wheel slip is modified to:

$$\kappa = 2V_{sx}/(V_{th} + V_x^2/V_{th}).$$

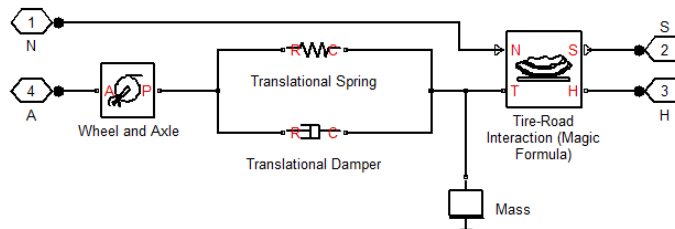
This modification allows for a nonsingular, nonzero slip at zero wheel velocity. For example, for perfect slipping (nontranslating spinning tire),  $V_x = 0$  while  $\kappa = 2r_w\Omega/V_{th}$  is finite.

## Deformation

If the tire is modeled with compliance, it is also flexible. Because in this case, the tire deforms, the tire-road contact point turns at a slightly different angular velocity  $\Omega'$  from the wheel  $\Omega$  and requires, instead of the wheel slip, the *contact point or contact patch slip*  $\kappa'$ . The block models the deforming tire as a translational spring-damper of stiffness  $C_{Fx}$  and damping  $b_{Fx}$ .

If the tire is not modeled with compliance, then  $\Omega' = \Omega$ ,  $V'_{sx} = V_{sx}$ , and  $\kappa' = \kappa$ . In this case, the tire starts simulation undeformed and remains undeformed.

## Tire and Wheel Dynamics



The full tire model is equivalent to this Simscape-SimDrivelink component diagram. It simulates both transient and steady-state behavior and correctly represents starting from, and coming to, a stop. The Translational Spring and Translational Damper are equivalent to the tire stiffness  $C_{Fx}$  and damping  $b_{Fx}$ . Tire-Road Interaction (Magic Formula) models the longitudinal force  $F_x$  on the tire as a function of

# Tire (Magic Formula)

---

$F_z$  and  $\kappa'$  using the Magic Formula, with  $\kappa'$  as the independent slip variable [1].

The Wheel and Axle radius is the wheel radius  $r_w$ . The Mass value is the effective mass,  $I_w/r_w^2$ . The tire characteristic function  $f(\kappa', F_z)$  determines the longitudinal force  $F_x$ . Together with the driveshaft torque applied to the wheel axis,  $F_x$  determines the wheel angular motion and longitudinal motion.

Without tire compliance, the Translational Spring and Translational Damper are omitted, and contact variables revert to wheel variables. In this case, the tire effectively has infinite stiffness, and port P of Wheel and Axle connects directly to port T of Tire-Road Interaction (Magic Formula).

Without tire inertia, the Mass is omitted.

## Limitations

The Tire (Magic Formula) block assumes longitudinal motion only and includes no camber, turning, or lateral motion.

## Tire Compliance

Tire compliance implies a time lag in the tire response to the forces on it. Time lag simulation increases model fidelity but reduces simulation performance. See “Adjust Model Fidelity”.

## Examples

These SimDriveline example models use the Tire (Magic Formula) block to include tires and tire-road load:

- `sdl_dual_clutch`
- `sdl_limited_slip_differential`
- `sdl_torsen_differential`
- `sdl_4wd_dynamics`
- `sdl_vehicle`

## References

[1] Pacejka, H. B. *Tire and Vehicle Dynamics*, Society of Automotive Engineers and Butterworth-Heinemann, Oxford, 2002, chapters 1,4,7, and 8

## See Also

Differential, Tire-Road Interaction (Magic Formula), Translational Damper, Translational Spring, Vehicle Body, Wheel and Axle  
“Complete Vehicle Model”

# Tire-Road Interaction (Magic Formula)

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**Purpose** Tire tread in contact with road surface approximated by Magic Formula

**Library** Tires & Vehicles/Tire Subcomponents

## Description



The Tire-Road Interaction (Magic Formula) block models the longitudinal tire-road contact force using the Magic Formula. Use this block as a component in tire models.

The convention for the vertical load is positive downward. If the vertical load is zero or negative, the horizontal tire force vanishes. In that case, the tire is just touching the ground or has left the ground. The longitudinal direction lies along the forward-backward axis of the tire. For model details, see “Tire-Road Interaction Model” on page 1-391.

## Ports

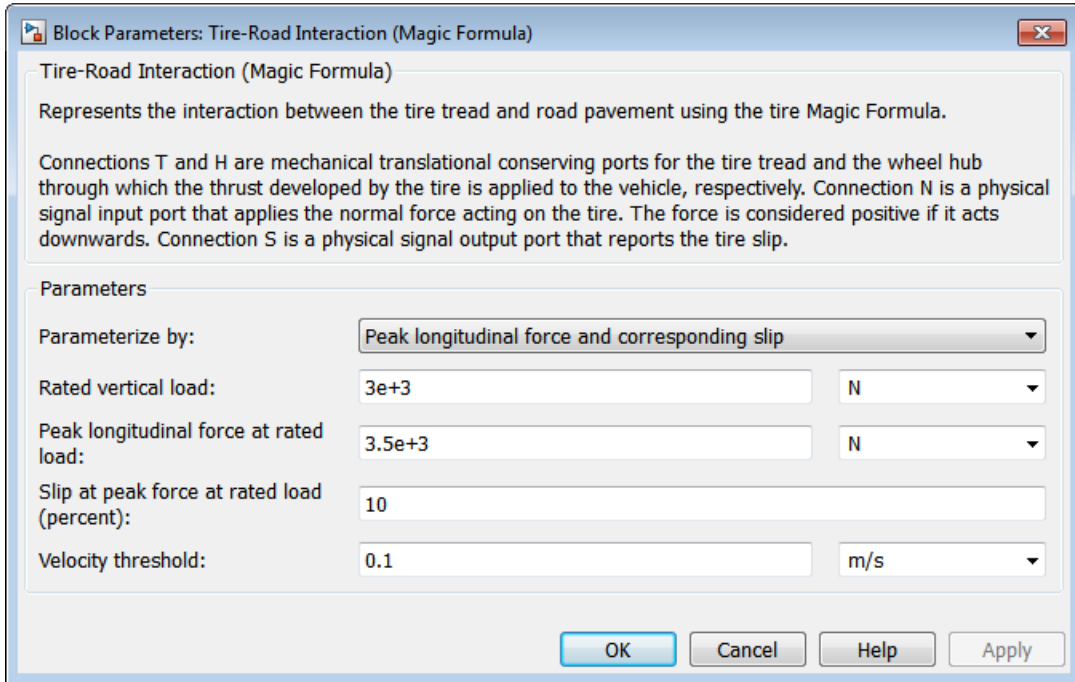
You specify the downward vertical load  $F_z$  (in newtons) through a physical signal input at port N. The block reports the developed tire slip  $\kappa$ , as a decimal fraction, through a physical signal output at port S.

The tire tread in contact with the road and the transfer of horizontal thrust reaction to the vehicle at the wheel hub are represented by the translational conserving ports T and H, respectively.



# Tire-Road Interaction (Magic Formula)

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**.

### Parametrize by

Select how to use the Magic Formula to model the tire-road interaction. The default is Peak longitudinal force and corresponding slip.

- Peak longitudinal force and corresponding slip — Parametrize the Magic Formula with physical characteristics of the tire.

# Tire-Road Interaction (Magic Formula)

## Peak Force and Corresponding Slip

Parameterize by:	Peak longitudinal force and corresponding slip	
Rated vertical load:	3e+3	N
Peak longitudinal force at rated load:	3.5e+3	N
Slip at peak force at rated load (percent):	10	

### Rated vertical load

Rated vertical load force  $F_{z0}$ . The default is 3000.

From the drop-down list, choose units. The default is newtons (N).

### Peak longitudinal force at rated load

Maximum longitudinal force  $F_{x0}$  that the tire exerts on the wheel when the vertical load equals its rated value  $F_{z0}$ . The default is 3500.

From the drop-down list, choose units. The default is newtons (N).

### Slip at peak force at rated load (percent)

The value of the contact slip  $\kappa_0$ , in percent (%), when the longitudinal force equals its maximum value  $F_{x0}$  and the vertical load equals its rated value  $F_{z0}$ . The default is 10.

- **Constant Magic Formula coefficients** — Parametrize the Magic Formula directly with constant coefficients. If you select this option, the panel changes from its default.

# Tire-Road Interaction (Magic Formula)

## Constant Magic Formula Coefficients

Parameterize by:	Constant Magic Formula coefficients
Magic formula B coefficient:	10
Magic formula C coefficient:	1.9
Magic formula D coefficient:	1
Magic formula E coefficient:	0.97

## Magic formula coefficients

Dimensionless Magic Formula *B*, *C*, *D*, and *E* coefficients.  
The defaults are:

<b>B</b>	<b>C</b>	<b>D</b>	<b>E</b>
10	1.9	1	0.97

- Load-dependent Magic Formula coefficients — Parametrize the Magic Formula directly with load-dependent coefficients. If you select this option, the panel changes from its default.

# Tire-Road Interaction (Magic Formula)

## Load-Dependent Magic Formula Coefficients

Parameterize by:	Load-dependent Magic Formula coefficients	
Tire nominal vertical load:	4e+3	N
Magic Formula C-coefficient parameter, p_Cx1:	1.685	
Magic Formula D-coefficient parameters, [p_Dx1 p_Dx2]:	[ 1.21 -0.037 ]	
Magic Formula E-coefficient parameters, [p_Ex1 p_Ex2 p_Ex3 p_Ex4]:	[ 0.344 0.095 -0.02 0 ]	
Magic Formula BCD-coefficient parameters, [p_Kx1 p_Kx2 p_Kx3]:	[ 21.51 -0.163 0.245 ]	
Magic Formula H-coefficient parameters, [p_Hx1 p_Hx2]:	[ -0.002 0.002 ]	
Magic Formula V-coefficient parameters, [p_Vx1 p_Vx2]:	[ 0 0 ]	

### Tire nominal vertical load

Nominal normal force  $F_{z0}$  on tire. The default is  $4e+3$ .

From the drop-down list, choose units. The default is newtons (N).

### Magic formula coefficient parameters

Specify the parameters that define the load-dependent  $C$ ,  $D$ ,  $E$ ,  $K$ ,  $H$ , and  $V$  coefficients as vectors, one for each coefficient. The defaults are:

Coefficient	Parameters	Default Values
K	[ p_Kx1 p_Kx2 p_Kx3 ]	[ 21.51 -0.163 0.245 ]
H	[ p_Hx1 p_Hx2 ]	[ -0.002 0.002 ]
V	[ p_Vx1 p_Vx2 ]	[ 0 0 ]
C	p_Cx1	1.685

# Tire-Road Interaction (Magic Formula)

Coefficient	Parameters	Default Values
D	[ p_Dx1 p_Dx2 ]	[ 1.21 -0.037 ]
E	[ p_Ex1 p_Ex2 p_Ex3 p_Ex4 ]	[ 0.344 0.095 -0.02 0 ]

## Velocity threshold

The wheel hub velocity  $V_{th}$  below which the slip calculation is modified to avoid singular evolution at zero velocity. Must be positive. The default is 0.1.

From the drop-down list, choose units. The default is meters per second (m/s).

## Tire-Road Interaction Model

The Tire-Road Interaction (Magic Formula) block models the longitudinal forces at the tire-road contact patch using the Magic Formula of Pacejka [7].

The figure displays the forces on the tire. The table defines the tire model variables.

# Tire-Road Interaction (Magic Formula)



## Tire-Road Contact Motion and Forces

### Tire-Road Contact Variables

Symbol	Description and Unit
$\Omega$	Wheel angular velocity
$r_w$	Wheel radius
$V_x$	Wheel hub longitudinal velocity
$r_w \Omega$	Tire tread longitudinal velocity
$V_{sx} = r_w \Omega - V_x$	Wheel slip velocity = tread velocity T – hub velocity H
$\kappa = V_{sx} /  V_x $	Wheel slip

# Tire-Road Interaction (Magic Formula)

## Tire-Road Contact Variables (Continued)

Symbol	Description and Unit
$F_z, F_{z0}$	Vertical load and nominal vertical load on tire
$F_x = f(\kappa, F_z)$	Longitudinal force exerted on the tire at the contact point. Also a characteristic function $f$ of the tire.

## Tire Response

### Forces and Characteristic Function

A tire model provides a steady-state *tire characteristic function*  $F_x = f(\kappa, F_z)$ , the longitudinal force  $F_x$  on the tire, based on:

- Vertical load  $F_z$
- Wheel slip  $\kappa$

### Magic Formula with Constant Coefficients

The Magic Formula is a specific form for the tire characteristic function, characterized by four dimensionless coefficients ( $B, C, D, E$ ), or stiffness, shape, peak, and curvature:

$$F_x = f(\kappa, F_z) = F_z \cdot D \cdot \sin( C \cdot \arctan[ \{ B\kappa - E \cdot [ B\kappa - \arctan(B\kappa) ] \} ] ) .$$

The slope of  $f$  at  $\kappa = 0$  is  $BCD \cdot F_z$ .

### Magic Formula with Load-Dependent Coefficients

A more general Magic Formula uses dimensionless coefficients that are functions of the tire load. A more complex set of parameters  $p_i$ , entered in the dialog box, specifies these functions:

$$F_{x0} = D_x \cdot \sin( C_x \cdot \arctan[ \{ B_x \kappa_x - E_x \cdot [ B_x \kappa_x - \arctan(B_x \kappa_x) ] \} ] ) + S_{Vx} ,$$

where

# Tire-Road Interaction (Magic Formula)

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$$df_z = (F_z - F_{z0})/F_z ,$$

$$\kappa_x = \kappa + S_{Hx} ,$$

$$C_x = p_{Cx1} ,$$

$$D_x = \mu_x \cdot F_z ,$$

$$\mu_x = p_{Dx1} + p_{Dx2} \cdot df_z ,$$

$$E_x = (p_{Ex1} + p_{Ex2} \cdot df_z + p_{Ex3} \cdot df_z^2)[1 - p_{Ex4} \cdot \text{sgn}(\kappa_x)] ,$$

$$K_{xx} = F_z \cdot (p_{Kx1} + p_{Kx2} \cdot df_z) \cdot \exp(p_{Kx3} \cdot df_z) ,$$

$$B_x = K_{xx} / (C_x D_x + \varepsilon_x) ,$$

$$S_{Hx} = p_{Hx1} + p_{Hx2} \cdot df_z ,$$

$$S_{Vx} = F_z \cdot (p_{Vx1} + p_{Vx2} \cdot df_z) .$$

$S_{Hx}$  and  $S_{Vx}$  represent offsets to the slip and longitudinal force in the force-slip function, or horizontal and vertical offsets if the function is plotted as a curve.  $\mu_x$  is the longitudinal load-dependent friction coefficient.  $\varepsilon_x$  is a small number inserted to prevent division by zero as  $F_z$  approaches zero.

## Peak Longitudinal Force and Corresponding Slip

The block uses a representative set of Magic Formula coefficients. The block scales the coefficients to yield the peak longitudinal force  $F_{x0}$  at the corresponding slip  $\kappa_0$  that you specify, for rated vertical load  $F_{z0}$ .

## Magic Formula Coefficients for Typical Road Conditions

Numerical values are based on empirical tire data. These values are typical sets of constant Magic Formula coefficients for common road conditions.



# Tire-Road Interaction (Magic Formula)

---

Surface	B	C	D	E
Dry tarmac	10	1.9	1	0.97
Wet tarmac	12	2.3	0.82	1
Snow	5	2	0.3	1
Ice	4	2	0.1	1

## Limitations

The Tire-Road Interaction (Magic Formula) block assumes longitudinal motion only and includes no camber, turning, or lateral motion.

## See Also

Tire (Magic Formula)

“Complete Vehicle Model”

# Tire (Simple)

**Purpose** No-slip tire model with minimal parameters

**Library** Tires & Vehicles

**Description** The block represents a simple, no-slip model of a tire parameterized by its radius. The model assumes an infinite torque transfer capability.

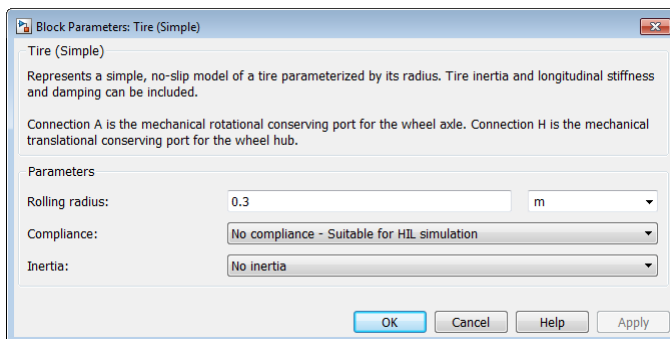


Connection A is a mechanical rotational conserving port associated with the axle of the tire. Connection H is a mechanical translational conserving port associated with the hub of the tire.

You can optionally add inertia, stiffness, or damping. These dynamics require additional computation and might make the model less suitable for real-time simulation.

## Assumptions and Limitations

- The tire does not slip.
- There is an infinite torque transfer capability.



## Dialog Box and Parameters

### Rolling radius

The rolling radius of the tire. The rolling radius must be greater than zero. Default is 0.3 m.

## Compliance

Specifies whether the model includes longitudinal stiffness and damping.

- **No compliance** - suitable for HIL simulation — (Default) Does not include longitudinal stiffness and damping.
- **Specify stiffness and damping** — Includes longitudinal stiffness and damping. Exposes the **Longitudinal stiffness** and **Longitudinal damping** parameters.

## Longitudinal stiffness

The longitudinal stiffness of the tire. The parameter must be greater than zero. Default is  $1e6$  N/m.

The **Longitudinal stiffness** parameter appears if the **Compliance** parameter is Specify stiffness and damping.

## Longitudinal damping

The longitudinal damping of the tire. The parameter must be greater than zero. Default is  $100$  N/(m/s).

The **Longitudinal damping** parameter appears if the **Compliance** parameter is Specify stiffness and damping.

## Inertia

Specifies whether the model includes tire inertia.

- **No inertia** — (Default) Does not include tire inertia.
- **Specify inertia and initial velocity** — Includes tire inertia and initial velocity. Exposes the **Tire inertia** and **Initial velocity** parameters.

## Tire inertia

The rotational inertia of the tire. The parameter must be greater than zero. Default is  $1$  kg\*/m<sup>2</sup>.

# Tire (Simple)

---

The **Tire inertia** parameter appears if the **Inertia** parameter is Specify inertia and initial velocity.

## Initial velocity

The initial rotational velocity of the tire. Default is 0 rad/s.

The **Initial velocity** parameter appears if the **Inertia** parameter is Specify inertia and initial velocity.

## Ports

Port	Description
A	Conserving mechanical rotational port associated with the axle of the tire
H	Conserving mechanical translational port associated with the hub of the tire

## Example

The `sdl_4wd_dynamics` example model shows the use of the Tire (Simple) block.

## See Also

[Tire \(Friction Parameterized\)](#)

[Tire \(Magic Formula\)](#)

[Tire-Road Interaction \(Magic Formula\)](#)

[“Complete Vehicle Model”](#)

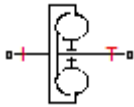
## Purpose

Hydrodynamic torque converter transferring torque between two driveshafts

## Library

Couplings & Drives

## Description



A torque converter couples two driveline axes, transferring torque and angular motion by the hydrodynamic action of a viscous fluid. Unlike a friction clutch, a torque converter cannot lock the axes together. The Torque Converter block acts between the two ports I and T. The block acts as a lookup function of the relative angular velocity of the two connected driveline axes. This function is defined at discrete angular velocities. For model details, see “Torque Converter Model” on page 1-404.

## Ports and Conventions

The impeller or pump port I and turbine port T are rotational conserving ports.

The Torque Converter block follows these conventions:

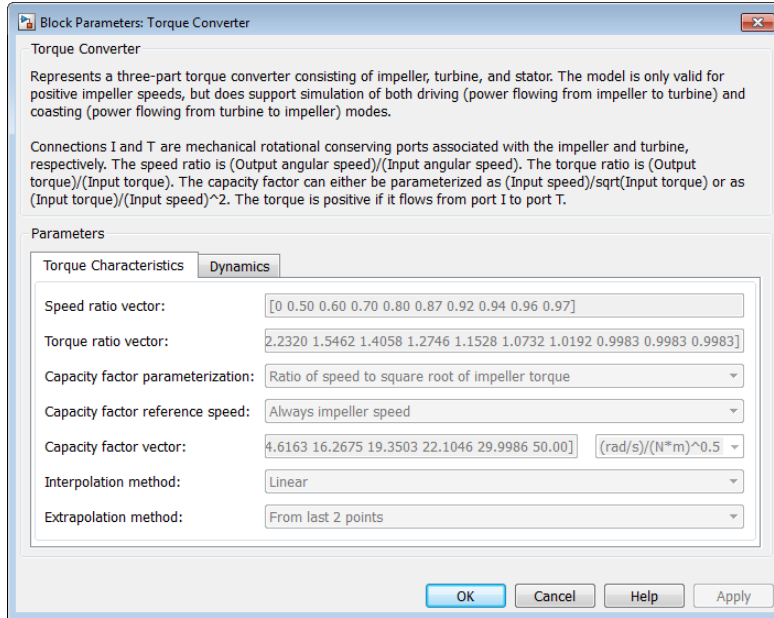
- The impeller port I is the port that connects to the engine, and the turbine port T is the port that connects to the load. In normal operation, power thus flows from the impeller to the turbine.
- Forward power flow implies power flowing from I to T. Reverse power flow implies power flowing from T to I.
- The power input is through the shaft with the larger speed. The power output is through the shaft with the smaller speed.

# Torque Converter

## Dialog Box and Parameters

The dialog box has one active area, **Parameters**, with two tabs.

### Torque Characteristics



### Speed ratio vector

Vector of values of the independent variable, the dimensionless speed ratio  $R_\omega$ . You must order these values in ascending order.

### Torque ratio vector

Vector of values of the block function's first dependent variable, the dimensionless torque ratio  $R_\tau$ . Each torque ratio value corresponds to a speed ratio value.

### Capacity factor parameterization

Definition of the capacity factor, either  $K$  (ratio of impeller speed  $\omega_1$  to square root of impeller torque  $\tau_1$ ) or  $K^*$  (ratio of  $\tau_1$  to  $\omega_1^2$ ). The default is  $K$ .

## Capacity factor reference speed

Choice of speed in the capacity factor definition, depending on speed ratio  $R_\omega$ . Select either:

- Impeller speed  $\omega_I$  for all values of  $R_\omega$ .
- Impeller speed  $\omega_I$  for  $R_\omega < 1$ , and turbine speed  $\omega_T$  for  $R_\omega > 1$ .

## Capacity factor vector

Vector of values of the block function's second dependent variable, the torque conversion capacity factor  $K$ . Each capacity factor value corresponds to a speed ratio value.

From the drop-down list, choose units.

- If you choose the default capacity factor definition  $K$ , the default units are radians/second/ $\sqrt{\text{newton-meters}}$  ( $\text{rad/s} / (\text{N}\cdot\text{m})^{0.5}$ ).
- If you choose the alternative capacity factor definition  $K^*$ , the default units are newton-meters/(radians/second)<sup>2</sup> ( $\text{N}\cdot\text{m} / (\text{rad/s})^2$ ).

## Interpolation method

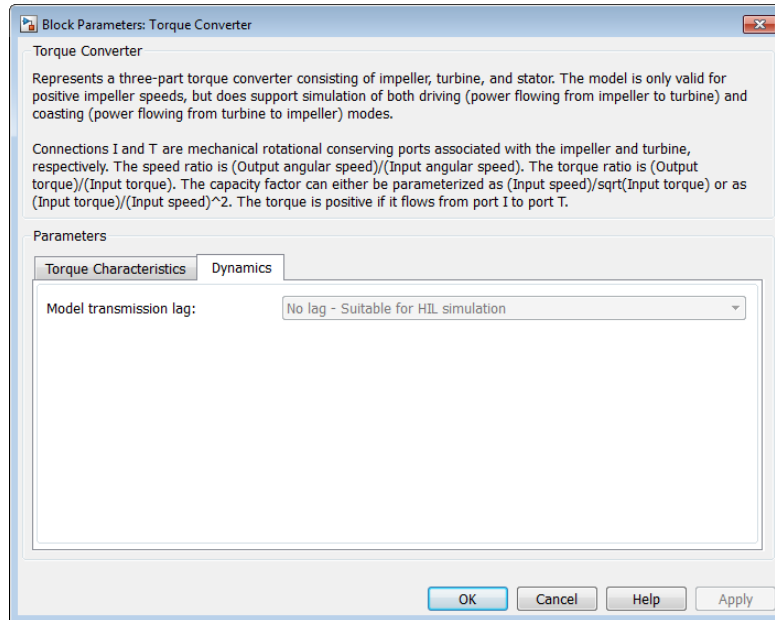
Interpolates torque ratio and capacity factor functions between discrete relative velocity values within the range of definition. The default is Linear.

## Extrapolation method

Extrapolates torque ratio and capacity factor functions outside the range of definition. The default is From last 2 points.

# Torque Converter

## Dynamics



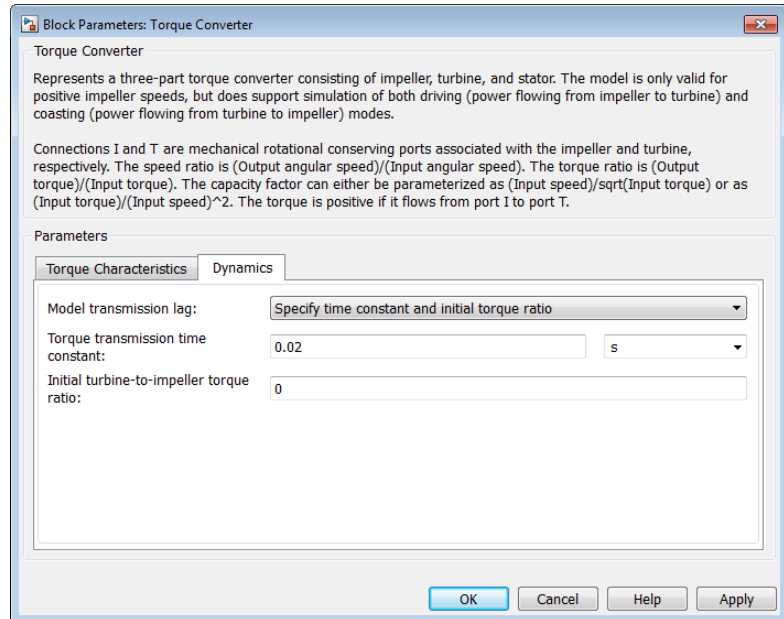
### Model transmission lag

Select how to model transmission lag from input to output driveshaft. The default is **No lag**.

- **No lag** Suitable for HIL simulation — Torque transfer is instantaneous.
- **Specify time constant and initial torque ratio** — Torque is transferred with a time lag. If you select this option, the panel changes from its default.



## Time Constant and Initial Torque Ratio



### Torque transmission time constant

Dynamic torque transfer time constant  $t_c$ . The default is 0.02.

From the drop-down list, choose units. The default is seconds (s).

### Initial turbine-to-impeller torque ratio

Ratio of turbine torque  $\tau_T$  to impeller torque  $\tau_I$  at the start of the simulation.

# Torque Converter

---

## Torque Converter Model

Torque Converter is a mechanism for transferring motion and torque between impeller and turbine. Because the coupling of I and T occurs by viscous action, the torque transfer depends on the difference  $\omega = \omega_T - \omega_I \neq 0$ , or the speed ratio  $R_\omega \neq 1$ . In normal operation, the two axes have different speeds, and the output axis speed never exactly reaches the input axis speed. The torque transfer is largest when  $R_\omega \rightarrow 0$  or  $\infty$ , and shrinks as  $R_\omega \rightarrow 1$ . Because  $R_\omega$  can never reach exactly one, a torque converter always transfers some torque.

### Speed Ratio, Torque Ratio, and Capacity Factor

You specify the torque ratio and the capacity factor of the torque converter as discrete functions of the speed ratio with tabular vector entries. The three vectors of the variable values must have the same length.

- The speed ratio  $R_\omega$  is the turbine angular speed divided by the impeller angular speed:

$$R_\omega = \omega_T / \omega_I .$$

- The torque ratio  $R_\tau$  is the output (turbine) torque divided by the input (impeller) torque:

$$R_\tau = \tau_T / \tau_I .$$

- The capacity factor  $K$  or  $K^*$  is defined in two ways for  $R_\omega < 1$ , with either the default or the alternative definition:

- Default, the input speed divided by the square root of the input torque:

$$K = \omega_I / \sqrt{\tau_I} .$$

- Alternative, the input torque divided by the square of the input speed:

$$K^* = \tau_I / \omega_I^2 .$$

- The capacity factor reference speed for  $R_\omega > 1$  is  $\omega_I$  by default. That is, the input speed  $\omega_I$  is used in the ratio that defines either  $K$  or  $K^*$ .

For  $R_\omega > 1$ , the alternative choice for reference speed is to replace  $\omega_I$  by the output speed  $\omega_T$  in this defining ratio.

The two dependent variables,  $R_\tau$  and  $K$ , are functions of the independent variable  $R_\omega$ . They specify the characteristics of the torque converter:

$$R_\tau = R_\tau(R_\omega) , K = K(R_\omega) .$$

### Continuation Near Speed Ratio of Unity

If  $R_\omega$  falls outside the specified range during simulation, the torque ratio and capacity ratio functions are extrapolated. The extrapolation must satisfy the requirement that  $\tau_{out}$  be zero when  $R_\omega$  is 1. To meet this requirement, the block uses a mathematical extension that starts with the last  $R_\tau$  value below 1, which is taken to be the locking point.  $\tau_I$  is extrapolated with:

$$\tau_I = \text{sgn}(1 - \omega_T / \omega_I) \cdot [\omega_I / K(L)]^2 \cdot \sqrt{[|1 - R_\omega| / (1 - R_\omega(L))]} .$$

# Torque Converter

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$R_\omega(L)$  and  $K(L)$  are, respectively, the speed ratio and capacity factor at the locking point.

If you provide speed ratio data for  $R_\omega > 1$ , a similar interpolation determines  $\tau_I$  for the range defined by the two speed ratio values closest to either side of 1. If you provide a point where  $R_\omega = 1$ , the corresponding  $K$  value is not used. However, at such a point, the value used for  $R_\omega$  is always determined directly from the tabulated data.

## Instantaneous Torque Transfer

When there is no time lag, the input impeller (I) and output turbine (T) torques are:

$$\tau_I = \text{sgn}(1 - \omega_T/\omega_I) \cdot [\omega_I / K]^2, \tau_T = \tau_I \cdot R_\omega,$$

in normal operation (forward power flow).

## Lagged Torque Transfer

You can optionally include the effect of torque transmission time lag, caused by internal fluid flow and compressibility. Instead of  $\tau_T$  and  $\tau_I$  being instantaneously constrained to one another, a first-order time lag introduces a delayed response in the impeller torque:

$$t_c \cdot (d\tau_I/dt) + \tau_I = \tau_I(\text{steady state}).$$

The preceding instantaneous function of the capacity factor  $K$  determines the steady-state value of  $\tau_I$ .

## Limitations

The impeller shaft must always rotate in a positive direction. Simulation is not valid for  $\omega_I < 0$ .

If you drive the Torque Converter from a torque source, such as the Generic Engine, you must include an inertia in the source, to represent the engine, shaft inertia, or other source components. You must set the initial speed for this inertia to a positive value to ensure that the impeller starts by rotating in a positive direction.

## Torque Transmission Time Lag

Torque converters lag in their response to changing input torque. By default, Torque Converter includes no time lag in its response. You can include a response lag by specifying a time constant. Time lag simulation increases model fidelity but reduces simulation performance. See “Adjust Model Fidelity”.

### Examples

These SimDriveline example models include torque converters as part of complete drivetrains:

- `sdl_4wd_dynamics`
- `sdl_vehicle`

### Reference

Society of Automotive Engineers, *Hydrodynamic Drive Test Code (Surface Vehicle Recommended Practice)*, SAE J643, May 2000.

### See Also

Disk Friction Clutch, Fundamental Friction Clutch, Generic Engine  
“Complete Vehicle Model”

# Torsional Spring-Damper

---

**Purpose** Rotational spring and damper coupling, with Coulomb friction, locking, and hard stops

**Library** Couplings & Drives

**Description** The Torsional Spring-Damper block represents a dynamic element that imposes a combination of internally generated torques between the two connected driveshaft axes, the rod and the case. The complete torque includes these components:



- Linear damped spring
- Coulomb friction (including locking static friction)
- Hard stop compliance

The second and third components are optional. For model details, see “Torsional Spring-Damper Model” on page 1-413.

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**Note** Torsional Spring-Damper is based on the Loaded-Contact Rotational Friction block and the Simscape Rotational Spring, Rotational Damper, and Rotational Hard Stop blocks. (The first and fourth blocks are required for Coulomb friction and hard stops, respectively.) For more information, see these block reference pages.

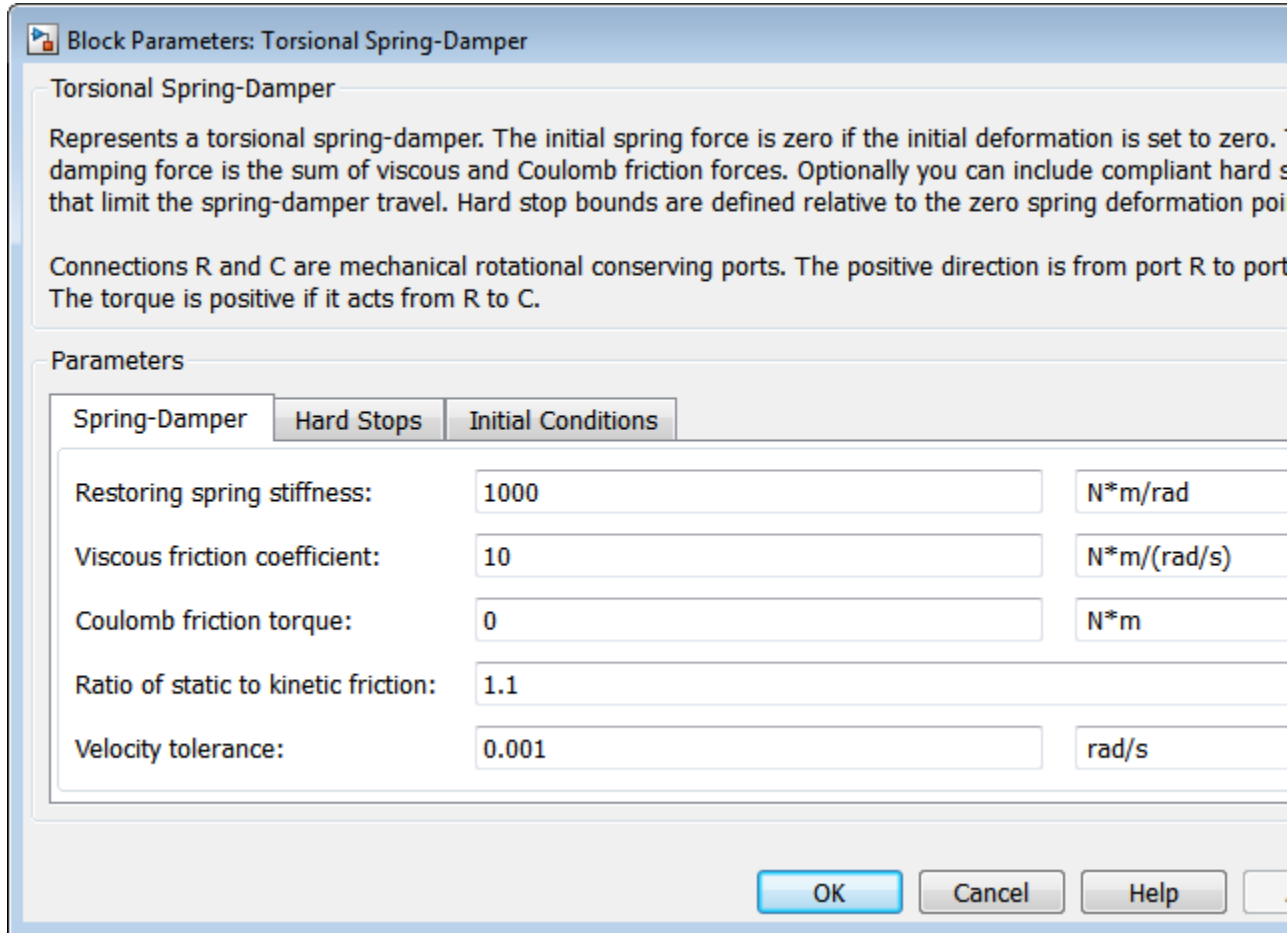
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## Ports

R and C are rotational conserving ports representing, respectively, the rod and case driveshafts. The relative motion is measured as  $\omega = \omega_R - \omega_C$ , the angular velocity of rod relative to case.

# Torsional Spring-Damper

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

# Torsional Spring-Damper

---

## Spring-Damper

### Restoring spring stiffness

Torsional spring stiffness  $k$  acting between connected driveshafts. Must be greater than zero. The default is 1000.

From the drop-down list, choose units. The default is newton-meters/radian ( $\text{N}\cdot\text{m}/\text{rad}$ ).

### Viscous friction coefficient

Torsional damping  $\mu$  acting between the connected driveshafts. Must be greater than or equal to zero. The default is 10.

From the drop-down list, choose units. The default is newton-meters/(radian/second) ( $\text{N}\cdot\text{m}/(\text{rad}/\text{s})$ ).

### Coulomb friction torque

Constant kinetic friction torque  $\tau_K$  acting between connected driveshafts. Must be greater than or equal to zero. The default is 0.

From the drop-down list, choose units. The default is newton-meters ( $\text{N}\cdot\text{m}$ ).

### Ratio of static to kinetic friction

Constant ratio  $R$  of static Coulomb friction torque  $\tau_S$  to kinetic Coulomb friction torque  $\tau_K$  acting between connected driveshafts. Must be greater than one. The default is 1.1.

### Velocity tolerance

Minimum relative angular speed  $\omega_{\text{Tol}}$  below which the two connected driveshafts can lock and rotate together. Must be greater than zero. The default is 0.001.

From the drop-down list, choose units. The default is radians/second ( $\text{rad}/\text{s}$ ).



## Hard Stops

The screenshot shows a software interface with three tabs: "Spring-Damper", "Hard Stops", and "Initial Conditions". The "Hard Stops" tab is active. Below the tabs, there is a label "Hard stop model:" followed by a dropdown menu. The dropdown menu is open, showing the selected option "No hard stops - Suitable for HIL simulation".

### Hard stop model

Select how to model the hard stops. The default is No hard stops Suitable for HIL simulation.

- No hard stops — Do not include hard stops in relative motion of connected driveshafts.

### No hard stops

Hard stop model:

- Compliant hard stops — Model friction geometry in terms of annulus dimensions. If you select this option, the panel changes from its default.

### Compliant hard stops

Hard stop model:

Upper bound:

Lower bound:

Contact stiffness:

Contact damping:

# Torsional Spring-Damper

---

## Upper bound

Upper hard stop angular displacement  $\delta_+$  from the zero-torque reference angle  $\varphi = 0$ . Must be greater than  $\delta_-$ . The default is 10.

From the drop-down list, choose units. The default is degrees (deg).

## Lower bound

Lower hard stop angular displacement  $\delta_-$  from the zero spring force reference angle  $\varphi = 0$ . Must be less than  $\delta_+$ . The default is -10.

From the drop-down list, choose units. The default is degrees (deg).

## Contact stiffness

Hard stop stiffness  $k_{HS}$  applied if the relative angle  $\varphi$  moves into the hard stop region. Must be greater than zero. The default is 1e+6.

From the drop-down list, choose units. The default is newton-meters/radian (N\*m/rad).

## Contact damping

Hard stop damping  $\mu_{HS}$  applied if the relative angle  $\varphi$  moves into the hard stop region. Must be greater than zero. The default is 10.

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Initial Conditions

The screenshot shows a software interface with three tabs: 'Spring-Damper', 'Hard Stops', and 'Initial Conditions'. The 'Initial Conditions' tab is active. It contains a label 'Initial deformation:' followed by a text input field containing the number '0' and a dropdown menu currently showing 'deg'.

### Initial deformation

Initial deformation of the torsional spring relative to the zero-torque reference angle  $\varphi = 0$ . The default is 0.

From the drop-down list, choose units. The default is degrees (deg).

## Torsional Spring-Damper Model

The complete torque  $\tau$  imposed by Torsional Spring-Damper between the connected driveshafts is the sum of three terms: stiff-damping, hard stop compliance, and Coulomb.

$$\tau = \tau_{SD} + \tau_{HS} + \tau_C .$$

The table summarizes the torsional spring-damper variables.

### Torsional Spring-Damper Variables

Symbol	Definition	Significance
$\varphi$	Relative angle between ring and hub	Relative angular position of ring and hub
$\omega$	Relative angular velocity	$\omega = \omega_R - \omega_C$
$k$	Torsional stiffness	<i>See the following model</i>

# Torsional Spring-Damper

## Torsional Spring-Damper Variables (Continued)

Symbol	Definition	Significance
$\mu$	Torsional damping	<i>See the following model</i>
$\delta_+, \delta_-$	Upper and lower hard stop angular displacements	<i>See the following model</i>
$k_{HS}$	Contact stiffness applied in hard stop regions	<i>See the following model</i>
$\mu_{HS}$	Contact damping applied in hard stop regions	<i>See the following model</i>
$\tau_K$	Kinetic friction	Constant sliding Coulomb friction
$\tau_S$	Static friction	Constant locking Coulomb friction
$R$	$\tau_S/\tau_K$	Ratio of static to kinetic Coulomb friction
$\omega_{Tol}$	Maximum relative speed for clutch locking	<i>See the following model</i>

### Stiffness and Damping

The stiff-damping torque is a simple linear spring-damping:

$$\tau_{SD} = -k\varphi - \mu\omega .$$

### Hard Stops

The hard stop torque is applied if  $\varphi$  moves outside the angular gap between the upper and lower hard stop bounds.

HS	Range
$-k_{\text{HS}}(\varphi - \delta_+) - \mu_{\text{HS}}\omega$	$\varphi > \delta_+$
0	$\delta_- < \varphi < \delta_+$
$-k_{\text{HS}}(\varphi - \delta_-) - \mu_{\text{HS}}\omega$	$\varphi < \delta_-$

## Coulomb Friction – Locking

The Coulomb friction torque is a constant  $\tau_K$  if  $\omega$  is nonzero (unlocked), and a constant  $\tau_S$  if  $\omega$  is zero (locked).

$$\tau_S = R\tau_K .$$

## Locking Conditions

The Torsional Spring-Damper locks the connected driveshafts together if both:

- $|\omega| < \omega_{\text{Tol}}$
- The torque across the torsional spring-damper is less than  $\tau_S$ .

If the clutch locks,  $\omega$  is reset to zero. If the torque across the torsional spring-damper exceeds  $\tau_S$ , the driveshafts unlock from one another, and  $\omega$  becomes nonzero.

## Limitations

Hard stop and Coulomb friction simulation increases model fidelity but reduces simulation performance. See “Adjust Model Fidelity”.

## See Also

Dog Clutch, Loaded-Contact Rotational Friction, Rotational Damper, Rotational Hard Stop, Rotational Spring

“Gear Coupling Control with Clutches”

# Translational Detent

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**Purpose** Double-sided, spring-loaded translational detent

**Library** Brakes & Detents/Translational

**Description**



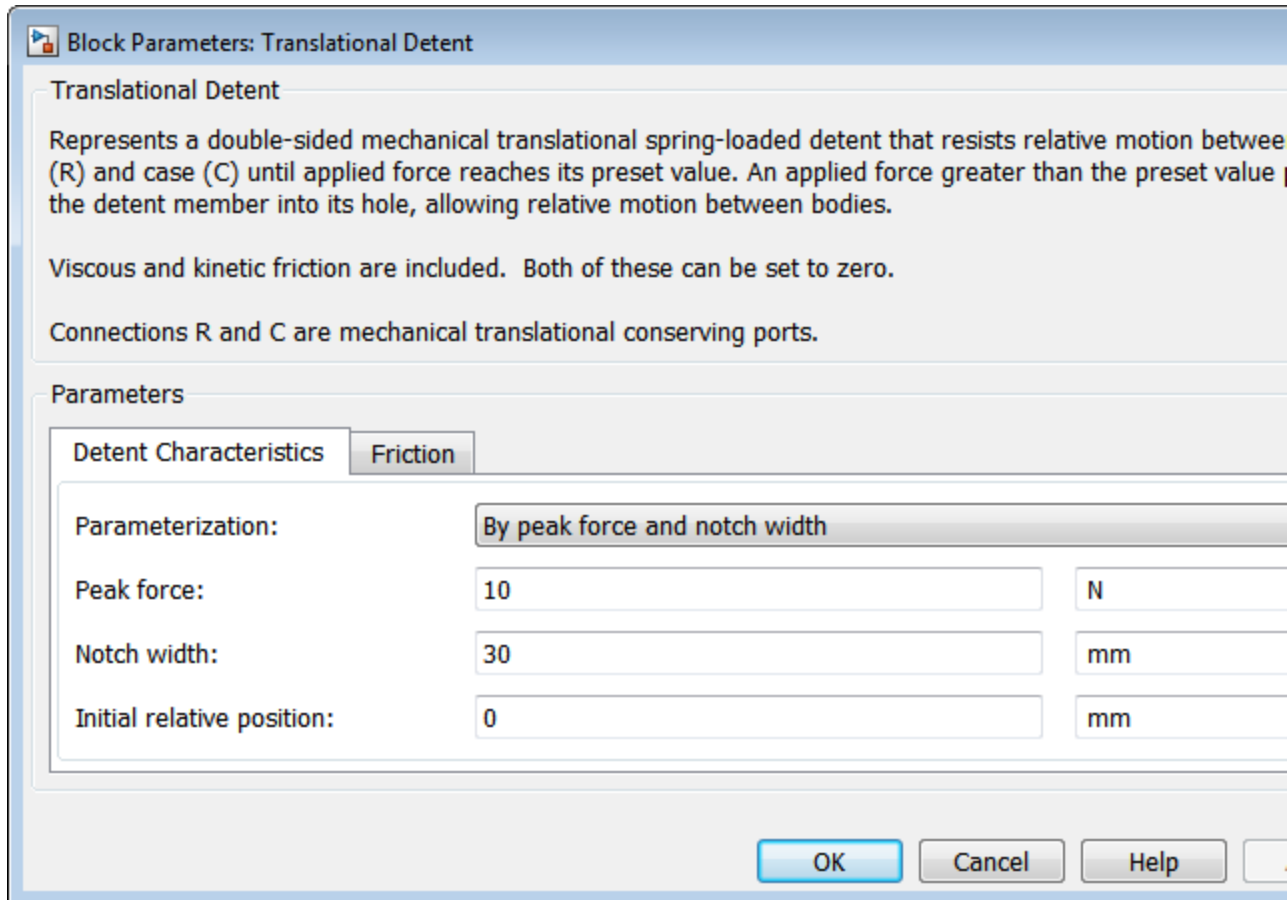
The Translational Detent block represents a double-sided, spring-loaded translational detent with a ball and conical notch.

The detent slider slides horizontally over the detent case. These two parts develop a horizontal shear force between them. The inside of the slider is the conical notch. The detent case contains a vertical spring. The detent ball lies between the spring and the conical notch. With the block, you can model ball and angled notch detents with geometric and spring characteristics, peak force and notch width, or a lookup table. The model also includes viscous damping and kinetic friction between the slider and case. You can set the friction to zero.

**Ports**

R and C are translational conserving ports associated with the slider and case, respectively.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with two tabs.

# Translational Detent

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## Detent Characteristics

### Parameterization

Select how to specify the detent characteristics. The default is By peak force and notch width.

- By peak force and notch width — Specify detent characteristics by the peak shear force and the notch width.

### Peak Force and Notch Width

#### Peak force

The peak shear force developed inside the detent. The default is 10.

From the drop-down list, choose units. The default is newtons (N).

#### Notch width

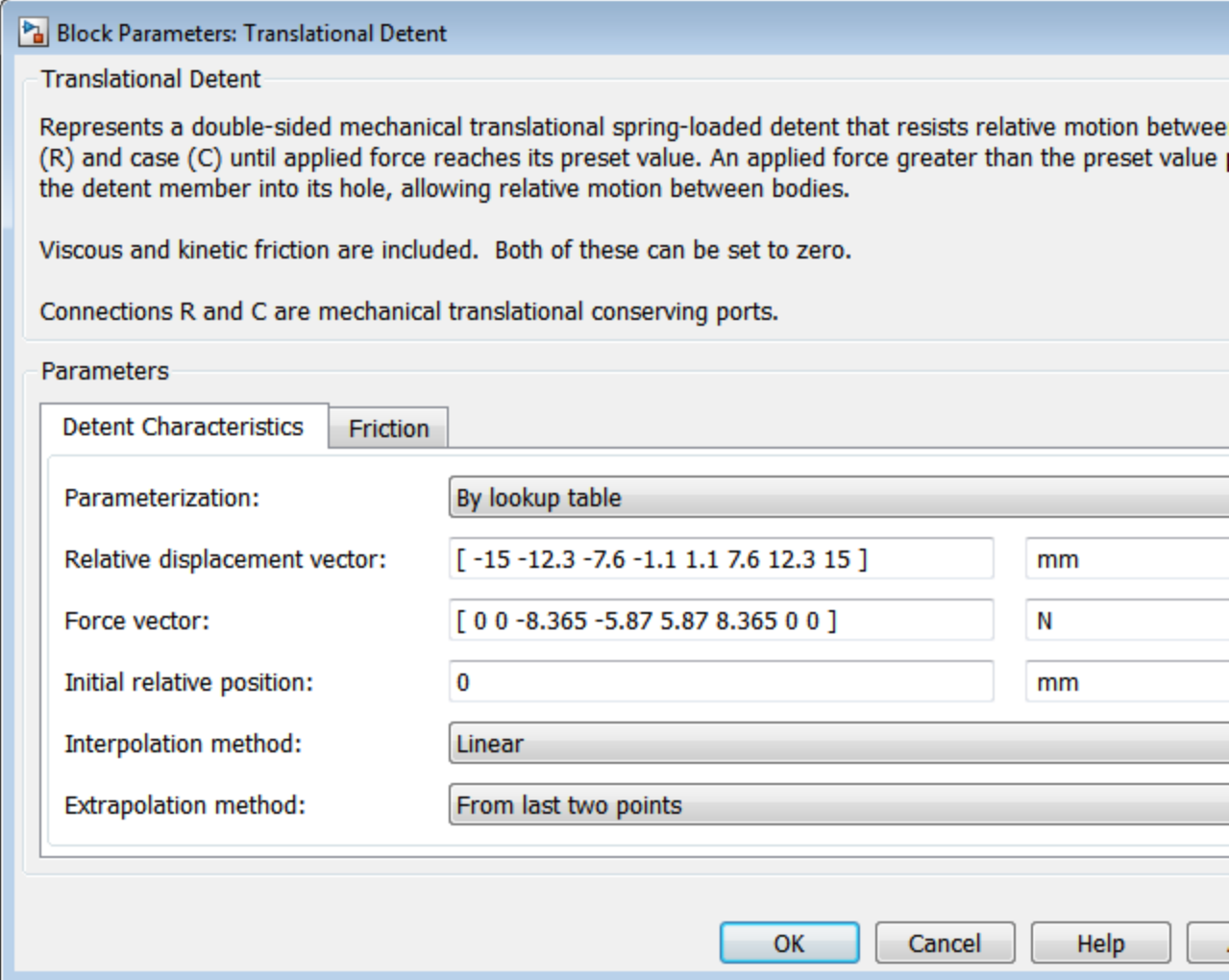
The width of the region where the shear force develops. The default is 30.

From the drop-down list, choose units. The default is millimeters (mm).

- By lookup table — Define the detent characteristics by one-dimensional table lookup based on the relative displacement between the slider and case. If you select this option, the panel changes from its default option.



## Lookup Table

The image shows a software dialog box titled "Block Parameters: Translational Detent". It contains a description of the block, a "Parameters" section with two tabs ("Detent Characteristics" and "Friction"), and several input fields for defining the detent's behavior. The "Friction" tab is currently selected. The input fields include a parameterization dropdown set to "By lookup table", a relative displacement vector field containing the values [-15 -12.3 -7.6 -1.1 1.1 7.6 12.3 15] with a unit of mm, a force vector field containing [0 0 -8.365 -5.87 5.87 8.365 0 0] with a unit of N, an initial relative position field set to 0 with a unit of mm, an interpolation method dropdown set to "Linear", and an extrapolation method dropdown set to "From last two points". At the bottom right, there are buttons for "OK", "Cancel", and "Help".

**Block Parameters: Translational Detent**

**Translational Detent**

Represents a double-sided mechanical translational spring-loaded detent that resists relative motion between (R) and case (C) until applied force reaches its preset value. An applied force greater than the preset value pushes the detent member into its hole, allowing relative motion between bodies.

Viscous and kinetic friction are included. Both of these can be set to zero.

Connections R and C are mechanical translational conserving ports.

**Parameters**

Detent Characteristics    **Friction**

Parameterization: By lookup table

Relative displacement vector: [ -15 -12.3 -7.6 -1.1 1.1 7.6 12.3 15 ] mm

Force vector: [ 0 0 -8.365 -5.87 5.87 8.365 0 0 ] N

Initial relative position: 0 mm

Interpolation method: Linear

Extrapolation method: From last two points

OK    Cancel    Help

## Relative displacement vector

Specify the vector of input relative displacements that correspond to the values in **Force vector**. These two vectors must be of the same size. The displacements must be specified in strictly increasing order. The default values are [ -15 -12.3 -7.6 -1.1 1.1 7.6 12.3 15].

From the drop-down list, choose units. The default is millimeters (mm).

## Force vector

Specify the vector of input values for the shear force that correspond to the values in **Relative displacement vector**. The default values are [0 0 -8.365 -5.87 5.87 8.365 0 0].

From the drop-down list, choose units. The default is newtons (N).

## Interpolation method

Select one of the following interpolation methods for approximating the output value when the input value is between two consecutive grid points:

- **Linear** — Uses a linear interpolation function.
- **Cubic** — Uses the Piecewise Cubic Hermite Interpolation Polynomial (PCHIP).
- **Spline** — Uses the cubic spline interpolation algorithm.

For more information on interpolation algorithms, see the PS Lookup Table (1D) block reference page.

## Extrapolation method

Select one of the following extrapolation methods for determining the output value when the input value is outside the range specified in the argument list:

- **From last 2 points** — Extrapolates using the linear method (regardless of the interpolation method specified), based on the last two output values at the appropriate end of the range. If the input value is below the specified range, the block uses the first and second specified output values. If the input value is above the specified range, the block uses the two last specified output values.
- **From last point** — Uses the last specified output value at the appropriate end of the range. The block uses the last specified output value for all input values greater than the last specified input argument. The block uses the first specified output value for all input values less than the first specified input argument.

For more information on extrapolation algorithms, see the PS Lookup Table (1D) block reference page.

- **By geometry** — Define the detent characteristics by ball-notch geometry and dynamics. If you select this option, the panel changes from its default option.

# Translational Detent

## Geometric and Spring Parameters

**Block Parameters: Translational Detent**

**Translational Detent**

Represents a double-sided mechanical translational spring-loaded detent that resists relative motion between rotor (R) and case (C) until applied force reaches its preset value. An applied force greater than the preset value pushes the detent member into its hole, allowing relative motion between bodies.

Viscous and kinetic friction are included. Both of these can be set to zero.

Connections R and C are mechanical translational conserving ports.

**Parameters**

Detent Characteristics    Friction

Parameterization:	By geometry	
Ball diameter:	12	mm
Notch angle:	120	deg
Notch depth:	4.5	mm
Preload force:	15	N
Spring stiffness:	3e+3	N/m
Shear reversal region:	0.1	mm
Initial relative position:	0	mm

OK    Cancel    Help    Apply

## Ball diameter

Specify the diameter of the ball. The default is 12.

From the drop-down list, choose units. The default is millimeters (mm).

## Notch angle

Specify the full angle of the notch. Must be greater than 0 and less than or equal to 179 degrees. The default is 120.

From the drop-down list, choose units. The default is degrees (deg).

## Notch depth

Specify the depth of the notch in the slider. Must be greater than zero. The default is 4.5.

From the drop-down list, choose units. The default is millimeters (mm).

## Preload force

Specify the force to which the detent spring is preloaded. Must be greater than or equal to 0. The default is 15.

From the drop-down list, choose units. The default is newtons (N).

## Spring stiffness

Specify the stiffness of the detent spring. Must be greater than 0. The default is 3000.

From the drop-down list, choose units. The default is newtons/meter (N/m).

## Shear reversal region

Specify the width of the region where the ball moves from one notch face to the other. Must be greater than 0. The default is 0.1.

# Translational Detent

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From the drop-down list, choose units. The default is millimeters (mm).

## **Initial relative position**

The relative displacement of the slider and case when simulation starts. The default is 0.

From the drop-down list, choose units. The default is millimeters (mm).

## Friction

Block Parameters: Translational Detent

**Translational Detent**

Represents a double-sided mechanical translational spring-loaded detent that resists relative motion between (R) and case (C) until applied force reaches its preset value. An applied force greater than the preset value pushes the detent member into its hole, allowing relative motion between bodies.

Viscous and kinetic friction are included. Both of these can be set to zero.

Connections R and C are mechanical translational conserving ports.

**Parameters**

Detent Characteristics	Friction
Viscous friction coefficient:	<input type="text" value="0.1"/> N/(m/s)
Friction to peak force ratio:	<input type="text" value="0.01"/>
Friction velocity threshold:	<input type="text" value="0.05"/> m/s

OK Cancel Help

# Translational Detent

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## **Viscous friction coefficient**

Specify the viscous friction coefficient for the ball-notch contact. Must be greater than or equal to 0. The default is 0.1.

From the drop-down list, choose units. The default is newtons/(meters/second) (N/ (m/s)).

## **Friction to peak force ratio**

Specify the kinetic friction coefficient for the ball-notch contact. Must be greater than or equal to 0. The default is 0.01.

The kinetic friction is this ratio multiplied by the peak shear force.

## **Friction velocity threshold**

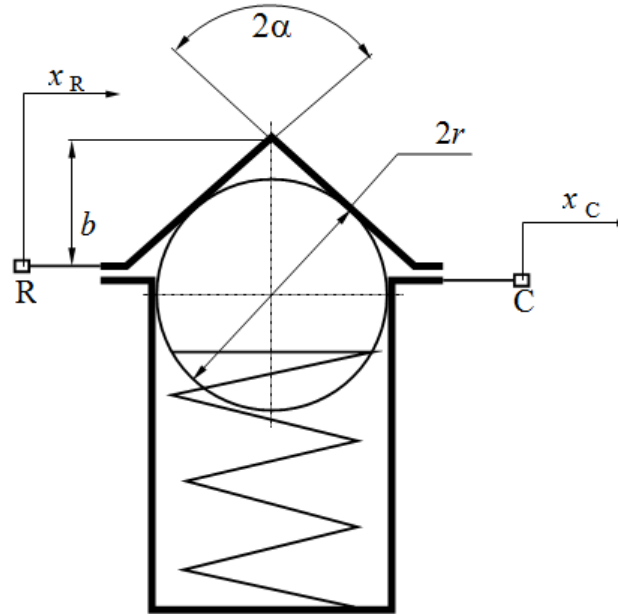
Specify the relative velocity required for peak kinetic friction in the detent. Must be greater than 0. The default is 0.05.

From the drop-down list, choose units. The default is meters/second (m/s).



## Translational Detent Model

The geometry of the ball-notch detent is shown in the following figure. As the conical slider slides horizontally over the case, the relative displacement  $x = x_R - x_C$  causes a horizontal shear force  $F$  to develop.



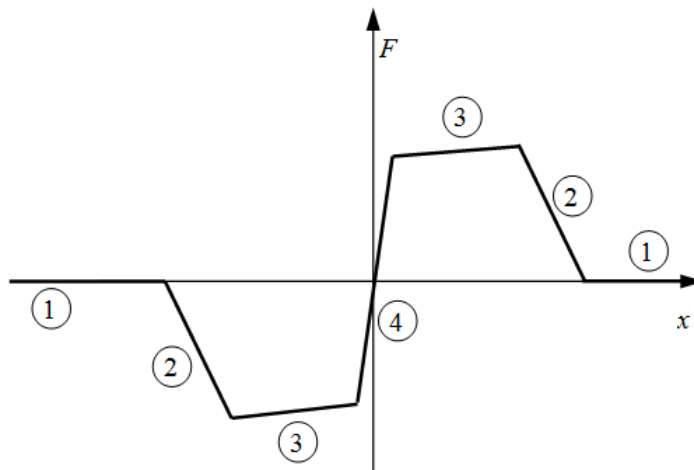
- $b$  Notch depth
- $r$  Ball radius
- $a$  Notch half angle

Depending on your choice of parameterization, the shear force model is defined by geometric and spring parameters, by the peak force and notch width, or by a lookup table specifying relative displacement versus force.

# Translational Detent

## Geometric and Spring Parameters

The geometric and spring parameterization uses four regions to define the shear force. The following figure displays these regions.

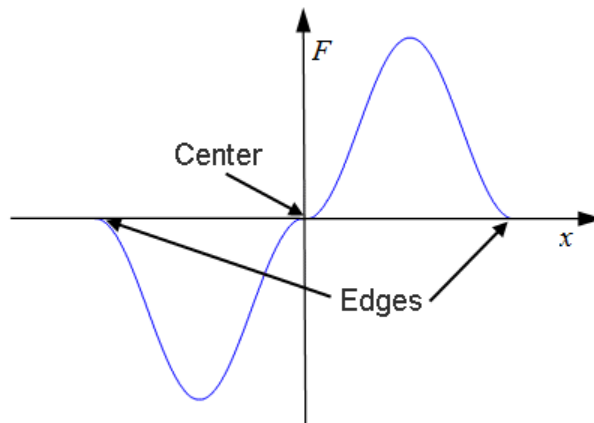


Because the ball is spherical and the notch is symmetric, the horizontal force versus displacement function is symmetric about the origin. (See “Peak Force and Notch Width” on page 1-429.)

Region	Ball Position and Contact Angle	Shear Force
1	Ball is outside conical notch. Contact angle is vertical.	Viscous damping and kinetic friction only. No spring contribution.
2	Ball is entering notch, in contact with notch corner. Contact angle rotates from vertical.	Spring, viscous damping, and kinetic friction contributions. Peaks at transition from region 2 to region 3.
3	Ball slides along face of notch. Contact angle is constant.	Force decreases as spring extends, but does not go to zero (spring preload force).
4	Ball moves from one face to the other. Contact angle switches direction.	Force switches direction. Shear reversal region width controls how smooth this switch is.

## Peak Force and Notch Width

If you choose the peak force and notch width parameterization, the block ensures that the force-relative displacement curve provides a continuous force and force derivative over the detent region. The peak forces are halfway between the detent center and detent edge, as shown in the following figure.



## Lookup Table

With the lookup table parameterization, you can create an arbitrary function relating shear force to relative displacement. If you create such a function, consider the following best practices.

### Energy Conservation

If you want to ensure that the detent conserves energy, the total integral of the force-relative displacement curve (area under the curve) must be zero.

### Simulation Performance

To stabilize simulation of the detent, avoid discontinuities in the force-relative displacement function. The most important requirement is to have a shear reversal region of nonzero width, analogous to region 4 in the geometric parameterization.

# Translational Detent

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## **Limitations**

The model does not account for inertia. Add mass terms externally to the R and C ports as required.

If you use the peak force-notch width or the lookup table parameterization, the kinetic friction is independent of the detent normal force.

## **Example**

The `sdl_stepping_mechanism_with_detents` example model shows the use of the Translational Detent block.

## **See Also**

Loaded-Contact Translational Friction, Rotational Detent  
“Clutches, Clutch-Like Elements, and Coulomb Friction”

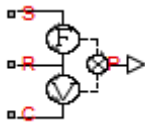
## Purpose

Mechanical sensor used to measure average or instantaneous translational power

## Library

Sensors

## Description



The block represents a sensor that you can use to measure the translational power change due to one or more driveline components connected between ports R and C. The power output can be an instantaneous or average value. Two averaging options are available: period and vibration averaging.

The instantaneous power is the product of the current velocity and force. Use the instantaneous power to analyze peak power transmission and identify heavily-loaded components.

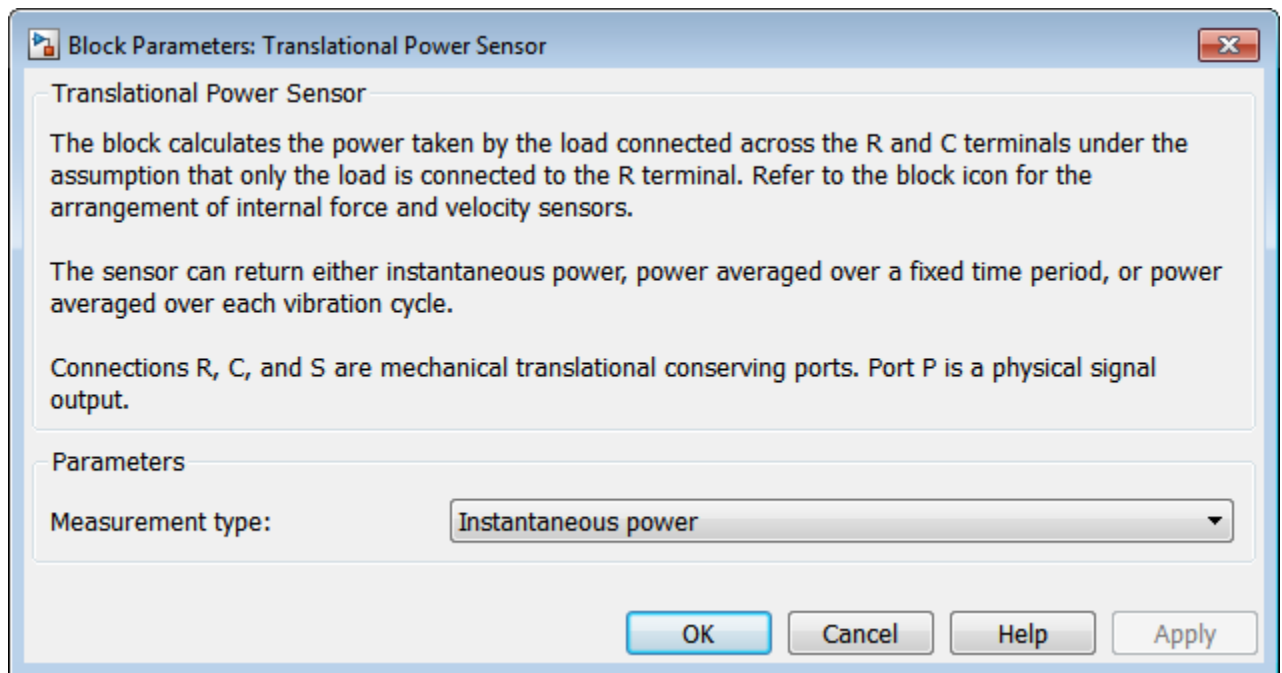
The time-averaged power is the integral of the instantaneous power over one sample period, divided by that period. For simulation times smaller than the sample period, the average power is zero. The power is constant for each sample period.

The vibration-averaged power is the integral of the power over one vibration period, divided by that period. The vibration period is the time between two consecutive crossings from negative to positive velocity values. Use this averaging method only for oscillations with velocity passing through zero. The averaging period can change if the vibration has a variable frequency. For simulation times smaller than the vibration period, the average power is zero. The power output is constant for each vibration period.

## Dialog Box and Parameters

The dialog box contains one active area, **Parameters**.

# Translational Power Sensor

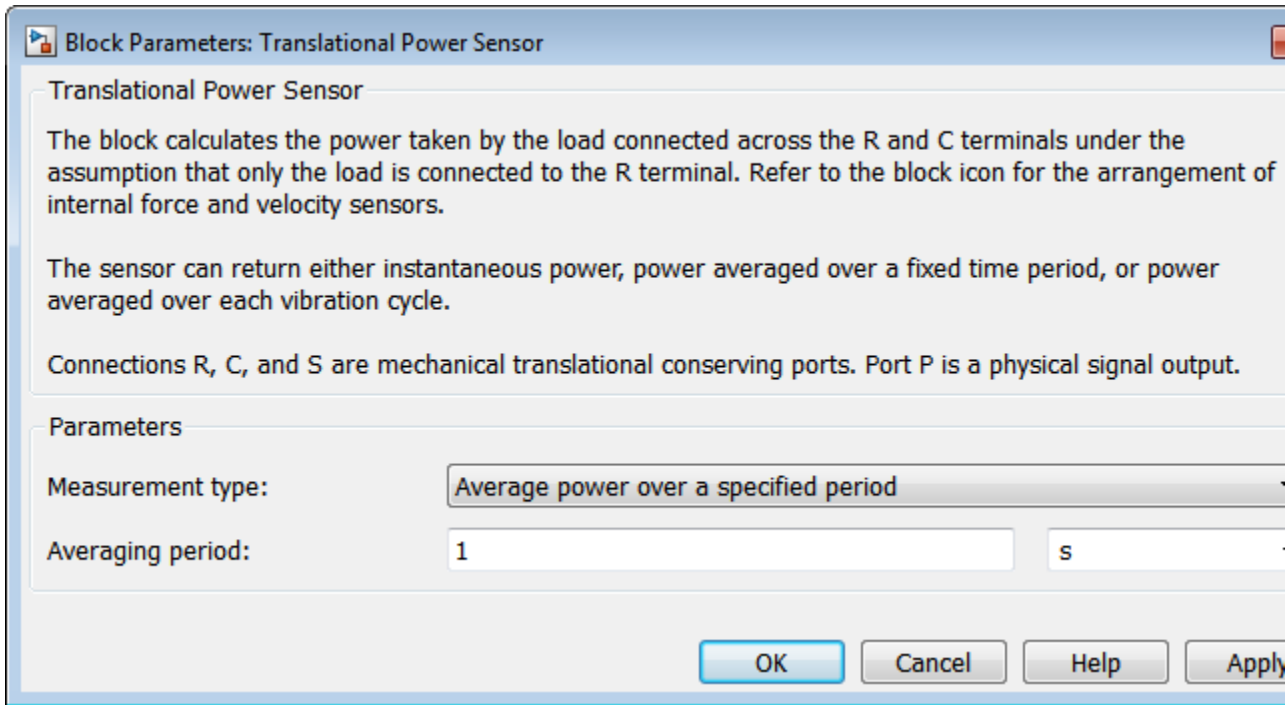


## Measurement Type

Select either instantaneous or average power measurement. The default type is Instantaneous Power.

- Instantaneous power — Output the power measured for the current time step.
- Average power over a specified period— Output the power averaged over a fixed time period.

# Translational Power Sensor



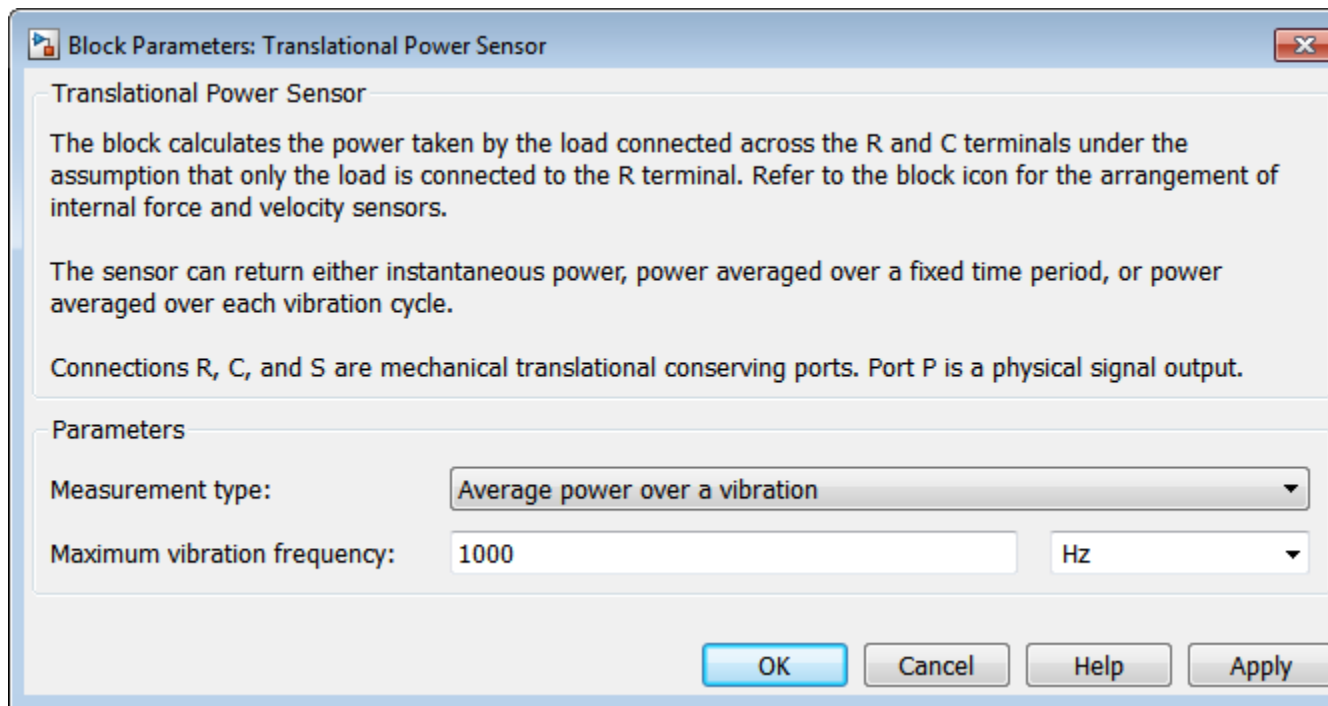
## Averaging period

Enter the time period used to compute the average power. Typical values include the cycle length of a reciprocating device or the actuation force.

The default value is 1. The default unit is s.

- Average power over a vibration— Output the power averaged over one vibration period.

# Translational Power Sensor



## Maximum vibration frequency

Enter the maximum vibration frequency over which the power should be averaged. Very high values can reduce simulation speed. Very low values can result in lower power measurements.

The default value is 1000. The default unit is Hz.

## Ports

Port	Description
P	Physical signal output port for translational power
S	Rotational conserving port associated with driving source



# Translational Power Sensor

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Port	Description
C	Rotational conserving port associated with loaded driveline component(s)
R	Rotational conserving port associated with case holding loaded driveline component(s)

## See Also

Rotational Power Sensor

# Unidirectional Clutch

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**Purpose** Clutch transmitting torque in one direction only

**Library** Clutches

**Description**

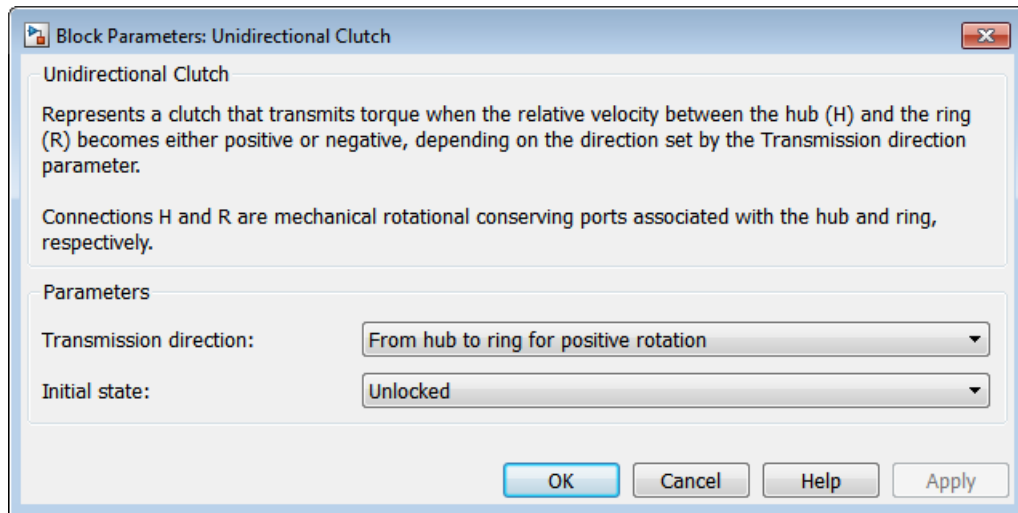


The Unidirectional Clutch block models an ideal one-way clutch based on a coupled ring-hub. It transmits torque when the relative velocity of hub and ring becomes positive or negative, depending on how you set the transmission direction. For model details, see “Unidirectional Clutch Model” on page 1-437.

**Ports**

H and R are rotational conserving ports representing, respectively, the hub and ring driveshaft axes. The clutch motion is measured as the *slip*  $\omega = \omega_H - \omega_R$ , the angular velocity of hub relative to ring.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**.

### Transmission direction

From the drop-down list, select the relative rotation direction (sign of  $\omega$ ) for which the clutch locks and transfers torque. The default is From hub to ring for positive rotation.

### Initial state

From the drop-down list, choose whether the clutch starts simulation in a locked or unlocked state. The default is Unlocked.

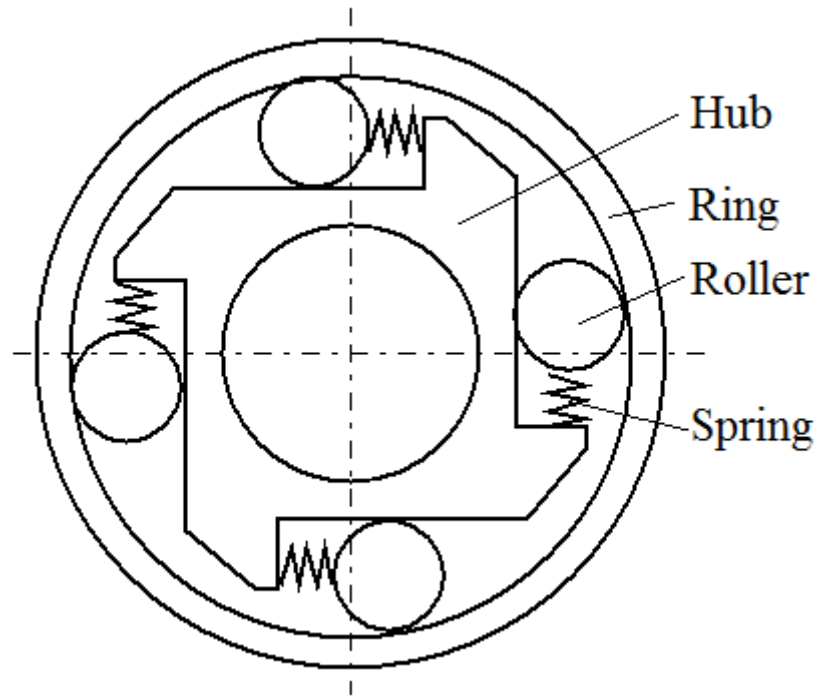
## Unidirectional Clutch Model

The Unidirectional Clutch, if locked, transmits torque from hub to ring. If it is unlocked, it transmits no torque. The clutch does not have a state of being unlocked but still engaged (transmitting torque).

### Clutch Geometry, Variable, and State Summary

The figure displays the geometry of the unidirectional clutch. The table summarizes the unidirectional clutch variables.

# Unidirectional Clutch



## Clutch Geometry

## Clutch Variable

Symbol	Definition	Significance
$\omega$	Relative angular velocity (slip)	$\omega_H - \omega_R$

The Unidirectional Clutch block implements an ideal model of a one-way clutch and does not include any internal dynamics, such as the rollers and springs.

## Locking Condition

The transmission direction determines the condition for locking.

- If you choose transmission for the positive direction, the clutch locks if  $\omega = \omega_H - \omega_R \geq 0$  and becomes zero from positive values. The slip  $\omega$  then remains zero, and the clutch transfers torque.

If the driveline dynamics then causes  $\omega < 0$  at a later time, the clutch abruptly unlocks and stops transferring torque.

- If you choose transmission for the negative direction, the clutch locks if  $\omega = \omega_H - \omega_R \leq 0$  and becomes zero from negative values. The slip  $\omega$  then remains zero, and the clutch transfers torque.

If the driveline dynamics then causes  $\omega > 0$  at a later time, the clutch abruptly unlocks and stops transferring torque.

For example, locking and torque transfer in the positive direction as follows.

- When the hub rotates faster than the ring ( $\omega > 0$ ), the small gap prearranged between the hub and the ring is cleared. The spring-loaded rollers wedge between the hub and the ring, causing the two to contact, lock, and rotate together;  $\omega$  resets to zero. (The rotation convention in the figure is clockwise.)
- If the ring rotates faster than the hub ( $\omega < 0$ ), the rollers are pushed out of contact and the clutch unlocks.

For visual details, consult the figure, Clutch Geometry on page 1-438.

### **Power Dissipated by the Clutch**

The unidirectional clutch is ideal and does not dissipate power. Either the clutch is unlocked and applies no kinetic friction, or it is locked (no slip) and applies only static friction.

### **Example**

The `sdl_stepping_mechanism` example model uses the Unidirectional Clutch to translate a mass in one direction only (ratcheting).

### **See Also**

Dog Clutch, Fundamental Friction Clutch  
“Gear Coupling Control with Clutches”

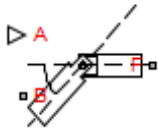
# Universal Joint

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**Purpose** Rotational coupling between two driveline shafts

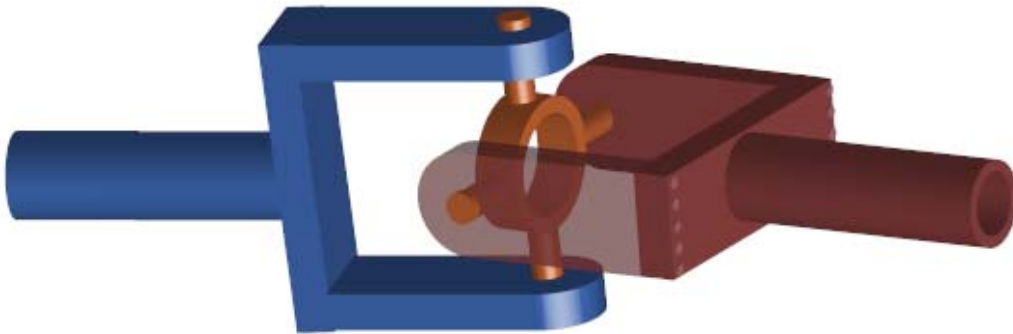
**Library** Couplings & Drives

## Description



This block represents a rotational coupling between two driveline shafts. The coupling transfers torque between the shafts so they spin as a unit under an applied load. Two rotational degrees of freedom, internal to the coupling, allow the shafts to connect at an angle. This intersection angle varies according to the physical signal input from port A. Compliance, modeled as a parallel spring-damper set, allows the coupling to deform under load.

You can use the Universal Joint block as a connection between two rotational driveline components—for example, between the driving and driven shafts in an automobile drive train.



## Universal Joint

The ratio of the shaft angular velocities depends on two parameters: the intersection angle between the two shafts and the rotation angle of the base shaft. A physical signal input provides the intersection angle while a dialog box parameter provides the initial base shaft angle. These two

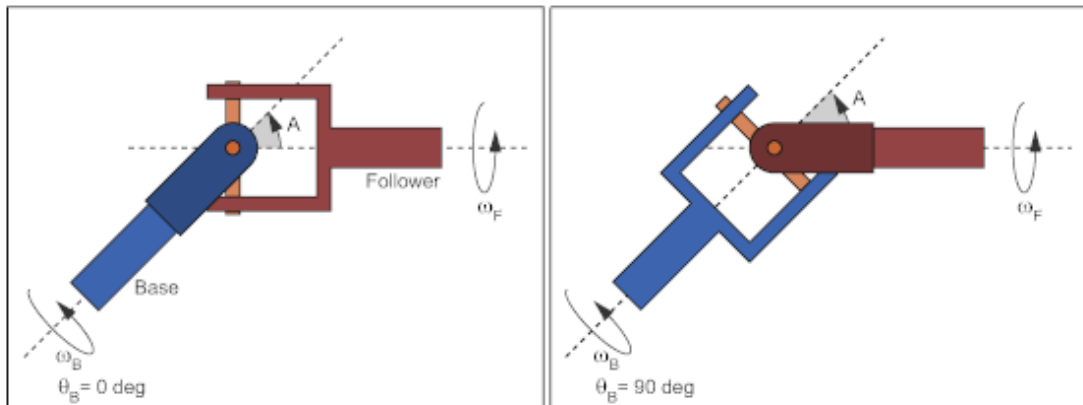
angles fix the ratio of the two shaft angular velocities according to the nonlinear equation:

$$\omega_F = \frac{\cos(A)}{1 - \sin^2(A) \cos^2(\theta_B)} \omega_B$$

where:

- $\omega_F$  is the angular velocity of the follower shaft about its length axis.
- $\omega_B$  is the angular velocity of the base shaft about its length axis.
- $\theta_B$  is the rotation angle of the base shaft about its length axis.
- $A$  is the intersection angle between base and follower shafts about the base shaft pin.

The two schematics in the figure illustrate the equation parameters. In each schematic, the left shaft represents the base shaft, while the right shaft represents the follower shaft. The right schematic shows the coupling seen in the left schematic after the shafts spin  $90^\circ$  about their length axes (dashed line segments).

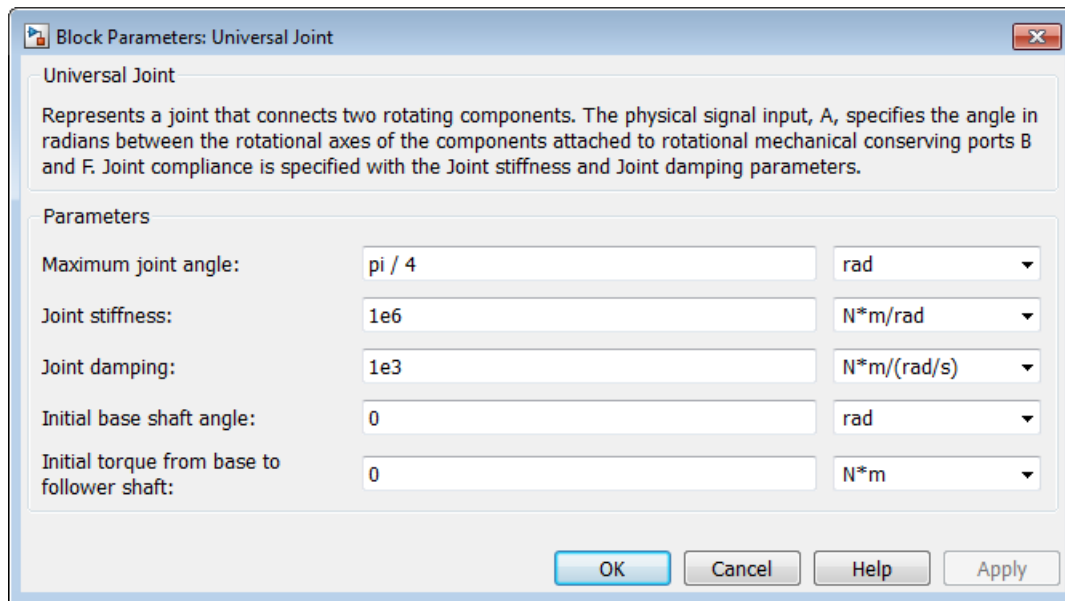


# Universal Joint

In the figure, the intersection angle is the angle between the two shafts about the base shaft's pin. The absolute value of this angle must fall in the range  $0 < A < \text{Maximum intersection angle}$ . The base shaft angle is the angle of the base shaft about its length axis. The base shaft angle is also the time-integral of the base shaft angular velocity,  $\omega_B$ .

## Dialog Box and Parameters

The dialog box contains one active area, **Parameters**, with no tabs.



### Maximum joint angle

Enter the maximum intersection angle the joint allows. This angle measures the rotation between base and follower shafts about the base shaft pin. The value of this angle must fall in the range  $0 < A < \pi/2$  ( $0 < A < 90$  deg). From the drop-down list, select a physical unit. The default value is  $\pi/4$  rad.



## Joint stiffness

Enter the linear spring stiffness of the joint. The spring stiffness accounts for elastic energy storage in the joint due to material compliance. In the drop-down list, select a physical unit. The default value is  $1e6 \text{ N}\cdot\text{m}/\text{rad}$ .

## Joint damping

Enter the linear damping coefficient of the joint. The damping coefficient accounts for energy dissipation in the joint due to material compliance. In the drop-down list, select a physical unit. The default value is  $1e3 \text{ N}\cdot\text{m}/(\text{rad}/\text{s})$ .

## Initial base shaft angle

Enter the rotation angle of the base shaft about its length axis at time zero. From the drop-down list, select a physical unit. The default value is  $0 \text{ rad}$ .

## Initial torque from base to follower shaft

Enter the torque that the base shaft transfers to the follower shaft at time zero. This torque determines the initial state of material compliance at the joint. Set this value to greater than zero to preload the shafts with torque. Changing this value alters the initial transient response due to material compliance. Select a physical unit. The default value is  $0 \text{ N}\cdot\text{m}$ .

## Ports

Port	Description
B	Conserving rotational port associated with the base shaft
F	Conserving rotational port associated with the follower shaft
A	Physical signal input port for the intersection angle

## See Also

Belt Drive | Belt Pulley | Chain Drive | Flexible Shaft | Rope Drum | Torsional Spring-Damper

# Variable Ratio Transmission

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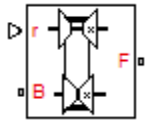
## Purpose

Dynamic gearbox with variable and controllable gear ratio, transmission compliance, and friction losses

## Library

Couplings & Drives

## Description



The Variable Ratio Transmission block represents a gearbox that dynamically transfers motion and torque between the two connected driveshaft axes, base and follower.

Ignoring the dynamics of transmission compliance, the driveshafts are constrained to corotate with a variable gear ratio that you control. You can choose whether the follower axis rotates in the same or opposite direction as the base axis. If they rotate in the same direction,  $\omega_F$  and  $\omega_B$  have the same sign. If they rotate in opposite directions,  $\omega_F$  and  $\omega_B$  have opposite signs.

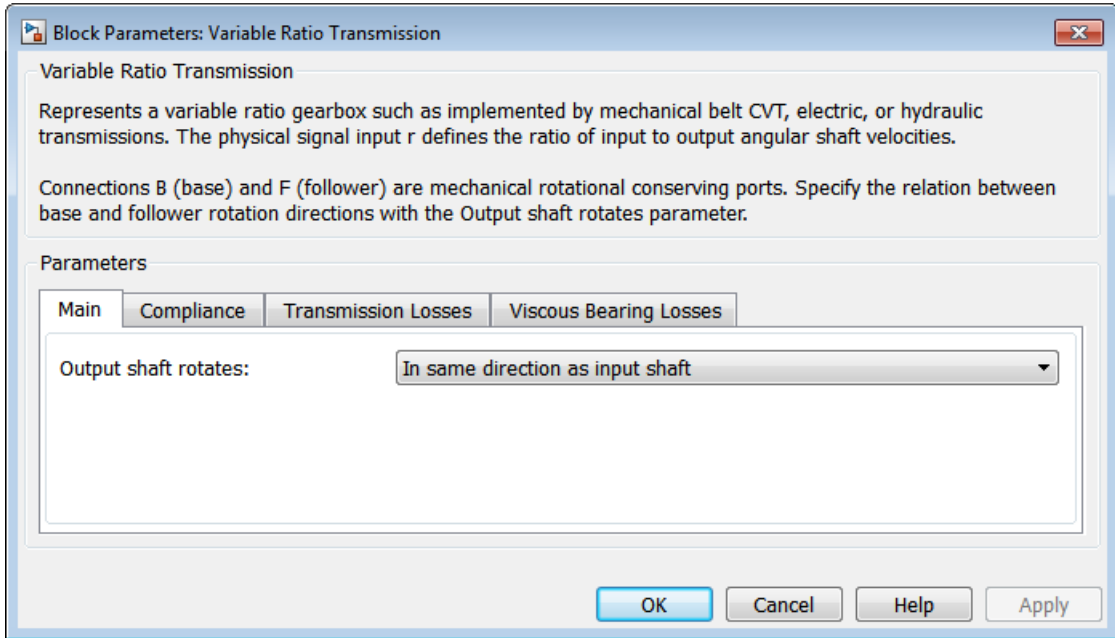
Transmission compliance introduces internal time delay between the axis motions. Unlike a gear, a variable ratio transmission therefore does not act as a kinematic constraint. You can also control torque loss caused by transmission and viscous bearing losses. For model details, see “Variable Ratio Transmission Model” on page 1-448.

## Ports

B and F are rotational conserving ports representing, respectively, the base and follower driveshafts.

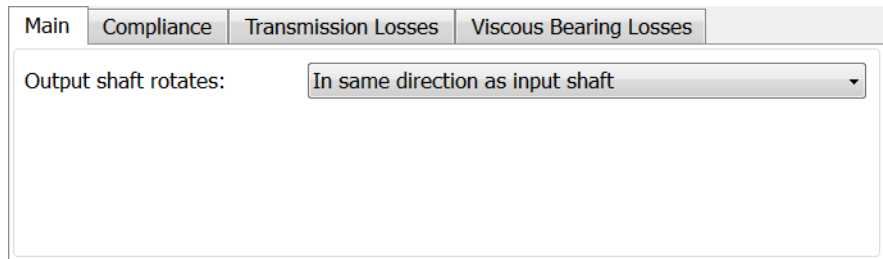
You specify the unitless variable gear ratio  $g_{FB}(t)$  as a function of time at the physical signal input at port r. If the signal value becomes zero or negative, the simulation stops with an error.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with four tabs.

### Main



# Variable Ratio Transmission

## Output shaft rotates

From the drop-down list, choose how the output driveshaft rotates relative to the input driveshaft. The default is In same direction as input shaft.

## Compliance

Main	Compliance	Transmission Losses	Viscous Bearing Losses
Transmission compliance at base (B):	<input type="text" value="3e+5"/>	<input type="text"/>	<input type="text"/>
Transmission damping at base (B):	<input type="text" value="0.05"/>	<input type="text"/>	<input type="text"/>
Initial input torque at base (B):	<input type="text" value="0"/>	<input type="text"/>	<input type="text"/>

## Transmission compliance at base (B)

Reciprocal of transmission angular compliance  $k_p$ , angular displacement per unit torque, measured at the base. The default is 30000.

From the drop-down list, choose units. The default is newton-meters/radian (N\*m/rad).

## Transmission damping at base (B)

Reciprocal of transmission angular compliance damping  $k_v$ , angular speed per unit torque, measured at the base. The default is 0.05.

From the drop-down list, choose units. The default is newton-meters/(radian/second) (N\*m/(rad/s)).

## Initial input torque at base (B)

Torque applied at the base driveshaft at the start of simulation ( $t = 0$ ). The default is 0.

From the drop-down list, choose units. The default is newton-meters (N\*m).

## Transmission Losses

Main	Compliance	Transmission Losses	Viscous Bearing Losses
Losses model:		No losses - Suitable for HIL simulation	

### Losses model

Select how to implement friction losses from nonideal torque transfer. The default is No losses.

- No losses Suitable for HIL simulation — Torque transfer is ideal.
- Constant efficiency — Transfer of torque across gearbox is reduced by a constant efficiency  $\eta$  satisfying  $0 < \eta \leq 1$ . If you select this option, the panel changes from its default.

### Constant Efficiency

Main	Compliance	Transmission Losses	Viscous Bearing Losses
Losses model:		Constant efficiency	
Efficiency:	0.8		
Follower angular velocity threshold:	0.01	rad/s	

### Efficiency

Effective torque transfer efficiency  $\eta$  between the base and follower driveshafts.

# Variable Ratio Transmission

## Follower angular velocity threshold

Absolute angular velocity threshold  $\omega_{th}$  above which full efficiency loss is applied, for follower velocity  $\omega_F$ .

From the drop-down list, choose units. The default is radians/second (rad/s).

## Viscous Bearing Losses

Main	Compliance	Transmission Losses	Viscous Bearing Losses
Viscous friction coefficients at base (B) and follower (F): <input type="text" value="[ 0 0 ]"/> <span>N*m/(rad/s)</span>			

## Viscous friction coefficients at base (B) and follower (F)

Vector of viscous damping coefficients  $[\mu_B \mu_F]$  applied at the base and follower driveshafts, respectively. The default is [ 0 0 ].

From the drop-down list, choose units. The default is newton-meters/(radian/second) (N\*m/ (rad/s)).

## Variable Ratio Transmission Model

### Ideal Motion and Torque Transfer

Variable Ratio Transmission is a dynamical mechanism for transferring motion and torque between base and follower.

If the relative compliance  $\varphi$  between the axes is absent, the block is equivalent to a gear with a variable ratio  $g_{FB}(t)$ . Such a gear imposes a time-dependent kinematic constraint on the motions of the two driveshafts:

$$\omega_B = \pm g_{FB}(t) \omega_F, \tau_F = \pm g_{FB}(t) \tau_B.$$

However, Variable Ratio Transmission does include compliance between the axes. Dynamical motion and torque transfer replace the kinematic constraint, with a nonzero  $\varphi$  that dynamically responds through base compliance parameters  $k_p$  and  $k_v$ :

$$d\varphi/dt = \pm g_{FB}(t) \omega_F - \omega_B ,$$

$$\tau_B = -k_p \varphi - k_v d\varphi/dt ,$$

$$\pm g_{FB}(t) \tau_B + \tau_F - \tau_{\text{loss}} = 0 .$$

$\tau_{\text{loss}} = 0$  in the ideal case.

### Estimating Compliance Parameters

- You can estimate the base angular compliance  $k_p$  from the transmission time constant  $t_c$  and inertia  $J$ :  $k_p = J(2\pi/t_c)^2$ .
- You can estimate the base angular velocity compliance  $k_v$  from the transmission time constant  $t_c$ , inertia  $J$ , and damping coefficient  $C$ :  $k_v = (2Ct_c)/2\pi = 2C\sqrt{J/k_p}$ .

### Nonideal Torque Transfer and Losses

With nonideal torque transfer,  $\tau_{\text{loss}} \neq 0$ . Losses in the Variable Ratio Transmission are modeled similarly to how losses are modeled in nonideal gears. For general information on nonideal gear modeling, see “Model Gears with Losses”.

In a nonideal gearbox, the angular velocity and compliance dynamics are unchanged. The transferred torque and power are reduced by:

- Coulomb friction (for example, between belt and wheel, or internal belt losses due to stretching) characterized by an efficiency  $\eta$ .
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients  $\mu$ .

$$\tau_{\text{loss}} = \tau_{\text{Coul}} \tanh(4\omega_{\text{out}}/\omega_{\text{th}}) + \mu_B \omega_B + \mu_F \omega_F , \tau_{\text{Coul}} = |\tau_F| (1 - \eta) .$$

# Variable Ratio Transmission

---

When the angular velocity changes sign, the hyperbolic tangent regularizes the sign change in the Coulomb friction torque.

Power Flow	Power Loss Condition	Output Driveshaft <small>out</small>
Forward	$\omega_B \tau_B > \omega_F \tau_F$	Follower, $\omega_F$
Reverse	$\omega_B \tau_B < \omega_F \tau_F$	Base, $\omega_B$

## Meshing Efficiency

The friction loss represented by efficiency  $\eta$  is fully applied only if the absolute value of the follower angular velocity  $\omega_F$  is greater than a velocity threshold  $\omega_{th}$ .

If this absolute velocity is less than  $\omega_{th}$ , the actual efficiency is automatically regularized to one at zero velocity.

## Limitations

Coulomb friction simulation increases model fidelity but reduces simulation performance. See “Adjust Model Fidelity”.

## Example

The `sdl_variable_gear` example model provides a basic example of a simple gear.

## See Also

Simple Gear, Torque Converter  
“Gear Coupling Control with Clutches”



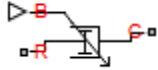
## Purpose

Rotational damper with variable damping coefficient

## Library

Couplings & Drives/Springs & Dampers

## Description



The block represents a rotational viscous damper with a variable damping coefficient. A physical signal input port provides the magnitude of the damping coefficient. The magnitude of the damping torque is equal to the product of the physical signal input and the relative angular velocity between the two rotational conserving ports, R and C. A minimum damping coefficient prevents non-physical negative damping values.

The damping torque satisfies the following expression:

$$T = \begin{cases} B \omega & B \geq B_{\min} \\ B_{\min} \omega & B < B_{\min} \end{cases}$$

The parameters are:

- $T$  — Torque transmitted through damper
- $B$  — Viscous damping coefficient
- $\omega$  — Relative angular velocity measured between the two rotational conserving ports according to  $\omega = \omega_R - \omega_C$
- $\omega_R$  — Angular velocity of rotational conserving port R
- $\omega_C$  — Angular velocity of rotational conserving port C

The block applies equal and opposite damping torques on the rotational conserving ports. The sign of the damping torque acting on port R is equal to the sign of the relative angular velocity. A positive relative angular velocity corresponds to a positive damping torque acting on port R, and a negative damping torque of equal magnitude acting on port C.

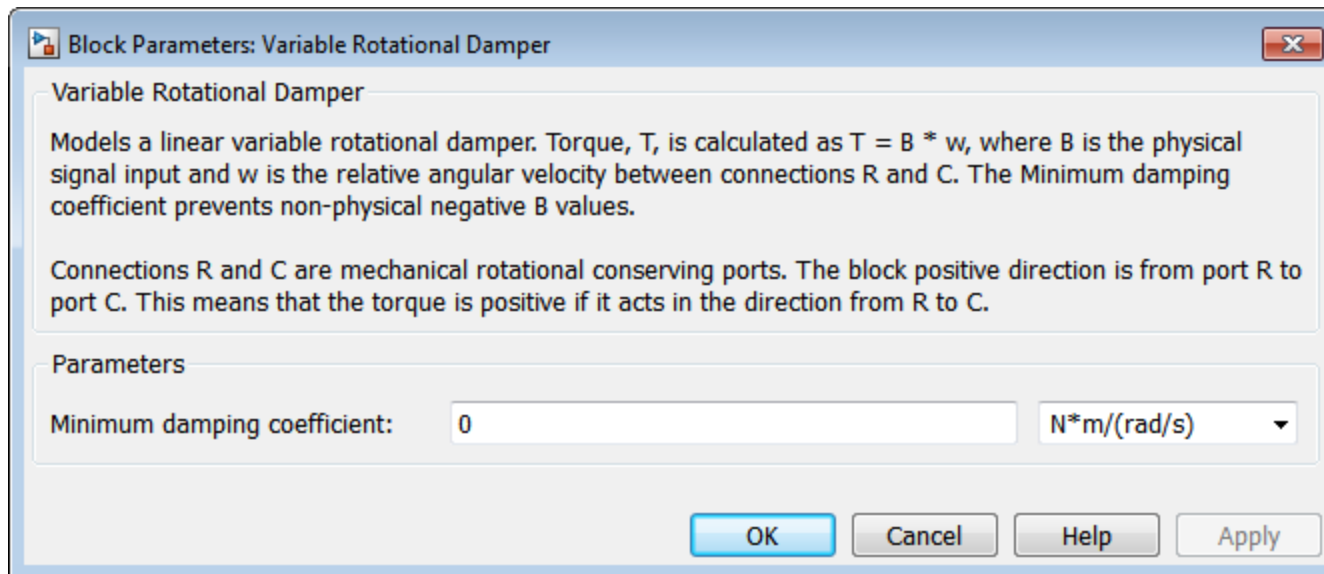
# Variable Rotational Damper

## Assumptions and Limitations

- The block represents only viscous damping
- The value of the damping coefficient must be equal to or greater than zero.

## Dialog Box and Parameters

The dialog box includes one tab, **Parameters**.



### Minimum damping coefficient

Minimum value allowed for the damping coefficient. The physical signal input saturates below this level. The parameter must be greater than or equal to zero. The default value is 0. The default unit is  $N*m/(rad/s)$ .

# Variable Rotational Damper

---

## Ports

Port	Description
B	Physical signal input port representing the variable damping coefficient
C	Rotational conserving port
R	Rotational conserving port

## See Also

Nonlinear Rotational Damper, Variable Translational Damper, Nonlinear Translational Damper

# Variable Rotational Spring

**Purpose** Rotational spring with variable spring stiffness

**Library** Couplings & Drives/Springs & Dampers

**Description** The block represents a torsional spring with variable spring stiffness. A physical signal input port provides the value of the variable spring stiffness. The magnitude of the spring torque is equal to the product of the physical signal input and the relative angular displacement between the two rotational conserving ports. A minimum spring rate value prevents non-physical values.



The rotational spring torque satisfies the following expression:

$$T = \begin{cases} K \theta & K \geq K_{\min} \\ K_{\min} \theta & K < K_{\min} \end{cases}$$

The parameters are:

- $T$  — Torque transmitted through the spring
- $K$  — Spring rate (stiffness)
- $K_{\min}$  — Minimum allowable spring rate (stiffness)
- $\theta$  — Relative angular displacement measured between the two rotational conserving ports according to  $\theta = \theta_{\text{init}} + \theta_R - \theta_C$
- $\theta_{\text{init}}$  — Initial angular deformation between the two rotational conserving ports
- $\theta_R$  — Absolute angular displacement of rotational conserving port R
- $\theta_C$  — Absolute angular displacement of rotational conserving port C

The block applies equal and opposite spring torques on the two rotational conserving ports. The sign of the spring torque acting on port R is equal to the sign of the relative angular velocity. A positive relative angular velocity corresponds to a positive damping torque acting on port R, and a negative damping torque of equal magnitude acting on port C.

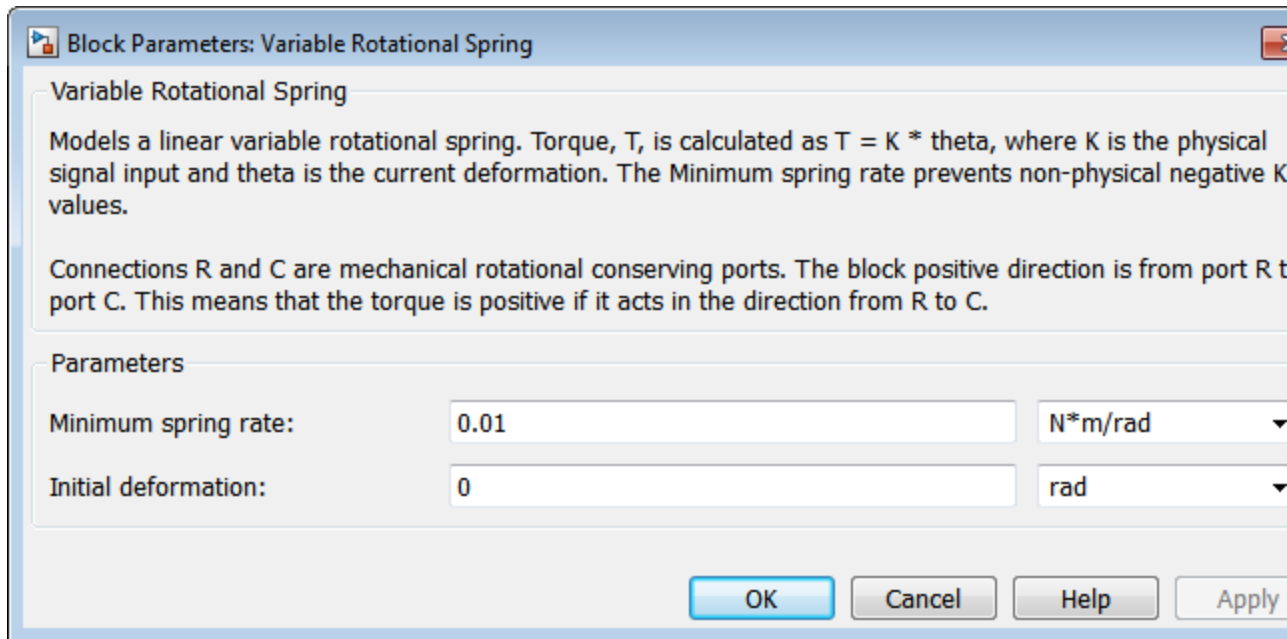
# Variable Rotational Spring

## Assumptions and Limitations

- The value of the spring rate must be greater than zero.

## Dialog Box and Parameters

The dialog box includes one tab, **Parameters**.



### Minimum spring rate

Minimum value allowed for the spring rate (stiffness). The physical signal input saturates below the specified value. The parameter must be greater than zero. The default value is 0.01. The default unit is (N\*m/rad).

# Variable Rotational Spring

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## Initial Deformation

Initial angular displacement between rotational conserving ports R and C. The initial value is 0. The initial unit is rad.

## Ports

Port	Description
K	Physical signal input port representing the variable spring stiffness
C	Rotational conserving port
R	Rotational conserving port

## See Also

Nonlinear Rotational Spring, Variable Translational Spring, Nonlinear Translational Spring

# Variable Translational Damper

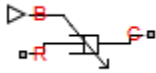
## Purpose

Translational viscous damper with variable damping coefficient

## Library

Couplings & Drives/Springs & Dampers

## Description



The block represents a translational viscous damper with variable damping coefficient. A physical signal input port provides the magnitude of the translational damping coefficient. The magnitude of the damping force is equal to the product of the physical signal input and the relative linear velocity between the two translational conserving ports. A minimum damping coefficient prevents non-physical negative damping values.

The translational damping force satisfies the following expression:

$$F = \begin{cases} B \cdot v, & B \geq B_{\min} \\ B_{\min} \cdot v, & B < B_{\min} \end{cases}$$

The parameters are:

- $F$  — Force transmitted through the translational damper between the two translational conserving ports
- $B$  — Viscous damping coefficient
- $B_{\min}$  — Minimum allowed damping coefficient
- $v$  — Relative linear velocity measured between the two translational conserving ports
- $v_R$  — Linear velocity of port R
- $v_C$  — Linear velocity of port C

The block applies equal and opposite damping forces on the two translational conserving ports. The sign of the damping force acting on port R is equal to the sign of the relative linear velocity. A positive linear velocity corresponds to a positive damping force acting on port R, and a negative damping force of equal magnitude acting on port C.

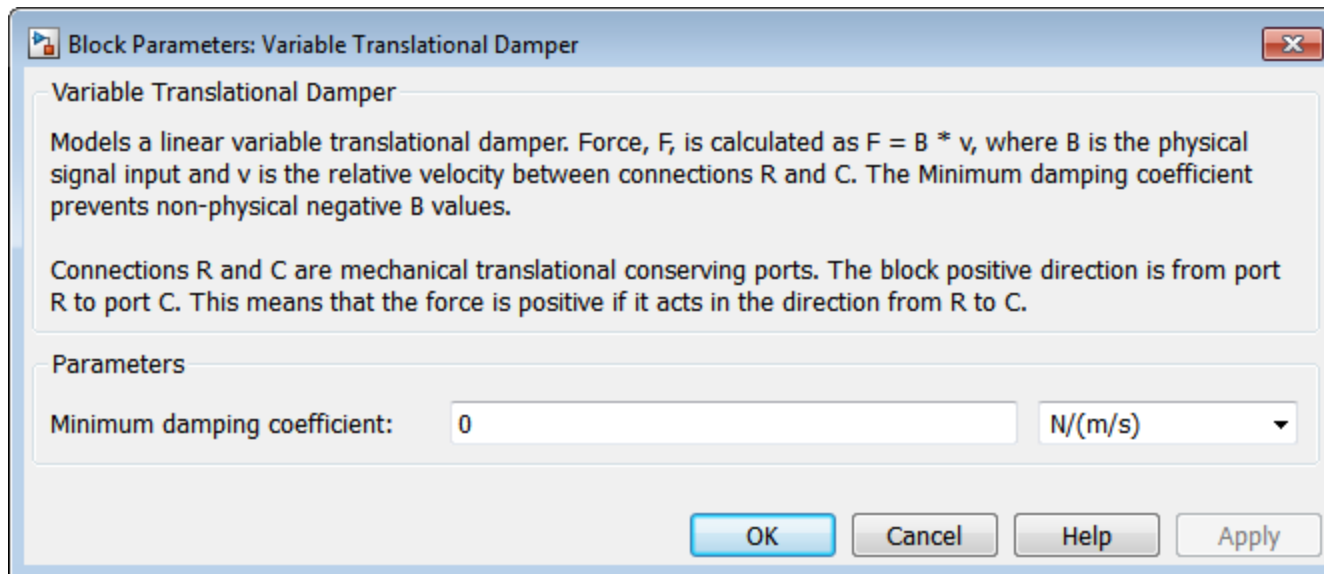
# Variable Translational Damper

## Assumptions and Limitations

- The block represents strictly viscous damping.
- The value of the damping coefficient must be greater than or equal to zero.

## Dialog Box and Parameters

The dialog box includes one tab, **Parameters**.



### Minimum damping coefficient

Minimum value allowed for the variable damping coefficient. The physical signal input saturates below the specified value. The parameter must be greater than or equal to zero. The default value is 0. The default unit is N/(m/s).



# Variable Translational Damper

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## Ports

Port	Description
B	Physical signal input port representing the variable translational damping coefficient
C	Translational conserving port
R	Translational conserving port

## See Also

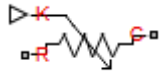
Nonlinear Translational Damper, Variable Rotational Damper, Nonlinear Rotational Damper

# Variable Translational Spring

**Purpose** Translational spring with variable spring stiffness

**Library** Couplings & Drives/Springs & Dampers

## Description



The block represents a translational spring with variable spring stiffness. A physical signal input port provides the variable spring stiffness. The magnitude of the spring force is equal to the product of the physical signal input and the relative linear displacement between the two translational conserving ports. A minimum spring rate prevents non-physical values.

The translational spring force satisfies the following expression:

$$F = \begin{cases} K x, & K \geq K_{\min} \\ K_{\min} x, & K < K_{\min} \end{cases}$$

The parameters are:

- $F$  — Force transmitted through the spring between the two translational conserving ports
- $K$  — Spring rate (stiffness)
- $K_{\min}$  — Minimum allowed spring rate (stiffness)
- $x$  — Relative displacement between the two translational conserving ports according to  $x = x_{\text{init}} + x_R - x_C$
- $x_{\text{init}}$  — initial spring deformation
- $x_R$  — Absolute displacement of translational conserving port R
- $x_C$  — Absolute displacement of translational conserving port C

The block applies equal and opposite spring forces on the two translational conserving ports. The sign of the spring force acting on port R is equal to the sign of the relative linear displacement between the two ports. A positive linear velocity corresponds to a positive spring force acting on port R, and a negative spring force of equal magnitude acting on port C.

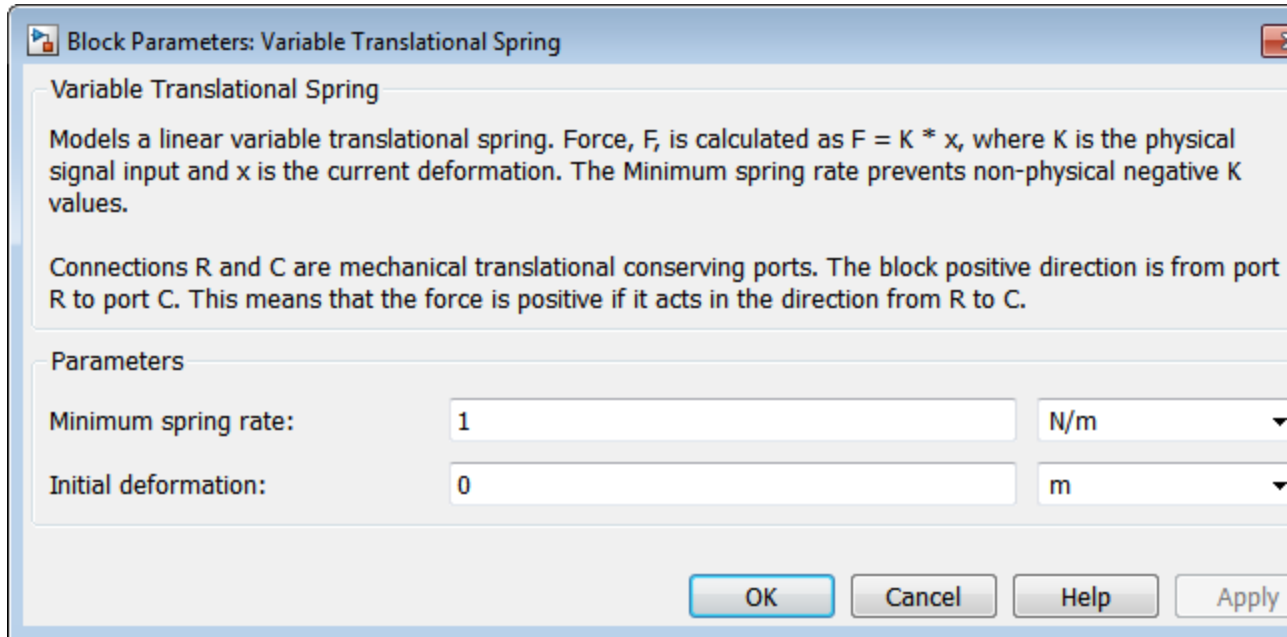
# Variable Translational Spring

## Assumptions and Limitations

- The value of the variable spring stiffness must be greater than zero.

## Dialog Box and Parameters

The dialog box includes one tab, **Parameters**.



### Minimum spring rate

Minimum value allowed for the spring rate. The physical signal input saturates below the specified value. The parameter must be greater than zero. The default value is 1. The default unit is N/m

### Initial deformation

Initial displacement between the two translational conserving ports. The default value is 0. The default unit is m.

# Variable Translational Spring

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## Ports

Port	Description
K	Physical signal input port representing the variable spring stiffness
C	Translational conserving port
R	Translational Conserving port

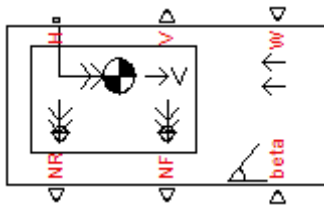
## See Also

Nonlinear Translational Spring, Variable Rotational Spring, Nonlinear Rotational Spring

**Purpose** Two-axle vehicle with longitudinal dynamics and motion and adjustable mass, geometry, and drag properties

**Library** Tires & Vehicles

## Description



The Vehicle Body block models a two-axle vehicle, with an equal number of equally sized wheels on each axle, moving forward or backward along its longitudinal axis.

The model includes the following vehicle properties.

- Mass
- Number of wheels on each axle
- Position of the vehicle's center of gravity (CG) relative to the front and rear axles and to the ground
- Effective frontal cross-sectional area
- Aerodynamic drag coefficient
- Initial longitudinal velocity

For model details, see “Vehicle Body Model” on page 1-467.

## Ports

You specify the headwind speed  $V_w$  (in meters/second) and the road inclination angle  $\beta$  (in radians) through physical signal inputs at ports W and beta, respectively.

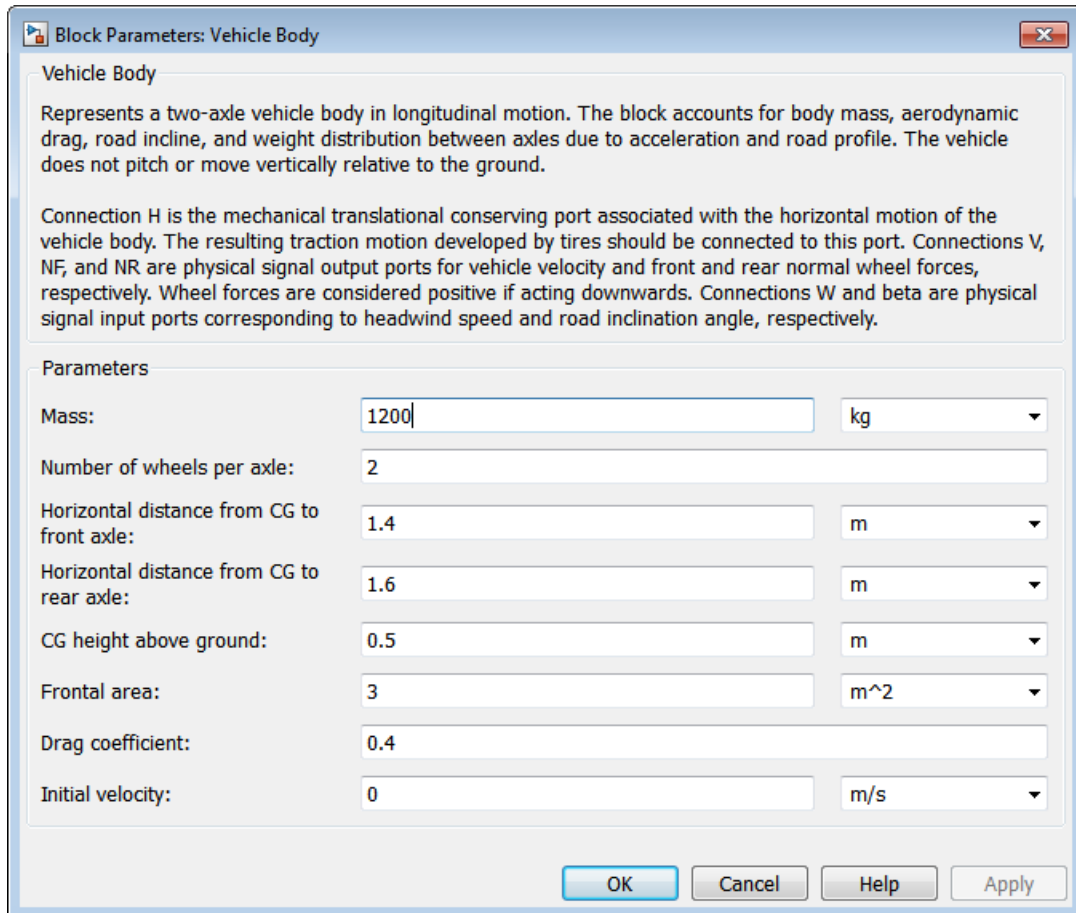
# Vehicle Body

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The block reports the longitudinal vehicle velocity  $V_x$  and the front and rear normal forces (load on wheels)  $F_{zf}$ ,  $F_{zr}$  as physical signal outputs at ports V, NF, and NR, respectively.

The horizontal motion of the vehicle is represented by the translational conserving port H.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**.

# Vehicle Body

---

## **Mass**

Mass  $m$  of the vehicle. The default is 1200.

From the drop-down list, choose units. The default is kilograms (kg).

## **Number of wheels per axle**

Number  $n$  of equally-sized wheels on each axle, forward and rear. The default is 2.

## **Horizontal distance from CG to front axle**

Horizontal distance  $a$  from the vehicle's center of gravity to the vehicle's front wheel axle. The default is 1.4.

From the drop-down list, choose units. The default is meters (m).

## **Horizontal distance from CG to rear axle**

Horizontal distance  $b$  from the vehicle's center of gravity to the vehicle's rear wheel axle. The default is 1.6.

From the drop-down list, choose units. The default is meters (m).

## **CG height above ground**

Height  $h$  of the vehicle's center of gravity from the ground. The default is 0.5.

From the drop-down list, choose units. The default is meters (m).

## **Frontal area**

Effective cross-sectional area  $A$  presented by the vehicle in longitudinal motion, to compute the aerodynamic drag force on the vehicle. The default is 3.

From the drop-down list, choose units. The default is meters-squared ( $\text{m}^2$ ).

## **Drag coefficient**

The dimensionless aerodynamic drag coefficient  $C_d$ , for the purpose of computing the aerodynamic drag force on the vehicle. The default is 0.4.



## Vehicle Body Model

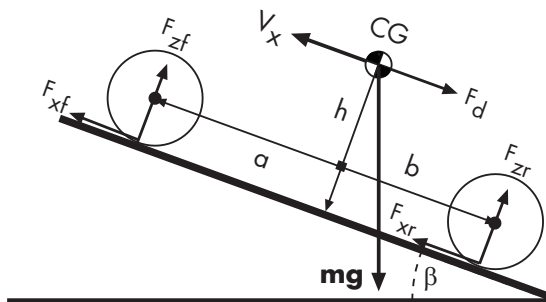
### Initial velocity

The initial value  $V_x(0)$  of the vehicle's horizontal velocity. The default is 0.

From the drop-down list, choose units. The default is meters/second (m/s).

The vehicle axles are parallel and form a plane. The longitudinal  $x$  direction lies in this plane and perpendicular to the axles. If the vehicle is traveling on an incline slope  $\beta$ , the normal  $z$  direction is not parallel to gravity but is always perpendicular to the axle-longitudinal plane.

This figure and table define the vehicle motion model variables.



### Vehicle Dynamics and Motion

### Vehicle Model Variables

Symbol	Description and Unit
$g$	Gravitational acceleration = 9.81 m/s <sup>2</sup>
$\beta$	Incline angle
$m$	Vehicle mass
$h$	Height of vehicle CG above the ground

# Vehicle Body

---

## Vehicle Model Variables (Continued)

Symbol	Description and Unit
$a, b$	Distance of front and rear axles, respectively, from the normal projection point of vehicle CG onto the common axle plane
$V_x$	Longitudinal vehicle velocity
$V_W$	Headwind speed
$n$	Number of wheels on each axle
$F_{xf}, F_{xr}$	Longitudinal forces on each wheel at the front and rear ground contact points, respectively
$F_{zf}, F_{zr}$	Normal load forces on the each wheel at the front and rear ground contact points, respectively
$A$	Effective frontal vehicle cross-sectional area
$C_d$	Aerodynamic drag coefficient
$\rho$	Mass density of air = 1.2 kg/m <sup>3</sup>
$F_d$	Aerodynamic drag force

## Vehicle Dynamics and Motion

The vehicle motion is determined by the net effect of all the forces and torques acting on it. The longitudinal tire forces push the vehicle forward or backward. The weight  $mg$  of the vehicle acts through its center of gravity (CG). Depending on the incline angle, the weight pulls the vehicle to the ground and pulls it either backward or forward. Whether the vehicle travels forward or backward, aerodynamic drag slows it down. For simplicity, the drag is assumed to act through the CG.

$$\begin{aligned}m\dot{V}_x &= F_x - F_d - mg \cdot \sin\beta, \\F_x &= n(F_{xf} + F_{xr}), \\F_d &= \frac{1}{2}C_d\rho A(V_x - V_W)^2 \cdot \text{sgn}(V_x - V_W)\end{aligned}$$

Zero normal acceleration and zero pitch torque determine the normal force on each front and rear wheel:

$$\begin{aligned}F_{zf} &= \frac{-h(F_d + mg \sin\beta + m\dot{V}_x) + b \cdot mg \cos\beta}{n(a+b)}, \\F_{zr} &= \frac{+h(F_d + mg \sin\beta + m\dot{V}_x) + a \cdot mg \cos\beta}{n(a+b)}\end{aligned}$$

The wheel normal forces satisfy  $F_{zf} + F_{zr} = mg \cdot \cos\beta/n$ .

## Limitations

The Vehicle Body block lets you model only longitudinal dynamics, parallel to the ground and oriented along the direction of motion. The vehicle is assumed to be in pitch and normal equilibrium. The block does not model pitch or vertical movement.

## Examples

These SimDriveline example models contain working examples of vehicle bodies:

- `sdl_dual_clutch`
- `sdl_manual_transmission`
- `sdl_4wd_dynamics`
- `sdl_vehicle`

## See Also

Tire (Magic Formula)  
“Complete Vehicle Model”

# Worm Gear

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## Purpose

Worm gear with adjustable gear ratio and friction losses

## Library

Gears

## Description

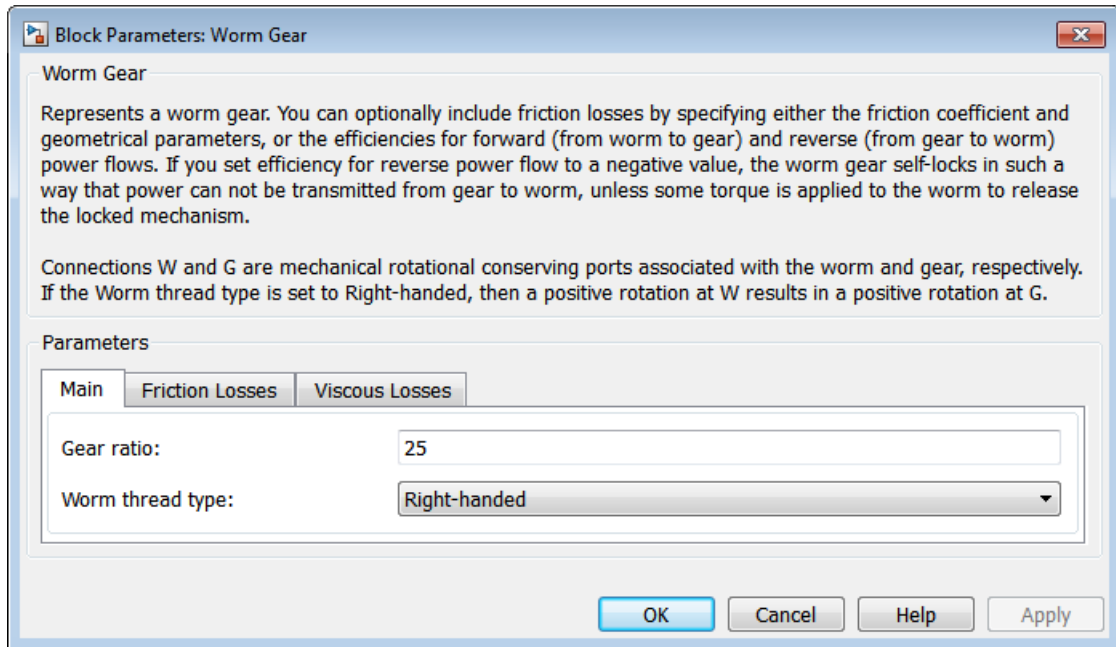


The block represents a rotational gear that constrains the two connected driveline axes, worm (W) and gear (G), to rotate together in a fixed ratio that you specify. You can choose whether the gear rotates in a positive or negative direction. Right-handed rotation is the positive direction. If the worm thread is right-handed,  $\omega_W$  and  $\omega_G$  have the same sign. If the worm thread is left-handed,  $\omega_W$  and  $\omega_G$  have opposite signs.

## Ports

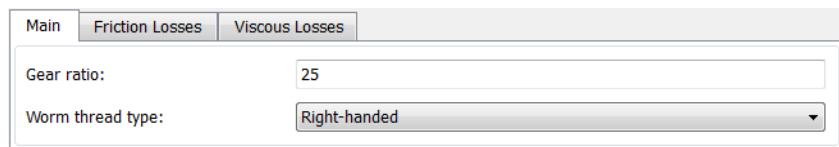
W and G are rotational conserving ports. The ports represent the worm and the gear, respectively.

## Dialog Box and Parameters



The dialog box has one active area, **Parameters**, with three tabs.

### Main



# Worm Gear

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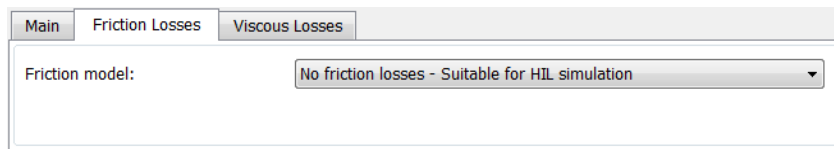
## Gear ratio

Gear or transmission ratio  $R_{WG}$  determined as the ratio of the worm angular velocity to the gear angular velocity. The default is 25.

## Worm thread type

Choose the directional sense of gear rotation corresponding to positive worm rotation. The default is **Right-handed**. If you select **Left-handed**, rotation of the worm in the generally-assigned positive direction results in the gear rotation in negative direction.

## Friction Losses



Friction model: No friction losses - Suitable for HIL simulation

## Friction model

Select how to implement friction losses from nonideal meshing of gear threads. The default is **No friction losses**.

- **No friction losses** Suitable for HIL simulation — Gear meshing is ideal.
- **Constant efficiency** — Transfer of torque between worm and gear is reduced by friction. If you select this option, the panel expands.

## Constant Efficiency

Main	Friction Losses	Viscous Losses
Friction model:	Constant efficiency	
Friction parameterization:	Friction coefficient and geometrical parameters	
Normal pressure angle:	17.5	deg
Lead angle:	20	deg
Friction coefficient:	0.08	
Angular velocity threshold:	0.01	rad/s

### Friction parametrization

Select how to characterize the friction between gear threads. The default is Friction coefficient and geometrical parameters.

- Friction coefficient and geometrical parameters — Friction is determined by contact friction between surfaces.

#### Normal pressure angle

The thread pressure angle  $\alpha$  in the normal plane. Must be greater than zero and less than 90 degrees. The default is 17.5.

From the drop-down list, choose units. The default is degrees (deg).

#### Lead angle

Thread helix angle  $\lambda = \arctan[L/(\pi d)]$ .  $L$  is the worm lead,  $d$  is the worm pitch diameter. Must be greater than zero. The default is 20.

From the drop-down list, choose units. The default is degrees (deg).

## Friction coefficient

Dimensionless coefficient of normal friction in the thread. Must be greater than zero. The default is 0.08.

- **Efficiencies** — Friction is determined by constant efficiencies  $0 < \eta < 1$ . If you select this option, the panel changes from its default.

## Efficiencies

Parameter	Value	Unit
Friction model	Constant efficiency	
Friction parameterization	Efficiencies	
Worm-gear efficiency	0.74	
Gear-worm efficiency	0.65	
Angular velocity threshold	0.01	rad/s

## Worm-gear efficiency

Efficiency  $\eta_{WG}$  of the power transfer from worm to gear.

## Gear-worm efficiency

Efficiency  $\eta_{GW}$  of the power transfer from gear to worm.

## Angular velocity threshold

Absolute angular velocity threshold above which full efficiency loss is applied. Must be greater than zero. The default is 0.01.

From the drop-down list, choose units. The default is radians/second (rad/s).



## Viscous Losses

Main
Friction Losses
Viscous Losses

Viscous friction coefficients at worm (W) and gear (G):  N\*m/(rad/s) ▾

### Viscous friction coefficients at worm (W) and gear (G)

Vector of viscous friction coefficients  $[\mu_W \mu_G]$ , for the worm and gear, respectively. The default is [0 0].

From the drop-down list, choose units. The default is newton-meters/(radians/second) (N\*m/(rad/s)).

## Worm Gear Model

### Model Variables

$R_{WG}$	Gear ratio
$\omega_W$	Worm angular velocity
$\omega_G$	Gear angular velocity
$a$	Normal pressure angle
$\lambda$	Worm lead angle
$L$	Worm lead
$d$	Worm pitch diameter
$\tau_G$	Gear torque
$\tau_W$	Torque on the worm
$\tau_{loss}$	Torque loss due to meshing friction. The loss depends on the device efficiency and the power flow direction. To avoid abrupt change of the friction torque at $\omega_G = 0$ , the friction torque is introduced via the hyperbolic function.
$\tau_{fr}$	Steady-state value of the friction torque at $\omega_G \rightarrow \infty$ .

# Worm Gear

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$k$	Friction coefficient
$\eta_{WG}$	Torque transfer efficiency from worm to gear
$\eta_{GW}$	Torque transfer efficiency from gear to worm
$\omega_{th}$	Absolute angular velocity threshold
$[\mu_W$ $\mu_G]$	Vector of viscous friction coefficients for the worm and gear

## Ideal Gear Constraint and Gear Ratio

Worm gear imposes one kinematic constraint on the two connected axes:

$$\omega_W = R_{WG}\omega_G .$$

The two degrees of freedom are reduced to one independent degree of freedom. The forward-transfer gear pair convention is (1,2) = (W,G).

The torque transfer is:

$$R_{WG}\tau_W - \tau_G - \tau_{loss} = 0 ,$$

with  $\tau_{loss} = 0$  in the ideal case.

## Nonideal Gear Constraint

In a nonideal worm-gear pair (W,G), the angular velocity and geometric constraints are unchanged. But the transferred torque and power are reduced by:

- Coulomb friction between thread surfaces on W and G, characterized by friction coefficient  $k$  or constant efficiencies  $[\eta_{WG} \eta_{GW}]$
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients  $\mu$

The loss torque has the general form:

$$\tau_{loss} = \tau_{fr} \tanh(4\omega_G/\omega_{th}) + \mu_G\omega_G + \mu_W\omega_W .$$

The hyperbolic tangent regularizes the sign change in the friction torque when the gear velocity changes sign.

<b>Power Flow Condition</b>	<b>Power Loss</b>	<b>Output Driveshaft</b>	<b>Friction Torque <math>\tau_{fr}</math></b>
Forward, $\omega_W \tau_W > \omega_G \tau_G$		Gear, $\omega_G$	$R_{WG}  \tau_W  (1 - \eta_{WG})$
Reverse, $\omega_W \tau_W \leq \omega_G \tau_G$		Worm, $\omega_W$	$ \tau_G  (1 - \eta_{GW})$

### Geometric Surface Contact Friction

In the contact friction case,  $\eta_{WG}$  and  $\eta_{GW}$  are determined by:

- The worm-gear threading geometry, specified by lead angle  $\lambda$  and normal pressure angle  $a$ .
- The surface contact friction coefficient  $k$ .

$$\eta_{WG} = (\cos a - k \tan \lambda) / (\cos a + k / \tan \lambda) ,$$

$$\eta_{GW} = (\cos a - k / \tan \lambda) / (\cos a + k \tan \lambda) .$$

### Constant Efficiencies

In the constant friction case, you specify  $\eta_{WG}$  and  $\eta_{GW}$ , independently of geometric details.

### Self-Locking and Negative Efficiency

$\eta_{GW}$  has two distinct regimes, depending on lead angle  $\lambda$ , separated by the *self-locking point* at which  $\eta_{GW} = 0$  and  $\cos a = k / \tan \lambda$ .

- In the *overhauling regime*,  $\eta_{GW} > 0$ , and the force acting on the nut can rotate the screw.
- In the *self-locking regime*,  $\eta_{GW} < 0$ , and an external torque must be applied to the screw to release an otherwise locked mechanism. The more negative is  $\eta_{GW}$ , the larger the torque must be to release the mechanism.

$\eta_{WG}$  is conventionally positive.

# Worm Gear

---

## Meshing Efficiency

The efficiencies  $\eta$  of meshing between worm and gear are fully active only if the absolute value of the gear angular velocity is greater than the velocity tolerance.

If the velocity is less than the tolerance, the actual efficiency is automatically regularized to unity at zero velocity.

## Viscous Friction Force

The viscous friction coefficient  $\mu_W$  controls the viscous friction torque experienced by the worm from lubricated, nonideal gear threads and viscous bearing losses. The viscous friction torque on a worm driveline axis is  $-\mu_W\omega_W$ .  $\omega_W$  is the angular velocity of the worm with respect to its mounting.

The viscous friction coefficient  $\mu_G$  controls the viscous friction torque experienced by the gear, mainly from viscous bearing losses. The viscous friction torque on a gear driveline axis is  $-\mu_G\omega_G$ .  $\omega_G$  is the angular velocity of the gear with respect to its mounting.

## Limitations

Gear ratios must be positive. Gear inertia and compliance are ignored. Coulomb friction reduces simulation performance. See “Adjust Model Fidelity”.

## See Also

Leadscrew, Sun-Planet Worm Gear

“Represent and Transfer Driveline Motion and Torque”

# Function Reference

---

# sdl\_update

---

## Purpose

Convert SimDriveline first-generation model or library to second-generation format

## Syntax

```
newsysname = sdl_update('sys')  
newsysname = sdl_update('sys','oldlib1','newlib1','oldlib2',  
    ...)
```

## Description

`newsysname = sdl_update('sys')` converts SimDriveline First Generation components in the Simulink® system `sys`.

`newsysname = sdl_update('sys','oldlib1','newlib1','oldlib2', ...)` converts the SimDriveline First Generation components in the Simulink system `sys` that contains references to the SimDriveline First Generation libraries `oldlib1`, `oldlib2`, ... which are already converted to `newlib1`, `newlib2`, ..., respectively.

In both forms, `sdl_update` creates a new system based on the SimDriveline transitional library (`sdl_classic`) and returns the name of the new system. The new system is saved in your current directory and remains loaded.

## Output Arguments

### **newsysname**

String variable containing name of new, converted Simulink system

## Examples

### **Converting a Model**

Update the model `drive_ex1`:

```
sdl_ex1 = sdl_update('drive_ex1');
```

### **Converting a Model and a Linked Library**

If a model `drive_model` references blocks from library `drive_userlib`, update `drive_userlib` into a new library and `drive_md1` into a new model:

```
sdl_userlib = sdl_update('drive_userlib'); % Update library first  
sdl_model = sdl_update('drive_model','drive_userlib',sdl_userlib);  
% Then update model linked to library
```

`sdl_userlib` and `sdl_model` are string variables containing system names, not the system names themselves.

## **Tutorials**

- “First-Generation Models and Libraries” on page 3-2
- “Automatic Conversion of First Generation Models and Libraries” on page 3-17

## **How To**

- “Model and Library Automatic Conversion Examples” on page 3-25

# **sdl\_update**

---



# Relationship of First and Second Generation Technologies

---

- “First-Generation Models and Libraries” on page 3-2
- “First and Second Generation Libraries” on page 3-5
- “Manual Reconstruction of First Generation Models” on page 3-13
- “Automatic Conversion of First Generation Models and Libraries” on page 3-17
- “Modification and Troubleshooting of Automatically Converted Models” on page 3-20
- “Model and Library Automatic Conversion Examples” on page 3-25

drivelib

## First-Generation Models and Libraries

In this section...
“Current First-Generation Functionality” on page 3-2
“Manual Conversion of First-Generation Models” on page 3-3
“Modification of Automatically Converted Models” on page 3-3

### Current First-Generation Functionality

SimDriveline contains two technologies: first- and second-generation. First-generation technology includes the legacy blocks of SimDriveline version 1, which are based on the Simulink language. Second-generation technology introduces a new block library that is based on the Simscape physical modeling language.

### First-Generation Models

The following list outlines the tasks you can perform with SimDriveline first-generation technology:

- Add first-generation blocks to a model.
- Modify and simulate first-generation models.
- Manually convert first-generation models to second-generation format.
- Automatically convert first-generation models to second-generation format with a conversion utility.
- Create and simulate hybrid models that contain first- and second-generation blocks.

### First-Generation User-Defined Libraries

First-generation user-defined libraries remain functional in SimDriveline software. The following list outlines the tasks you can perform with the user-defined libraries:

- Build, modify, and employ user-defined libraries that contain first-generation blocks.

- Manually convert first-generation user-defined libraries to second-generation format.
- Automatically convert first-generation models and libraries to second-generation format with a conversion utility.
- Create and change hybrid user-defined libraries that contain first- and second-generation blocks.

### **Conversion of Models with User-Defined Blocks**

If a model contains first-generation user-defined blocks, you can convert the user-defined blocks to second-generation format. Once you have converted the blocks, you can convert the model that employs the blocks. After conversion, the converted model automatically uses the converted blocks.

### **Limitations with Conversion and Multi-Generation Models**

Most first-generation blocks and block parameters have a second-generation equivalent, but exceptions exist. See “Correspondence of First and Second Generation Blocks” on page 3-8

First- and second-generation blocks connect only *indirectly*. The blocks use Simulink and Simscape signals, respectively, and you must use a converter block to convert between the two signal types. Two converter blocks exist in the Simscape Utilities library: PS-Simulink Converter and Simulink-PS Converter.

### **Manual Conversion of First-Generation Models**

Most first-generation blocks have a matching second-generation block. You can manually reconstruct a first-generation model by switching first-generation blocks with their second-generation equivalents. Manual conversion of first-generation models is more time-consuming and requires greater attention to detail. See “Manual Reconstruction of First Generation Models” on page 3-13.

### **Modification of Automatically Converted Models**

SimDriveline provides a conversion utility that you can use to automatically convert a first-generation model or library to second-generation format. The

utility carries out the conversion task with minimal input on your side. When conversion is complete, you can manually modify the model or library with additional second-generation blocks. See “Modification and Troubleshooting of Automatically Converted Models” on page 3-20.

## First and Second Generation Libraries

In this section...
“First and Second Generation Block Compatibility” on page 3-5
“First Generation Library” on page 3-6
“Second Generation Library” on page 3-6
“Transitional Library” on page 3-7
“Correspondence of First and Second Generation Blocks” on page 3-8

Second-generation technologies introduces blocks that rely on the Simscape physical modeling language. The second-generation libraries contain a matching block for nearly every first-generation block, with few exceptions. To facilitate the transition from first- to second-generation technology, SimDriveline provides a transitional block library.

### First and Second Generation Block Compatibility

The following considerations distinguish first- and second-generation blocks:

- First-generation blocks rely on Simscape software, but not on the Simscape library. Instead, the blocks exist in an independent library that relies on the Simulink language.
- Second-generation blocks rely on *both* Simscape software and library. The blocks exist in a library that reuses Simscape blocks.
- Second-generation blocks integrate fully with transitional library blocks. You can connect the blocks directly without restriction.
- First- and second-generation blocks can connect in a model or user-defined library through a Simscape converter block. Two converter blocks exist in the Simscape Utilities library:
  - PS-Simulink Converter—Convert a Simscape signal with physical units to a dimensionless Simulink signal.
  - Simulink-PS Converter—Convert a dimensionless Simulink signal to a Simscape signal with physical units.

- First- and second-generation blocks can also connect through the first-generation Rotational Coupling block. Use the block to connect a first-generation connection line to a mechanical rotational connection line.
- You cannot automatically convert second-generation models or user-defined libraries to the first-generation format.

## First Generation Library

The first-generation block library provides the basis for first-generation models and user-defined libraries. You can continue to access the first-generation library:

- At the MATLAB command line, enter `drivelib` or `driveline`.

For more information on first-generation blocks, see “First Generation Block Libraries”.



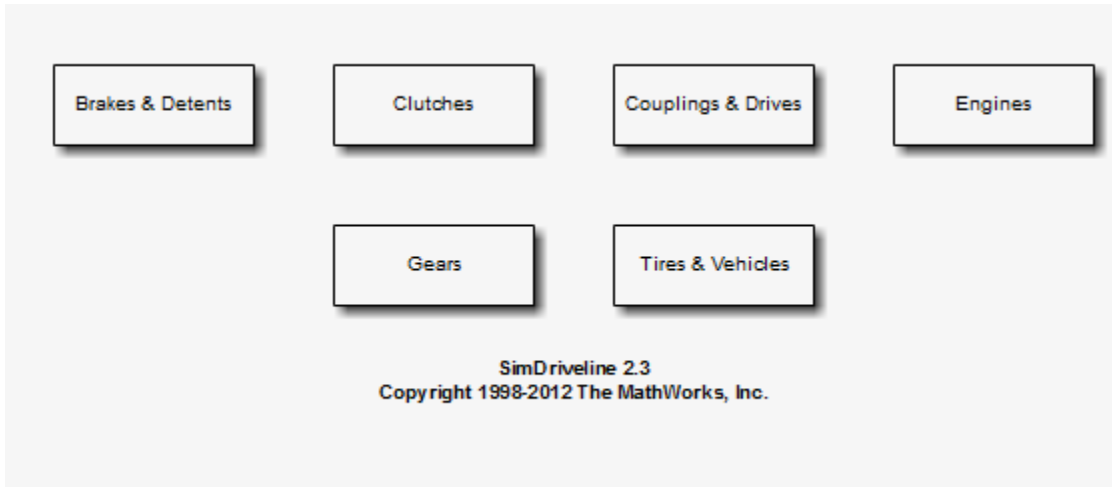
## First Generation Block Library

## Second Generation Library

The second-generation block library provides the basis for second-generation models and user-defined libraries. The library contains blocks that replicate the functionality of first-generation blocks. New blocks add functionality not present in the first-generation library.

You can access the second-generation product library:

- At the MATLAB command line, enter `sd1_lib`.



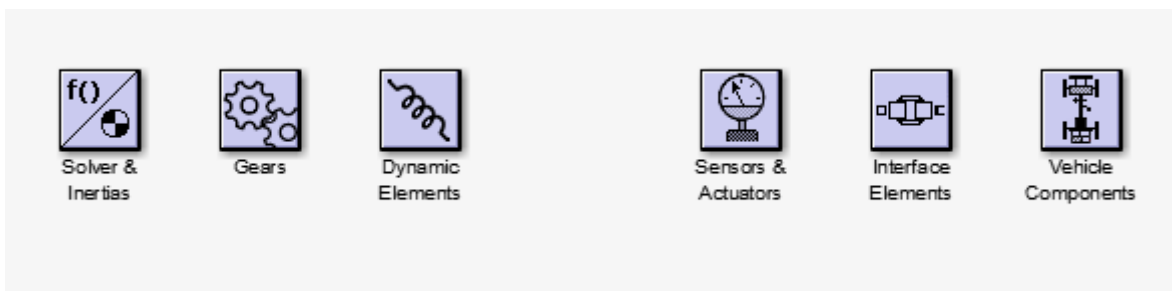
### Second Generation Block Library

## Transitional Library

SimDriveline provides a transitional block library. The library provides blocks that are functionally equivalent to first-generation blocks but compatible with second-generation blocks. Transitional library blocks are built on the blocks from the Simscape Foundation and SimDriveline Second Generation libraries.

You can access the transitional library:

- At the MATLAB command line, enter `sd1_classic`



### Transitional Library

The blocks of the transitional library are built with the Simscape language and their ports are mechanical conserving ports. This library is the bridge between the first- and second-generation products. The conversion function uses the transitional library to generate the second-generation equivalent of a first-generation model. See “How the SimDriveline Update Function Works” on page 3-17 and “About Automatically Converted Models” on page 3-18.

### **Correspondence of First and Second Generation Blocks**

- “Equivalent Blocks” on page 3-8
- “Replacement Blocks” on page 3-9
- “Missing Block Parameters” on page 3-10
- “Special Transitional Library Blocks” on page 3-11
- “Transmission Templates Blocks” on page 3-12

### **Equivalent Blocks**

The transitional library provides second-generation blocks that are functionally equivalent to first-generation blocks.

Blocks in the first-generation Gears library have exact equivalents in the second-generation Gears library. The Variable Ratio Gear is an exception. This block has no second-generation equivalent or replacement. Second-generation block Variable Ratio Transmission provides similar functionality to

The Rotational Coupling block in the first-generation Interface Elements library behaves as a Simscape physical connection line between two second-generation driveline ports.



## Replacement Blocks

**Tip** All transitional blocks can be replaced with functional equivalents using SimDriveline Second Generation and Simscape blocks. For natural and consistent Simscape implementation, MathWorks recommends manual block replacement and model reconstruction where appropriate, after automatic conversion. See “Modification and Troubleshooting of Automatically Converted Models” on page 3-20.

The following tables provide the relationship between first- and second-generation blocks.

### First-Generation Second-Generation Replacement Block

Controllable Friction Clutch	Disk Friction Clutch
Fundamental Friction Clutch	Fundamental Friction Clutch
Torque Converter	Torque Converter
Torsional Spring-Damper	Torsional Spring-Damper
Diesel Engine	Generic Engine configured to diesel ignition
Gasoline Engine	Generic Engine configured to spark ignition
Tire	Tire (Magic Formula) + related Simscape foundation components

### First-Generation Simscape Foundation Replacement Block

Driveline Environment	Solver Configuration
Housing	Mechanical Rotational Reference

**First-Generation Simscape Foundation Replacement Block**

Hard Stop	Rotational Hard Stop
Inertia	Inertia
Torque Sensor	Ideal Torque Sensor
Torque Actuator	Ideal Torque Source + Mechanical Rotational Reference
Motion Actuator	Ideal Angular Velocity Source + Mechanical Rotational Reference
Motion Sensor	Ideal Rotational Motion Sensor + Mechanical Rotational Reference

**First Generation Simulink Replacement Block**

Longitudinal Vehicle Dynamics	Pure Simulink masked subsystem
-------------------------------	--------------------------------

**Missing Block Parameters**

First-generation blocks accept input parameters that have no counterpart in transitional or second-generation blocks. Parameters without counterpart are lost during the conversion. The following table outlines the block parameters that do not carry over from first-generation blocks.

Missing block dialog field or check box	Missing block parameter
Fundamental Friction Clutch Block	
Use automatic clutch velocity tolerance for variable-step solvers	UseAutomaticVelocityTolerance
Use default clutch velocity tolerance from the Driveline Environment block	UseDefaultVelocityTolerance
Driveline Environment Block	

Missing block dialog field or check box	Missing block parameter
Default clutch velocity tolerance	DefaultClutchVelocityTolerance
Use automatic clutch velocity tolerance for variable-step solvers	DefaultUseAutoClutchVelocityTolerance
Fundamental Friction Clutch Block	
Use automatic clutch velocity tolerance for variable-step solvers	UseAutomaticVelocityTolerance
Use default clutch velocity tolerance from the Driveline Environment block	UseDefaultVelocityTolerance
Tire Block	
Relaxation length at rated load	sigma_k
Variable Ratio Gear Block	
Include Coriolis acceleration	Coriolis

### Special Transitional Library Blocks

Some first-generation blocks have no functional equivalent in SimDriveline Second Generation or Simscape Foundation libraries. Blocks without the functional equivalents appear in the transitional library as special blocks. Two special blocks exist in the transitional library: Shared Environment and Initial Condition. The two blocks integrate fully with SimDriveline Second Generation and Simscape Foundation blocks. However, unlike the remainder of the transitional library, the blocks but are built directly from *neither* library. The following table outlines the special blocks present in the transitional library.

First Generation Block	Transitional Library Block	Comments
Shared Environment	Shared Environment	—
Initial Condition	Initial Condition	The block parameter is equivalent to the <b>Initial Velocity</b> parameters of the Inertia block in the Simscape Foundation library

### **Transmission Templates Blocks**

The first-generation Transmission Templates library has no functional equivalent in the transitional or second-generation libraries. The library contains masked subsystems built from first-generation blocks that include inertia, gear and clutch elements. If you have a first-generation model with a Transmission Templates block, you can convert the model to a second-generation model. Conversion reproduces the model with the second-generation counterparts of first-generation blocks.

## Manual Reconstruction of First Generation Models

In this section...
“Automatic Conversion Versus Manual Reconstruction” on page 3-13
“Recreating Equivalent Blocks and Connections” on page 3-13
“Recreating Equivalent Parameters and Signals” on page 3-14
“Reconstruct Clutch Parameters and Signals” on page 3-15
“Recreate Specialized Model Functionality” on page 3-16

### Automatic Conversion Versus Manual Reconstruction

SimDriveline provides a conversion utility that you can use to automatically translate first-generation models and libraries to the second-generation format. The conversion utility replaces first-generation blocks with their second-generation functional equivalents. You can manually modify any automatically converted model or library. See “Automatic Conversion of First Generation Models and Libraries” on page 3-17 and “Modification and Troubleshooting of Automatically Converted Models” on page 3-20.

Manual reconstruction of a model or library is an option. During manual reconstruction of a model, you directly replace each first-generation block with a second-generation equivalent block. You must provide the appropriate input parameters to the blocks. You must also connect the blocks with the appropriate physical connection lines.

---

**Tip** Manual reconstruction of a first-generation model requires more time and attention to detail. Streamline the conversion process by using the conversion utility first. Then, after the utility automatically converts your model or library, review for accuracy. You can manually change any blocks and parameters

---

### Recreating Equivalent Blocks and Connections

Almost all first-generation blocks have equivalent second-generation blocks. With second-generation blocks, you can recreate all first-generation

functionality. For information about the mapping of first- to second-generation blocks, see “Correspondence of First and Second Generation Blocks” on page 3-8.

In a first-generation model, every driveline requires a Driveline Environment block. The second-generation equivalent is a Simscape Solver Configuration block for each physical network. In this case, a second-generation driveline is a physical network.

You recreate the first-generation driveline connection lines between first-generation blocks by connecting the equivalent second-generation blocks (based on the Simscape library) with mechanical rotational connection lines.

## **Recreating Equivalent Parameters and Signals**

Most of the equivalent second-generation blocks have parameters identical or similar to the parameters of the corresponding first-generation blocks. The primary difference is how first- and second-generation blocks implement signals.

- First-generation blocks accept and generate unitless Simulink signals. Some input signals are normalized by dividing by physical reference values with units. Output signals are converted to physical values and units by multiplying by physical reference values. These values and units are set in the blocks.
- Second-generation blocks accept and generate Simscape physical signals that carry units. The blocks accept these units, but do not implement them.

To achieve equivalent functionality in the second-generation model, you therefore might have to redefine your signals and to change how parameters are entered in a block. You can convert between Simulink signals and Simscape physical signals with Simulink-PS Converter and PS-Simulink Converter blocks. If you need to, use the Simulink Gain block to rescale signal values.

With certain blocks, there are also minor technical differences in how parameters are entered. You can determine these differences by comparing the old block and new block reference pages.

## Reconstruct Clutch Parameters and Signals

The first-generation Controlled Friction Clutch is functionally equivalent to the second-generation Disk Friction Clutch. If you reconstruct the Controlled Friction Clutch with a Disk Friction Clutch, the key parameters and signals that you need rearrange include:

- Input signal P (pressure).

The first-generation input is a normalized, unitless Simulink signal that modulates the physical value that the block parameters determine. When this signal value is 1, the kinetic friction across the clutch discs is the full kinetic friction torque implied by the block's parameters. The normalized pressure signal can have a threshold between 0 and 1.

The second-generation input is a Simscape physical signal carrying units of pressure. This signal is used directly in the calculation of the torque applied across the clutch. The physical pressure can have a threshold that you must enter as a physical value with correct units.

- Peak normal force  $F_{\max}$  in the first-generation block, which is replaced by engagement piston area  $A$  in the second-generation block. In second-generation, you multiply  $A$  by input pressure to determine peak force.
- Coefficient of friction table  $\mu$ , or mu, in the first-generation block. In the second-generation block, this option is available. But, as an alternative, you can also use a fixed friction coefficient.

To replace a Controlled Friction Clutch with a Disk Friction Clutch, using the same normalized Simulink signal for pressure:

- 1 Place a Gain block and a Simulink-PS Converter block, connected in that order, between the normalized input signal and the Disk Friction Clutch pressure port P.
- 2 Set the gain value to  $F_{\max}/A$ . In the Simulink-PS Converter block, set the **Input signal unit** to Pa (pascal). (If you use a different pressure unit in the Disk Friction Clutch, use that unit here instead.)

Map the old clutch to the new clutch parameters:

- 1 In the new block, set **Engagement threshold pressure** to  $F_{\max}/A$  times the value of **Engagement threshold for normalized pressure** in the old block.
- 2 If the old mu table parameter is a 2-by-2 matrix, and the two friction coefficient values are the same, then set the new **Kinetic friction coefficient** to the old friction coefficient value. Set the new **Static friction coefficient** to the old **Static friction peak factor** times this value.  
  
Otherwise, in the new block, select the Table lookup kinetic friction coefficient option for **Friction model**. Set **Kinetic friction coefficient relative velocity vector** to  $\mu(:, 1)$  and **Kinetic friction coefficient vector** to  $\mu(:, 2)$ .
- 3 Copy exactly the remaining parameters, or leave them as default in the new clutch.

## Recreate Specialized Model Functionality

First-generation models can have specialized features. You might have to perform additional tasks to recreate these features in second-generation format. Some issues to consider are reconstructing initial conditions, modeling physics, removing algebraic loops, and taking advantage of Simscape features not available in first-generation, such as local solvers.

These specialized modeling issues are similar to manually optimizing automatically converted models after translation. For additional guidance, see “Best Practices with Automatically Converted Models” on page 3-22.



# Automatic Conversion of First Generation Models and Libraries

## In this section...

“What Automatic Conversion Does” on page 3-17

“How the SimDriveline Update Function Works” on page 3-17

“About Automatically Converted Models” on page 3-18

## What Automatic Conversion Does

- You can convert models based on first-generation blocks to new models based on second-generation blocks.
- You can also convert existing user-defined libraries based on first-generation blocks to new user-defined libraries based on second-generation blocks.
- In both cases, conversion generates new models and libraries built from the transitional second-generation library.
- Conversion directly involves the first-generation product block library and the second-generation transitional library, but not the second-generation product library.

## How the SimDriveline Update Function Works

The `sd1_update` function converts first-generation models and user-defined libraries to second-generation equivalents. To convert a model, at the command line, enter:

```
newmdlname = sd1_update('modelname');
```

If the original model is open, the new one is open. If the original model is closed, the function loads the original model, then creates and loads the new model, without opening it. The new model is saved in your current directory, and its name is stored in the output string variable; in this case, `newmdlname`.

For additional details, see “Model and Library Automatic Conversion Examples” on page 3-25 and the `sd1_update` function reference page.

#### **Converting User-Defined Libraries**

To convert a first generation user-defined library, use the same function form used to convert a first-generation model.

#### **Converting Models Linked to User-Defined Libraries**

To convert a first-generation model linked to one or more user-defined libraries:

- 1 Convert the linked libraries.
- 2 Enter the function form specifying not only the model to convert, but both the first- and second-generation linked libraries:

```
newmdlname = sdl_update('modelname', 'oldlib1', 'newlib1', ...);
```

#### **About Automatically Converted Models**

“Transitional Library” on page 3-7 presents the characteristics of the transitional second-generation library and its relationship to the main second-generation product library.

#### **Looking Under Converted Block Masks**

The transitional library blocks are masked subsystems. You can view the underlying blocks by looking under the masks.

- Some first-generation blocks have exact second-generation equivalents, and the masked subsystems wrap the second-generation blocks into forms that look like the first-generation blocks.
- Upon conversion, first-generation blocks without exact second-generation equivalents are recreated as replacement masked subsystems built from second-generation, Simscape Foundation, or pure Simulink components. The replacement subsystems consist of multiple blocks that function the same way the corresponding first-generation block does.

#### **First-Generation Blocks Without Second-Generation Equivalents or Replacements**

For first-generation blocks without subsystem replacements or exact equivalents, the underlying subsystems in the converted blocks are built

from Simscape software, but not directly on the Simscape or SimDriveline Second Generation library.

### **Transmission Subsystem Conversion**

Upon conversion, first-generation Transmission Templates are recreated as unmasked subsystems built from second-generation blocks. The conversion translates below the subsystem level, at the level of individual blocks (gears, clutches, inertias, etc.) that are linked to the first-generation library.

## Modification and Troubleshooting of Automatically Converted Models

In this section...
“Manually Revise and Optimize Converted Models” on page 3-20
“Unlink and Replace Transitional Library Blocks” on page 3-21
“Best Practices with Automatically Converted Models” on page 3-22
“Transitional Library Input Handling Options” on page 3-23

### Manually Revise and Optimize Converted Models

The most important differences between first- and second-generation arise from Simscape features that include:

- SimDriveline components with realistic loss, threshold, and compliance dynamics
- Simscape Foundation library
- Physical signals with units
- Local solvers
- Data logging

The practical impact of these Simscape-based features lies in how you optimally build more complex physical models. Once you have automatically updated a first-generation model to second-generation format, you may want or need to manually revise it to take advantage of and better conform to the Simscape environment.

---

**Note** Manual optimization of a converted model sometimes requires detailed understanding of how equivalent second-generation blocks replace first-generation blocks. To learn in detail how this mapping works, see “Manual Reconstruction of First Generation Models” on page 3-13.

---

## Unlink and Replace Transitional Library Blocks

Because the converted model uses the transitional library `sdl_classic`, the model does not have to appear or be constructed as it would if you had built it from the start with the SimDriveline 2 and Simscape product libraries. Follow some or all of these steps to make the converted model more naturally fit the second-generation, Simscape-based environment.

## Disabling and Breaking Transitional Library Links—Unmasking Subsystems

You use the block context menus on the masked subsystems for these tasks. For more about this menu, see “Masking”.

- Because the converted blocks are masked subsystems, you can free the converted model from the transitional library by disabling and optionally breaking the converted block links to `sdl_classic`.
- You can also remove a subsystem mask entirely.
  - 1 Open the block dialog box and edit the subsystem mask to determine how the block dialog parameters are referenced in the underlying subsystem.
  - 2 Replace the block dialog parameters in the underlying subsystem with the model-level parameter values or variables.
  - 3 Unmask the subsystem.

---

**Tip** Before removing the mask, be sure to locate and copy the subsystem dialog parameter values or variables into the underlying subsystem blocks. You cannot retrieve these values or variables once you have removed the mask.

---

## Reconstructing Converted Blocks – Promoting Subsystem Blocks

Depending on your model appearance and construction, decide whether or not you want to promote unmasked subsystem blocks higher up in the model hierarchy.

You can rearrange and, where possible, replace or modify these underlying Simscape and SimDriveline 2 blocks, using the regular product libraries only. For block equivalents and replacements, consult “Transitional Library” on page 3-7.

#### **Reconstructing Driveline Initial Conditions**

There is no equivalent to the first-generation Initial Condition block. In a Simscape-based model, you instead set initial conditions on some blocks individually, while other blocks have no initial conditions. You must ensure that the physical network as a whole has a consistent initial state.

For more about initial conditions, see “Simulation Setup”.

#### **Best Practices with Automatically Converted Models**

To achieve optimal simulation performance and avoid simulation problems and failures with converted models, verify that your model conforms with these best practices. If it does not conform, modify the model.

- “Reconstructing Custom Physical Modeling” on page 3-22
- “Avoiding Algebraic Loops” on page 3-23
- “Changing Global Solvers — Adding Optional Local Solvers” on page 3-23

#### **Reconstructing Custom Physical Modeling**

Consider rebuilding how you model physical processes in your converted model. In the Simscape environment, the order of preference for physical modeling is to use:

- 1** Product library blocks in the SimDriveline and Simscape Foundation libraries
- 2** Simscape language
- 3** Physical Signal blocks, if the Simscape language is not adequate
- 4** Simulink signals and blocks with sensors and actuators, making sure to break any algebraic loops (see “Avoiding Algebraic Loops” on page 3-23)

For guidance on how to customize and specialize your driveline components, see “Specialized and Customized Driveline Components”. Continue using Simulink for algorithmic control, nonphysical signals, and model-level input/output.

- In first-generation, you model custom and specialized physical components in Simulink, connecting to SimDriveline blocks through sensors and actuators with nonphysical Simulink signals.
- In second-generation, you accomplish the same goal with Simscape-based components and language, bringing all physical modeling into the Simscape environment.

### **Avoiding Algebraic Loops**

Simscape simulation treats differential and algebraic constraint equations similarly. To the extent possible in your model, represent algebraic constraints between physical variables with Simscape-based components. Avoid physical signal algebraic loops that are fed back through Simulink. That is, avoid instantaneous, noncausal relationships between physical variables.

For more about algebraic loops, see “Algebraic Loops”.

### **Changing Global Solvers — Adding Optional Local Solvers**

The default Simulink global solver is ode45. Most physical models in the Simscape environment are best simulated using ode23 or ode15s. In addition, you can optionally use the fixed-step Simscape local solvers with Simulink global solvers to improve physical simulation.

For more information about selecting global and local solvers for physical simulations, see:

- “Driveline Simulation Performance”
- “Simulation”

### **Transitional Library Input Handling Options**

Transitional library blocks that convert Simulink signals into Simscape signals provide input handling control. The block dialog box contains an **Input Handling** tab that you can use to select a filtering type for the input

signal. You can select no filtering, first-order filtering, or second-order filtering. Alternatively, you can choose to specify up to two input derivatives. The **Input Handling** tab mirrors the Simulink-PS Converter. For more information, see “Input Handling”.

Only transitional library blocks that convert a Simulink signal into a physical signal provide an **Input Handling** tab. The blocks contain one or more Simulink inports and one or more Simscape physical signal outports. If the block contains multiple Simulink input ports, the input handling options you specify apply equally to all ports. If you want to apply different input handling options to each input port, open the block subsystem and specify the input handling options for each Simulink-PS Converter block directly.

Model topology and solver choice determine the number of input derivatives required for a solution. You have the option to specify the first- and second-order derivatives, but if you fail to specify enough input derivatives, Simscape issues an error. Errors caused by insufficient input derivatives point to the Simulink-PS Converter block used under the mask of the transitional library block.

The following table lists the transitional library blocks that provide **Input Handling** options.

<b>Library</b>	<b>Block</b>
Gears	Variable Ratio Gear
Dynamic Elements	Controllable Friction Clutch
	Fundamental Friction Clutch
Sensors & Actuators	Torque Actuator
	Motion Actuator
Vehicle Components	Diesel Engine
	Gasoline Engine
	Tire



## Model and Library Automatic Conversion Examples

### In this section...

“About the Automatic Conversion Examples” on page 3-25

“Convert Example Model” on page 3-25

“Convert Example Library” on page 3-25

“Convert Example Model Linked to Example Library” on page 3-26

### About the Automatic Conversion Examples

These conversion examples use two example models and one example library from first-generation. Models and libraries do not have to be open before you start converting them.

- `drive_sgear` — Standalone example model
- `drive_efficiency` — Examples library, similar to a user-defined library
- `drive_sgear_lossy` — Example model linked to `drive_efficiency`

---

**Tip** Proceed through the following examples in the order presented for complete coverage on how to use `sd1_update`.

---

### Convert Example Model

At the command line, enter:

```
v2mdlname1 = sd1_update('drive_sgear');
```

The second-generation model is generated in the current directory, and its name is stored in the string variable `v2mdlname1`.

### Convert Example Library

At the command line, enter:

```
v2libname = sd1_update('drive_efficiency');
```

The second-generation library is generated in the current directory, and its name is stored in the string variable `v2libname`.

## Convert Example Model Linked to Example Library

---

**Note** Complete the preceding example before attempting this one.

---

At the command line, enter:

```
v2mdlname2 = sd1_update('drive_sgear_lossy','drive_efficiency',  
v2libname);
```

This form of the function specifies the first-generation library name as a literal string, but the new library name as a string variable. It uses the same string variable as in the preceding example.

# Technical Conventions

---

## Driveline Abbreviations and Conventions

In this section...
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### Abbreviations

An important abbreviation is DoF, which means *degree of freedom* and refers to one coordinate of angular motion. All driveline rotational DoFs are measured with respect to a single absolute coordinate system at rest.

### Angular Motion

Standard symbols for angular motion analysis include the following:

Symbol	Meaning (Units)
$r$	Gear radius
$N$	Number of gear teeth
$g$	Gear ratio
$\theta$	Angle
$\omega$	Angular velocity
$\tau$	Torque

### Gear Ratios

For a pair of coupled, coplanar gear wheels, the gear ratio  $g_{21}$  of gear 2 to gear 1 is defined as the ratio of the second gear wheel radius to the first. This definition is equivalent to the ratio of the number of teeth on the second gear wheel to the number of teeth on the first.

$$g_{21} \equiv r_2/r_1 = N_2/N_1$$

The gear ratio is the ratio of torques and the reciprocal of the angular velocity ratio.

$$g_{21} = \tau_2/\tau_1 = \omega_1/\omega_2$$

For gear boxes made of more than two gear wheels, the gear ratio is defined to be the ratio of torques or the reciprocal of the ratio of angular velocities, between the output and input shafts.

If the gear is reversing, the output  $\omega$  and  $\tau$  have opposite signs from the input  $\omega$  and  $\tau$ .



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