SimDriveline™
Reference

R2014**a**
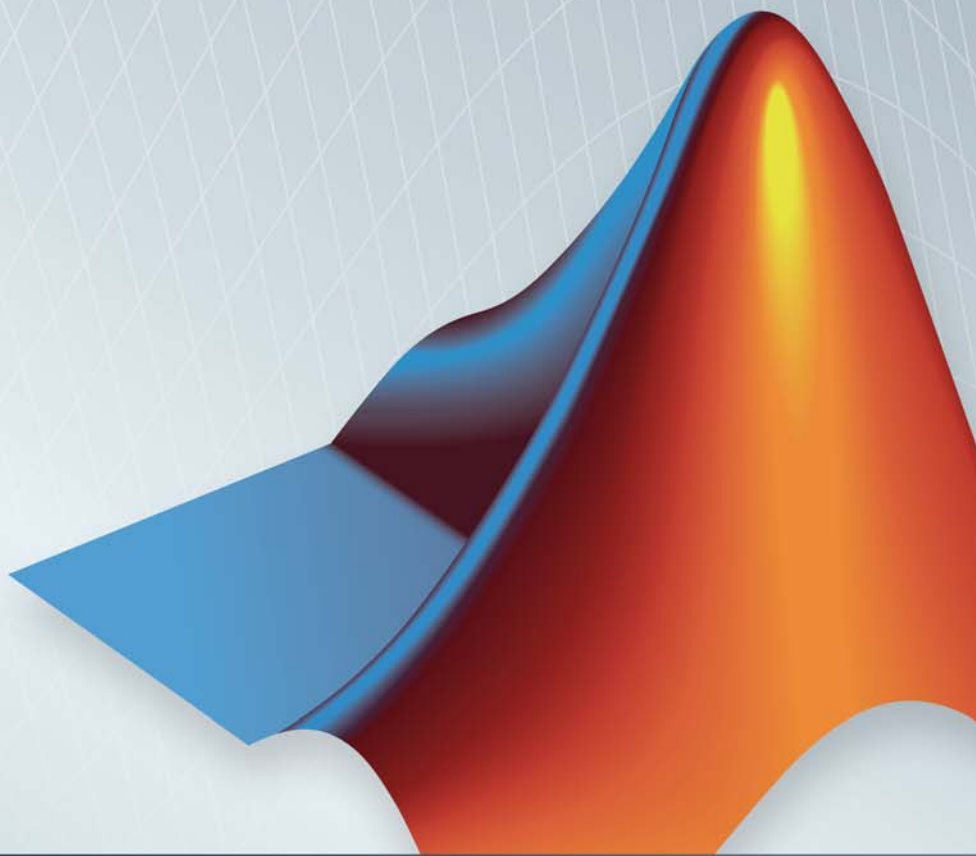
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SimDriveline™ Reference

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Technical Conventions

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Blocks — Alphabetical List

Band Brake

Purpose   Frictional brake with flexible band wrapped around rotating drum

Library   Brakes & Detents/Rotational
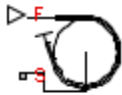
Description The block represents a frictional brake with a flexible band that wraps
around the periphery of a rotating drum to produce a braking action. A
positive actuating force causes the band to tighten around the rotating
drum and it places the friction surfaces in contact. Viscous and contact
friction 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.
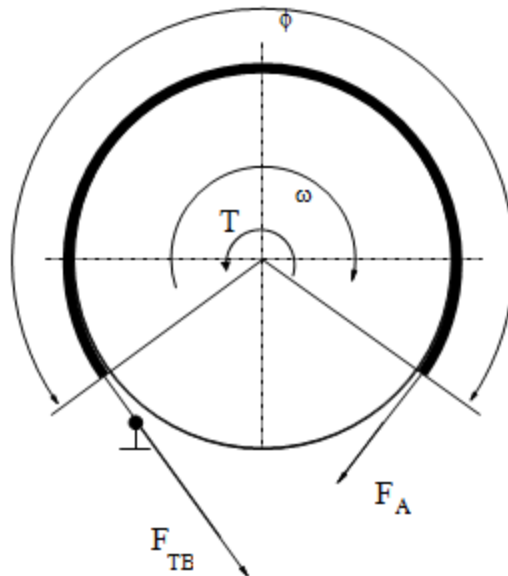
The following formula provides the braking torque as a function of the
external brake actuation force that tightens the belt.

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

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
μ_{visc}	Viscous friction coefficient
μ	Contact friction coefficient
φ	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}:

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

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

$$F_A = F_{in} \cdot \tanh\left(\frac{4\omega}{\omega_{threshold}} \right),$$

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

Block Parameters: Band Brake

Band Brake

Represents a brake arranged as a band wrapped around a drum and tightened with an external force. This force causes the band to exert friction torque on the shaft.

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

Settings

| Geometry | Friction |

Drum radius: 150   mm

Wrap angle: 270   deg

[OK]   [Cancel]   [Help]   [Apply]
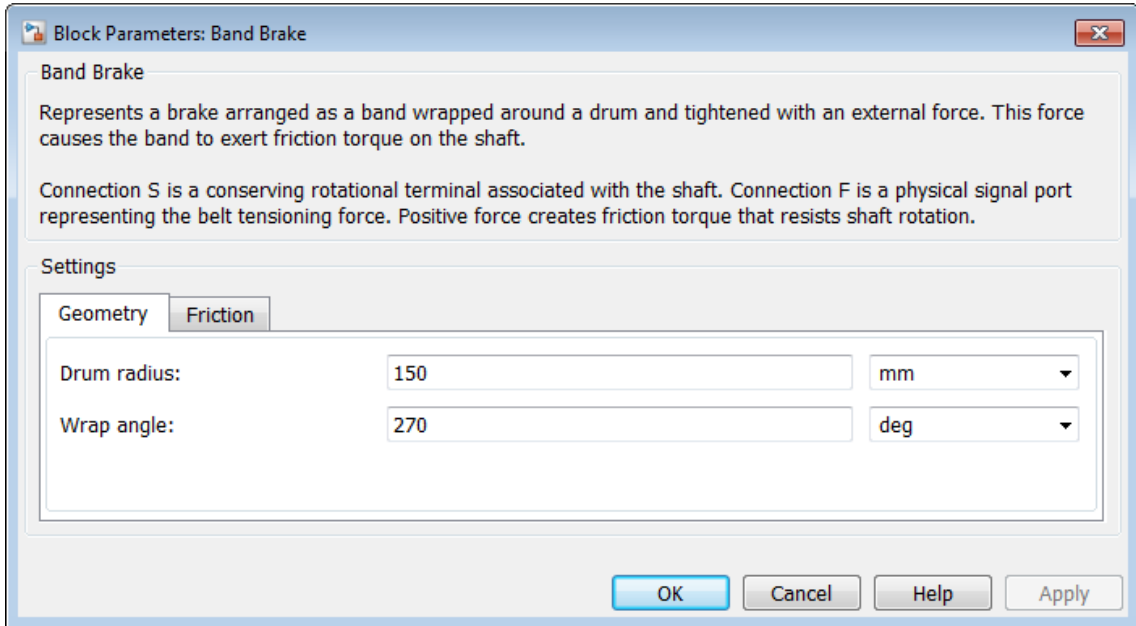
Geometry

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.

Band Brake

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.

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   Power transmission system with taut belt connecting two pulleys

Library   Couplings & Drives
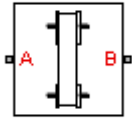
Description   The block represents a pair of pulleys connected with a flexible flat band or V-shaped belt. The model accounts for friction between the flexible belt and the pulley periphery. If the friction force is not sufficient to drive the load, the model allows slip. The relationship between the 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.

Open Belt Configuration   Crossed Belt Configuration

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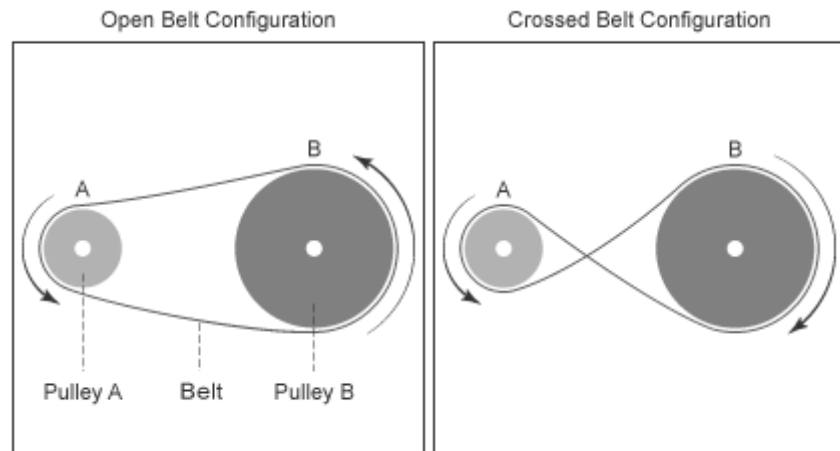
In this schematic, a) represents an open belt configuration, and b) represents a crossed belt configuration. In the open belt configuration, both pulleys tend to rotate in the same direction, and the larger pulley tends to have a larger belt wrap angle. In the crossed belt configuration, the pulleys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

Belt Drive

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

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

- θ_A is the wrap angle of pulley A.
- θ_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.
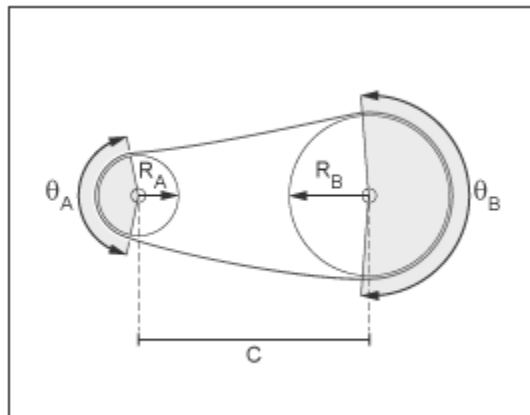
The wrap angles for crossed belt configurations use this equation:

$$\theta_A = \theta_B = \pi + 2 * \sin^{-1} \frac{R_A + R_B}{C}$$
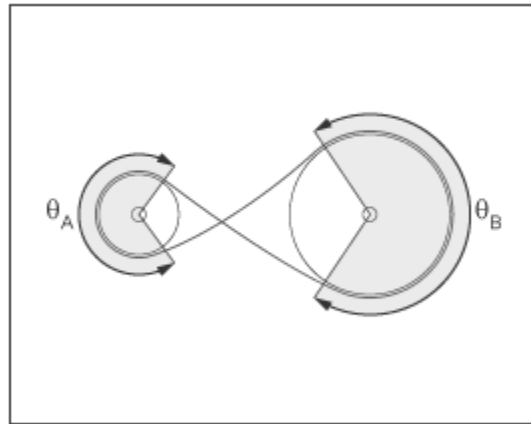
This schematic shows the open band configuration.

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.

Belt Drive

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
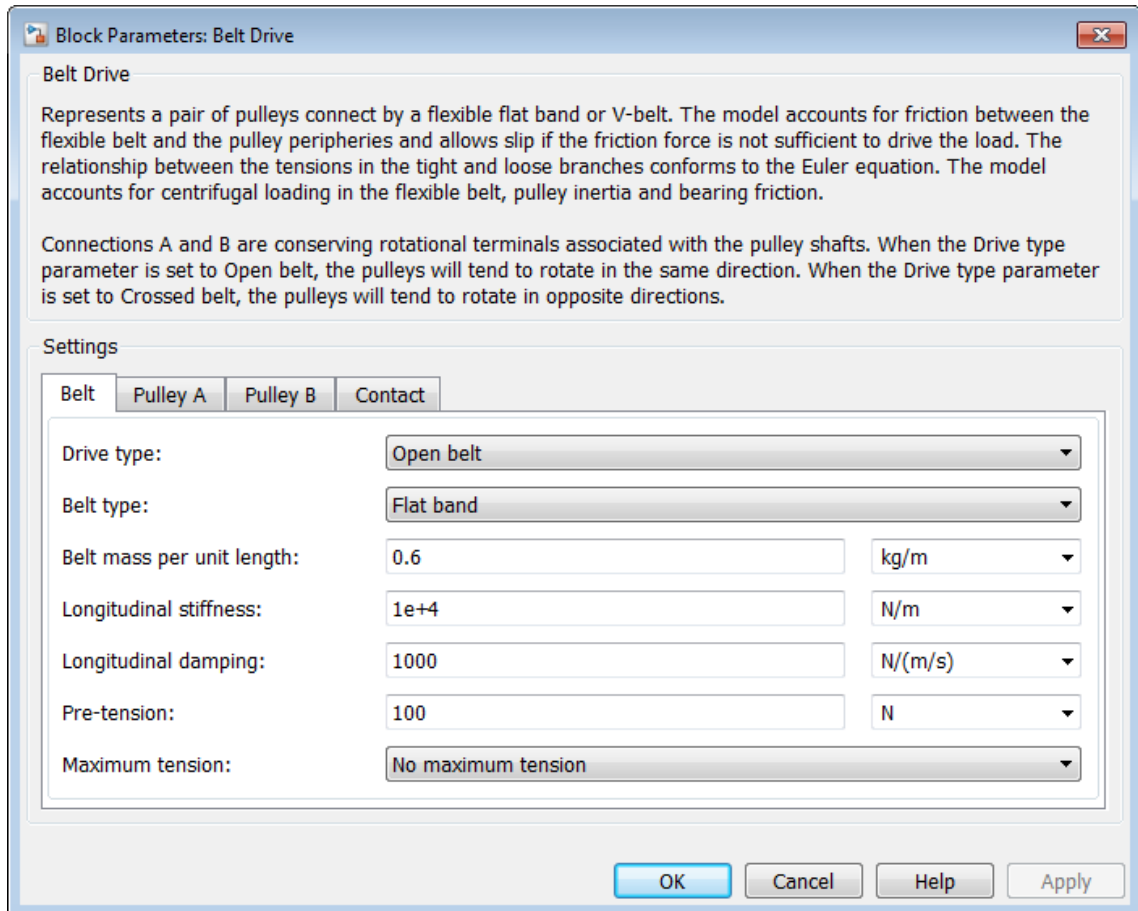
Drive type

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

Belt Drive

- `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.

Belt Drive

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^2` for pulley A, and `0.03 kg*m^2` 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.

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

Port	Description
A	Conserving rotational port associated with the shaft of pulley A
B	Conserving rotational port associated with the shaft of pulley B

Belt Drive

See Also   Chain Drive | Belt Pulley

Concepts   • "Rotational-Translational Couplings"

Purpose   Power transmission element with frictional belt wrapped around pulley circumference

Library   Couplings & Drives
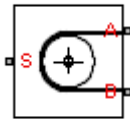
Description   This block represents a pulley wrapped in a flexible flat band or V-shaped belt. The model accounts for friction between the flexible belt and the pulley periphery. If the friction force is not sufficient to drive the load, the model allows slip. The relationship between the tensions in 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.
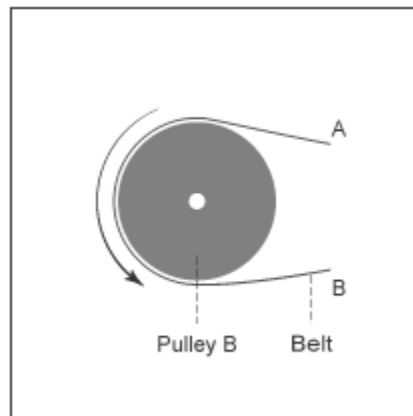
The block dialog box enables you to select the relative belt direction of motion. Depending on the chosen setting, the two belt ends can move in equal or opposite directions. The block model assumes noncompliance in the belt and no resistance to motion due to wrapping around the pulley.

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

Belt Pulley

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

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

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

$$V_A = -V_B$$

- V_{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.

- ω_S is the pulley angular velocity.

- R is the pulley radius.

- F_C is the belt centrifugal force.

- ρ is the belt linear density.

- F_{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.

- θ 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:

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

- f' is the effective friction coefficient for a V-belt.

- Φ is the V-belt sheave angle.

The idealization of the discontinuity at $V_{rel} = 0$ 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:

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

- μ 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 μ.

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 block equations model power transmission between the belt branches or to/from the pulley. The tight and loose branches use the

Belt Pulley

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 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

Block Parameters: Belt Pulley

Belt Pulley

Represents a pulley wrapped with a flexible flat band or V-belt. The model accounts for friction between the flexible belt and the pulley periphery and allows slip if the friction force is not sufficient to drive the load. 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.

Connection S is a conserving rotational terminal associated with the pulley shaft. Connections A and B are the conserving translational terminals associated with the belt branches. When the angular velocity at S is positive, the translational velocity at A is negative. The Belt direction parameter determines if the translational velocity at B is in the same or opposite direction as A.

Settings

| Belt | Pulley | Contact |

Belt type: Flat band

Belt mass per unit length: 0.6   kg/m

Belt direction: Ends move in opposite direction

Maximum tension: No maximum tension

Tension warning: Do not check belt tension

OK   Cancel   Help   Apply

Belt

Belt type

Type of flexible element modeled.

- Flat band — (Default)

Belt Pulley

- `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. The default value 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.

Belt direction
Relative direction of translational motion of one belt end with respect to the other. Options include:

- `Ends move in opposite direction`
- `Ends move in same direction`

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`.

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

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 kg*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 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

Contact friction coefficient

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

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

Concepts   • "Rotational-Translational Couplings"

Chain Drive

Purpose   Power transmission system with chain and two sprockets

Library   Couplings & Drives
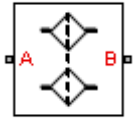
Description   This block represents a power transmission system with a chain and
two sprockets. The chain meshes with the sprockets, transmitting
rotary motion between the two. Power transmission can occur in
reverse, i.e., from driven to driver sprocket, due to external loads. This
condition is known as back-driving.

Sprocket A   Drive Chain   Sprocket B
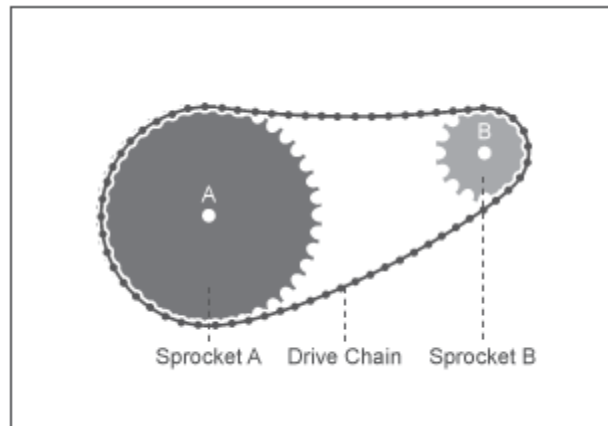
The drive chain is compliant. It can stretch under tension and slacken
otherwise. The compliance model consists of a linear spring-damper set
in a parallel arrangement. The spring resists tensile strain in the chain.
The damper resists tensile motion between chain elements.

The spring and damper forces act directly on the sprockets that the
chain connects. The spring force is present when one chain branch
is taut. The damper force is present continuously. If you specify a
maximum tension value, when the net tensile force in the chain exceeds
this value, simulation stops with an error.

The block accounts for viscous friction at the sprocket joint bearings.
During motion, viscous friction causes power transmission losses,

reducing chain-drive efficiency. These losses compound those due to chain damping. Setting viscous friction and chain damping to zero eliminates power transmission losses in the chain drive.

The tensile strain rate in the chain is the difference between the sprocket tangential velocities, each the product of the angular velocity and pitch radii. Mathematically,

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

where:

- x is the tensile strain.
- ω_A, ω_B are the sprocket angular velocities.
- R_A, R_B are the sprocket pitch radii.

The figure shows the relevant equation variables.
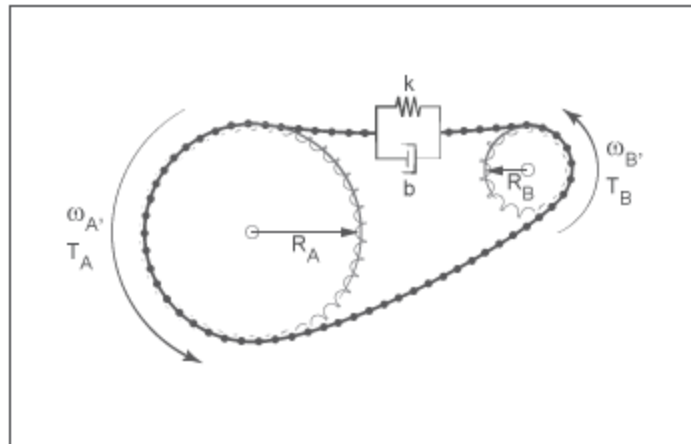
The chain tensile force is the net sum of spring and damper forces. The spring force is the product of the tensile strain and the spring stiffness constant. This force is zero whenever the tensile strain is smaller than

Chain Drive

the chain slack. The damper force is the product of the tensile strain rate and the damping coefficient. Mathematically,

$$
F = \begin{cases} -\left(x - \dfrac{S}{2}\right)k - \dot{x}b, & x > \dfrac{S}{2} \\[2ex] -\dot{x}b, & \dfrac{S}{2} \geq x \geq -\dfrac{S}{2}, \\[2ex] -\left(x + \dfrac{S}{2}\right)k - \dot{x}b, & x < -\dfrac{S}{2} \end{cases}
$$

where:

- S is the chain slack.

- k is the spring stiffness constant.

- b is the damper coefficient.

The chain exerts a torque on each sprocket equal to the product of the tensile force and the sprocket pitch radius. The two torques act in opposite directions according to the equations

$$T_A = -F \cdot R_A,$$

and

$$T_B = F \cdot R_B,$$

where:

- T_A is the torque that the chain applies on sprocket A.

- T_B is the torque that the chain applies on sprocket B.

Assumptions and Limitations

- The sprocket tooth ratio equals the sprocket pitch radius ratio.

- Chain inertia is negligible.

Dialog Box and Parameters

Block Parameters: Chain Drive

Chain Drive

Represents a drive built of two sprockets connected by a roller chain. The model accounts for slack in the chain and can be backdriven.

The drive ratio is the ratio of the sprocket A pitch radius to the sprocket B pitch radius. The model allows the chain to flex and accounts for chain stiffness and damping along with viscous damping in the sprocket bearings. All damping terms can be set to zero.

If a maximum chain tension is specified and the chain tension exceeds this maximum value, simulation stops with an error.

Connections A and B are the mechanical rotational conserving ports associated with sprocket A and sprocket B, respectively.

Settings

| Geometry | Dynamics | Maximum tension |

Sprocket A pitch radius:	80	mm
Sprocket B pitch radius:	40	mm
Chain slack length:	50	mm
Chain initial displacement:	0	mm

OK   Cancel   Help   Apply

Geometry

Specify the sprocket and chain dimensions.

Chain Drive

Sprocket A pitch radius

Radius of the sprocket A pitch circle. The pitch circle is an imaginary circle passing through the contact point between a chain roller and a sprocket cog at full engagement. The default value is 80 mm.

Sprocket B pitch radius

Radius of the sprocket B pitch circle. The pitch circle is an imaginary circle passing through the contact point between a chain roller and a sprocket cog at full engagement. The default value is 40 mm.

Chain slack length

Maximum distance the loose branch of the drive chain can move before it is taut. This distance equals the length difference between actual and fully taut drive chains.

If one sprocket is held in place while the top chain branch is taut, then the slack length is the tangential distance that the second sprocket must rotate before the lower chain branch becomes taut. The default value is 50 mm.

Chain initial displacement

Chain displacement from the neutral position at time zero. In the neutral position (the default value) the chain can move in either direction by half the chain length before it is taut. The default value is 0 mm.

Dynamics

Specify the internal forces acting in the chain and on the sprockets.

Chain stiffness

Linear spring constant in the chain compliance model. This constant describes the chain resistance to strain. The spring element accounts for elastic energy storage in the chain due to deformation. The default value is 1e+5 N/m.

Chain damping

Linear damping coefficient in the chain compliance model. This coefficient describes the chain resistance to tensile motion between adjacent chain elements. The damper element accounts for power losses in the chain due to deformation. The default value is 5 N/(m/s).

Viscous friction coefficient of sprocket A

Friction coefficient due to the rolling action of the sprocket A joint bearing in the presence of a viscous lubricant. The default value is 0.001 N*m/(rad/s).

Viscous friction coefficient of sprocket B

Friction coefficient due to the rolling action of the sprocket B joint bearing in the presence of a viscous lubricant. The default value is 0.001 N*m/(rad/s).

Maximum tension

Specify the upper tension limit in the drive chain.

Maximum tension

Select whether to constrain the maximum tensile force in the drive chain.

- No maximum tension — Chain tension can be arbitrarily large during simulation.

- Maximum tension — Chain tension must remain lower than a maximum value. Simulation stops with an error when tension exceeds this value.

Chain maximum tension

Maximum allowed value of the tensile force acting in the chain. The default value is 1e+6 N.

Chain Drive

Ports	

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

Example   Open sdl_sheet_metal_feeder for an example using the Chain Drive block.

See Also   Simple Gear | Variable Ratio Transmission

Related Examples   • "Gear Coupling Control with Clutches"

Purpose   Planetary gear train with stepped planet gear set

Library   Gears
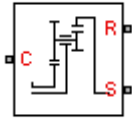
Description   This block represents a planetary gear train with a set of stepped planet gears. Each stepped planet gear consists of two rigidly connected gears possessing different radii. The larger gear engages a centrally located sun gear, while the smaller gear engages an outer ring gear.
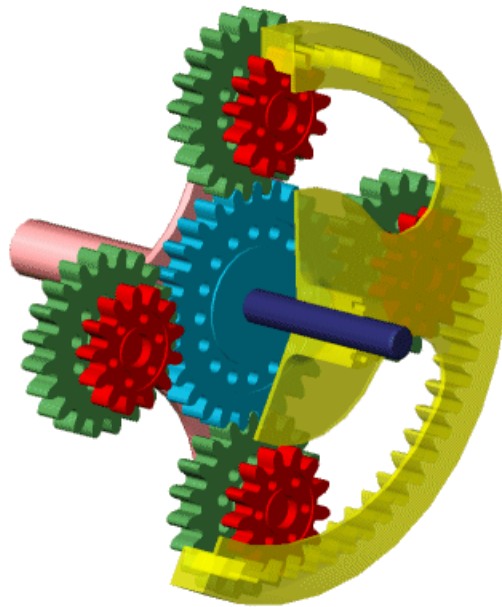
The stepped planet gear set enables a larger speed-reduction ratio in a more compact geometry than an ordinary planetary gear can provide. The compound reduction ratio depends on two elementary reduction ratios, those of the sun-large planet and ring-little planet gear pairs. Because of this dependence, compound planetary gears are also known as dual-ratio planetary gears. For more information, see "Compound Planetary Gear Model" on page 1-37.

Compound Planetary Gear

This block is a composite component with two underlying blocks:

- Sun-Planet
- Ring-Planet

The figure shows the connections between the two blocks.
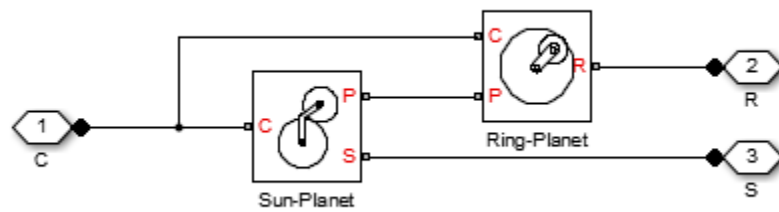
Dialog Box and Parameters
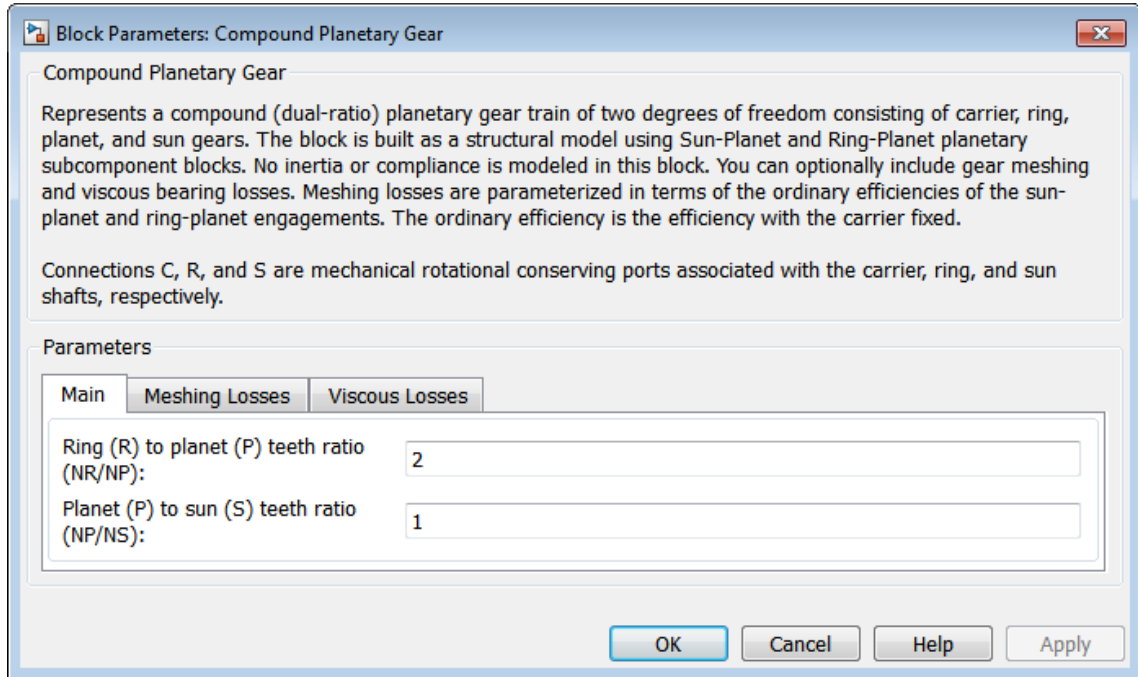
Main

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.

Compound Planetary Gear

Meshing Losses

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 η satisfying $0 < \eta$
  ≤ 1. If you select this option, the panel changes from its default.

Constant Efficiency

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 ω_{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

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):

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

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

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:

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

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:

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

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 inertia is negligible. It does not impact gear dynamics.

- Gears are rigid. They do not deform.

- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

Ports

- C — Rotational conserving port representing the planet gear carrier.

Compound Planetary Gear

- R — Rotational conserving port representing the ring gear.

- S — Rotational conserving port representing the sun gear.

See Also   Planetary Gear | Ravigneaux Gear | Ring-Planet | Sun-Planet

Related Examples
- "Represent and Transfer Driveline Motion and Torque"

Purpose   Slip clutch based on friction between conical surfaces

Library   Clutches
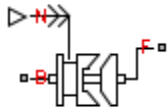
Description   This block represents a friction clutch with a conical contact interface. The conical interface creates a wedging action between the clutch components, a cone and a cup, thereby reducing the normal force required for clutch engagement.

The cup component connects rigidly to the drive shaft, spinning with it as a unit. The cone component connects rigidly to the driven shaft, which sits in axial alignment with the drive shaft. The clutch engages when the cone slides toward the cup and presses tightly against its internal surface.

Friction at the cone-cup contact interface enables the clutch to transmit rotational power between the drive and driven shafts. The friction model of this block includes both static and kinetic friction contributions, the latter of which leads to power dissipation during slip between the cone and cup components.
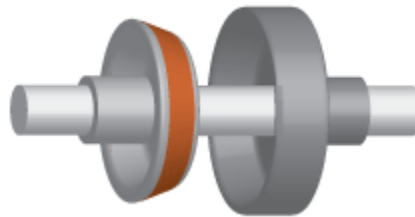
Cone clutches find real-world application in synchromesh gearboxes, which synchronize the drive and driven shaft speeds to enable smoother engagement between transmission gears. For block model details, see "Cone Clutch Model" on page 1-44.

Ports

B and F are rotational conserving ports representing, respectively, the clutch input (base) and output (follower) driveshaft axes. The clutch

Cone Clutch

motion is measured as the *slip* $\omega = \omega_{\mathrm{F}} - \omega_{\mathrm{B}}$, 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.

Dialog Box and Parameters
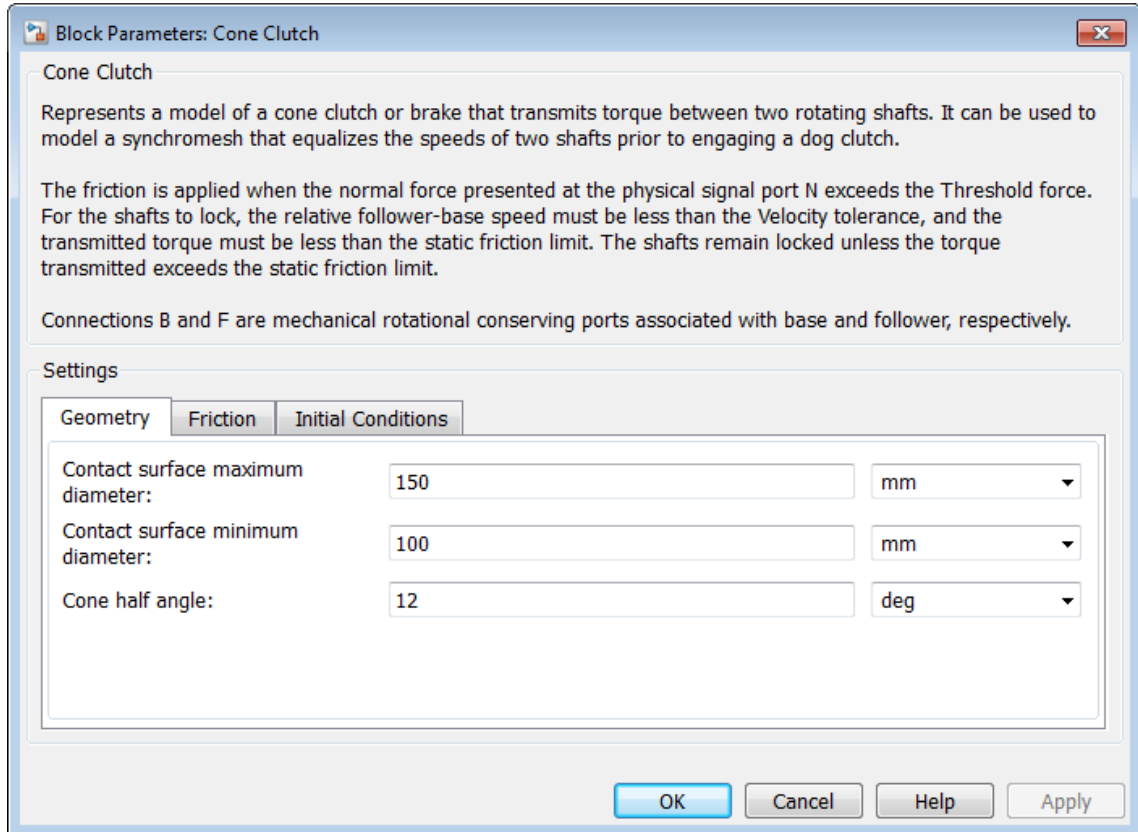
Dimensions

Contact surface maximum diameter

The outer conical diameter d_o. The default is 150.

Cone Clutch

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

Contact surface minimum diameter
The inner 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 α of the cone geometry. The default is 12.

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

Friction

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

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

Kinetic friction coefficient relative velocity vector

Vector of clutch slip values ω 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.

Velocity tolerance

Sets the minimum angular speed ω_{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).

Threshold force

The minimum normal force F_{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

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 shows the cone clutch geometry and some model parameters. Refer to the table for a summary of variable descriptions.
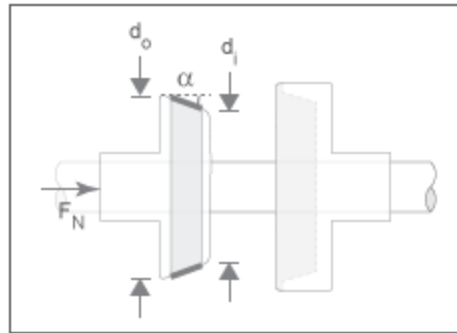
Clutch Variables

Parameter	Definition	Significance
d_o	Outer diameter of the conical contact surface	*See the preceding figure*
d_i	Inner diameter of the conical contact surface	*See the preceding figure*
a	Cone half angle	*See the preceding figure*
ω	Relative angular velocity (slip)	$\omega_F - \omega_B$
ω_{Tol}	Slip tolerance for clutch locking	*See the following model*
F_N	Normal force applied to conical surfaces	Normal force applied, if greater than threshold: $F_N > F_{th}$
a	Cone half-angle	*See the preceding figure*
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 ω.

Cone Clutch

Clutch Variables (Continued)

Parameter	Definition	Significance
k_S	Static friction coefficient	Dimensionless coefficient of static friction of conical friction surfaces.
τ_K	Kinetic friction torque	*See the following model*
τ_S	Static friction torque limit	(static friction peak factor)·(kinetic friction torque for $\omega \rightarrow 0$) (*See the following model*)

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:

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

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

You specify the *kinetic friction coefficient* k_K as either a constant or a tabulated discrete function of relative angular velocity ω. 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 ω only.

The *effective torque radius* r_{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 ω to zero and replacing the kinetic with the static friction coefficient:

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

$k_S > k_K$, so that the torque τ 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:

$$\tau_S \equiv \tau_S{}^+ = -\tau_S{}^- \; .$$

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

Cone Clutch

Related Examples

- "Gear Coupling Control with Clutches"

Purpose   High-ratio speed reducer based on cycloidal disk motion

Library   Gears

Description   This block represents a compact, high-ratio, speed-reduction mechanism that contains four key components:

- Eccentric cam
- Cycloidal disk
- Ring-gear housing
- Pin rollers

The eccentric cam, which extends from the base shaft, sits inside the cycloidal disk. This disk meshes with the ring-gear housing. The pin rollers, which extend from the follower shaft, sit in matching holes on the cycloidal disk.
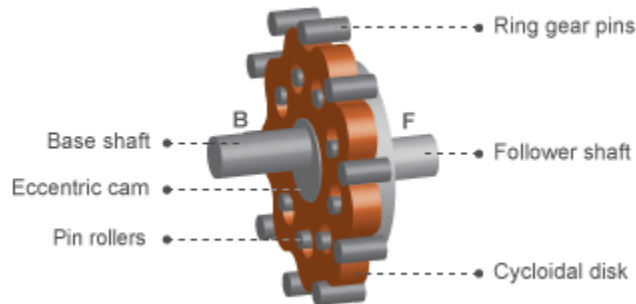
During normal operation, the base shaft drives the eccentric cam. The cam spins inside the cycloidal disk, causing it to rotate in an eccentric pattern about an offset axis. As it moves, the cycloidal disk engages the internal teeth of the ring-gear housing. The internal meshing reverses the rotational velocity direction.

Cycloidal Drive

Pin rollers extending from cycloidal disk holes transmit rotational motion to the follower shaft. This shaft spins counter to the base shaft at a highly reduced speed. The large reduction ratio results from the near-equal cycloidal disk and ring gear tooth numbers. The effective gear reduction ratio is

$$r = \frac{n_R - n_C}{n_C},$$

where:

- r is the gear reduction ratio.
- n_R is the number of teeth on the ring gear.
- n_C is the number of teeth on the cycloidal disk.

The gear reduction ratio constrains the angular velocities of the base and follower shafts according to the expression

$$\omega_F = -r\omega_B,$$

where:

- ω_F is the angular velocity of the follower shaft.
- ω_C is the angular velocity of the base shaft.

The gear reduction ratio also constrains the torques acting on the base and follower shafts, according to the expression

$$T_B = rT_F + T_f,$$

where:

- T_B is the net torque at the base shaft.
- T_F is the net torque at the follower shaft.
- T_f is the torque loss due to friction.

Cycloidal Drive

The magnitude of the frictional torque depends on the torque transfer efficiency and power flow direction according to the expression

$$T_f = \begin{cases} f \cdot abs(T_B) \cdot \left(\dfrac{1}{\eta_{BF}} - 1 \right), & T_B \cdot \omega_B \le 0 \\ f \cdot abs(T_B) \cdot (1 - \eta_{FB}), & T_B \cdot \omega_B > 0 \end{cases},$$

where:

- f is a blending factor that smooths the frictional torque function near zero (e.g., during reversal of the rotational velocity direction).

- η_{BF} is the torque transfer efficiency from the base shaft to the follower shaft.

- η_{FB} is the torque transfer efficiency from the follower shaft to the base shaft.

The blending factor f is a hyperbolic function of the cycloidal disk angular velocity:

$$f = \tanh\left(\dfrac{4\omega_B}{\omega_{Th}} \right),$$

where:

- ω_{Th} is the angular velocity threshold at the base shaft above which full efficiency losses are in effect.

The figure shows the cycloidal drive in front and side views. The kinematics of the drive system cause a reversal of the base and follower shaft angular velocities so that the two shafts spin in opposite directions.

Cycloidal Drive
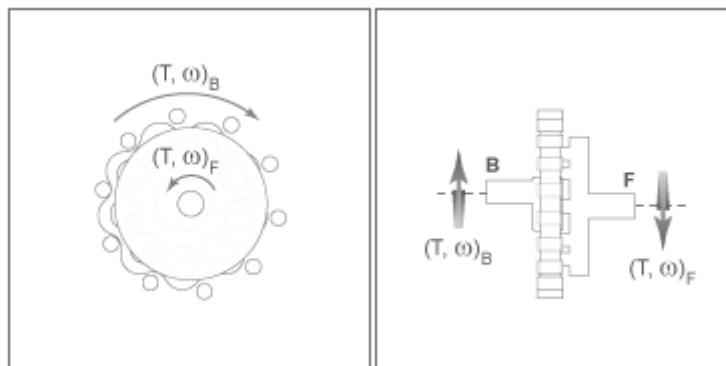
The cycloidal drive can operate in reverse mode, e.g., with power flowing from the follower shaft to the base shaft. In this mode, torque transfer efficiencies are typically negligible. You can adjust the efficiency value in the block dialog box using the **Efficiency from follower shaft to base shaft** parameter.

Dialog Box and Parameters
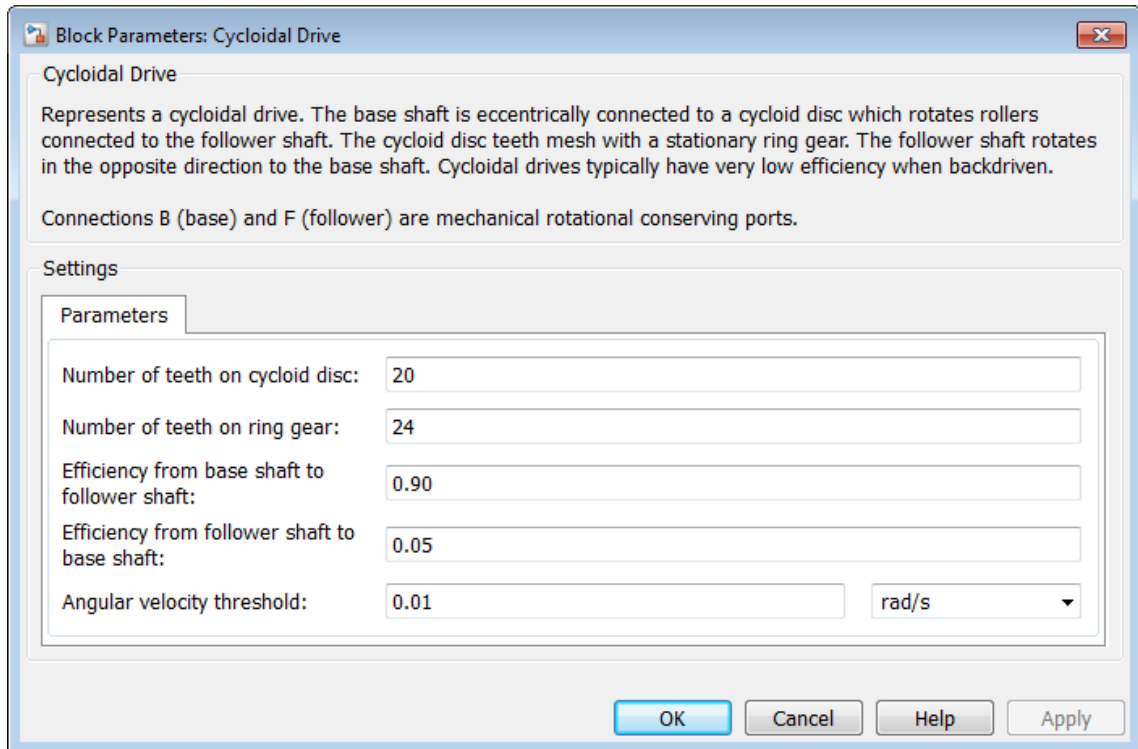
Number of teeth on cycloid disk

Total number of teeth projecting outward from the cycloidal disk perimeter. This number should be slightly smaller than the number of teeth or pins on the ring gear. The ratio of the gear tooth numbers defines the relative angular velocities of the base and follower shafts. The default value is 20.

Cycloidal Drive

Number of teeth on ring gear

Total number of teeth or pins projecting inward from the ring gear housing. This number should be slightly larger than the number of teeth on the cycloidal disk. The ratio of the two gear tooth numbers defines the relative angular velocities of the base and follower shafts. The default value is 24.

Efficiency from base shaft to follower shaft

Torque transfer efficiency in normal operation mode, e.g., with the base shaft driving the follower shaft. Efficiency values must fall in the interval [0, 1]. Larger efficiency values correspond to greater torque transfer between the base and follower shafts. Values approaching unity are typical. The default value is 0.90.

Efficiency from follower shaft to base shaft

Torque transfer efficiency in reverse operation mode, e.g., with the follower shaft driving the base shaft. Efficiency values must fall in the interval [0, 1]. Larger efficiency values correspond to greater torque transfer between base and follower shafts. Values approaching zero are typical. The default value is 0.05.

Angular velocity threshold

Absolute value of the cycloidal disk angular velocity above which the full efficiency factor is in effect. Below this value, a hyperbolic tangent function smooths the efficiency factor to one, lowering the efficiency losses to zero when at rest.

As a guideline, the angular velocity threshold should be lower than the expected angular velocity during simulation. Higher values might cause the block to underestimate efficiency losses. Very low values might, however, raise the computational cost of simulation.

The default value is 0.01 rad/s.

Ports

Port	Description
B	Conserving rotational port representing the base shaft
F	Conserving rotational port representing the follower shaft

See Also   Harmonic Drive | Simple Gear | Simple Gear with Variable Efficiency

Differential

Purpose   Gear train for transferring power to separate shafts spinning at different speeds

Library   Gears
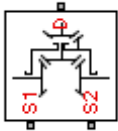
Description   This block represents a gear train for transferring power from one drive shaft to two driven shafts. A combination of simple and bevel gear constraints enable the driven shafts to spin at different speeds, when necessary, and in different directions. One example is an automobile differential, which during a turn enables the inner and outer wheels to spin at different speeds, these depending on the turning radius of each individual wheel.

Any of the shafts can provide the input that drives the remaining two shafts. The differential converts this input into rotation, torque, and power at the driven shafts. The drive gear ratio, which you specify directly in the block dialog box, helps determine the angular velocity of each driven shaft. For more information, see "Differential Gear Model" on page 1-60

This block is a composite component with three underlying blocks:

- Sun-Planet Bevel (two)
- Simple gear

The figure shows the connections between the three blocks.

Differential

Block Parameters: Differential

Differential

Represents a differential which is arranged as a planetary bevel gear train equipped with an additional bevel gear transmission between the driveshaft and the carrier. The pinion gear of this transmission is attached to the driveshaft while the large bevel crown gear is affixed to the carrier. No inertia or compliance is modeled in this block. You can optionally include gear meshing and viscous bearing losses.

Connections D, S1, and S2 are mechanical rotational conserving ports associated with the driveshaft and the two output shafts, respectively. S1 and S2 are the shafts belonging to the two sun gears. When there is no relative slippage across the differential, ports S1 and S2 both rotate in the positive direction when port D rotates in a positive direction if the crown wheel is located to the right of center-line.

Settings

| Main | Meshing Losses | Viscous Losses |

Crown wheel located: To the right of center-line

Carrier (C) to driveshaft (D) teeth ratio (NC/ND): 4

OK   Cancel   Help   Apply

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

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 ω_{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

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*m/(rad/s)).

Differential

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, τ_{S1} and τ_{S2}, are constrained to the longitudinal torque τ_D in such a way that the power flows into and out of the gear, less any power loss P_{loss}, sum to zero:

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

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, $\tau_{loss} \neq 0$. See "Model Gears with Losses".

Limitations
- Gear inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. 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

Ports
- D — Rotational conserving port representing the longitudinal drive shaft.
- S1 — Rotational conserving port representing one of the side shafts.

Differential

- S2 — Rotational conserving port representing one of the side shafts.

See Also   Disk Friction Clutch, Simple Gear, Sun-Planet Bevel

"Represent and Transfer Driveline Motion and Torque"

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 1-68.

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.

Disk Friction Clutch

Dialog Box and Parameters
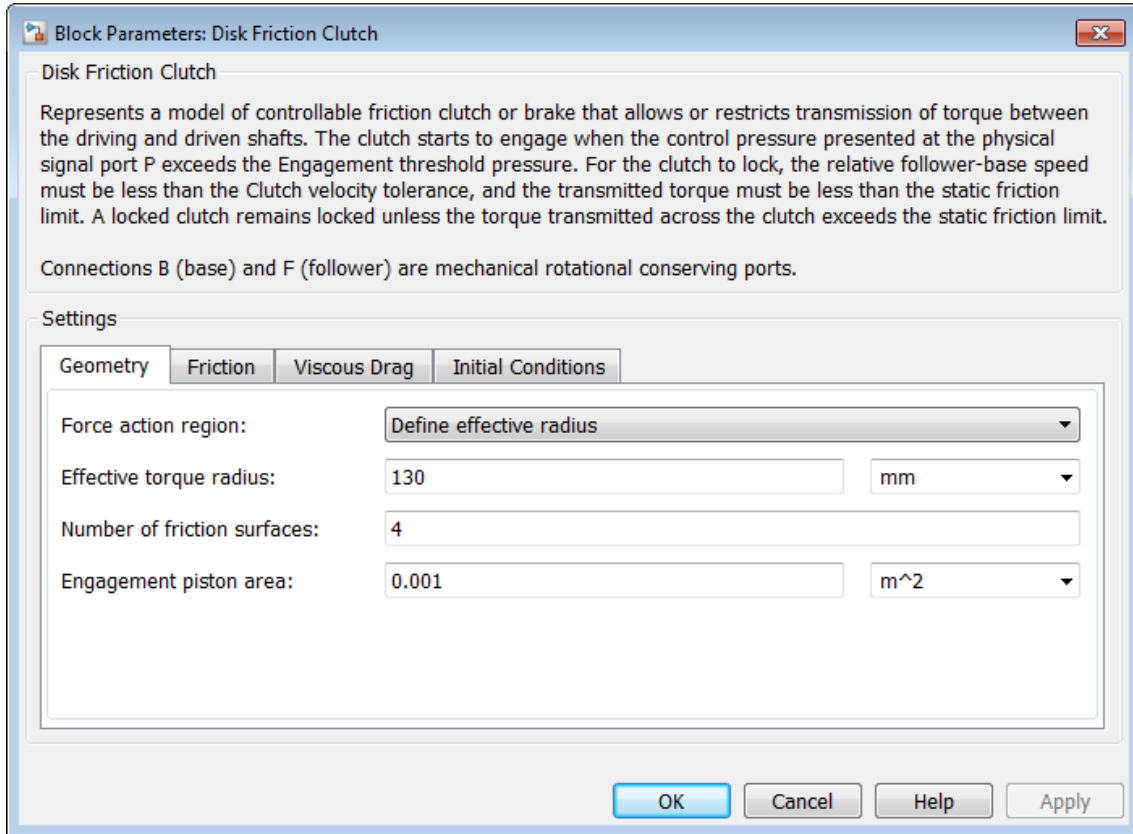
Dimensions

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

    Effective torque radius

    > The effective moment arm radius r_{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

    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.

Disk Friction Clutch

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`).

Friction

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
>
>   **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
>
>   **Kinetic friction coefficient relative velocity vector**
>
>   > Vector of clutch slip values ω 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.

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 ω_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).

Disk Friction Clutch

Viscous Drag

Viscous drag torque coefficient
> Viscous friction coefficient μ applied to the relative slip ω 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)).

Initial Conditions

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 ($P \geq P_{\text{th}}$), 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
ω	Relative angular velocity (slip)	$\omega_{\mathrm{F}} - \omega_{\mathrm{B}}$
ω_{Tol}	Slip tolerance for clutch locking	*See the following model*
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	$\max[(P - P_{\mathrm{th}}), 0]$
D	Clutch de-rating factor	*See the following model*
N	Number of friction surfaces	*See the following model*
A	Engagement surface area	*See the following model*
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 ω.
k_{S}	Static friction coefficient	Dimensionless coefficient of static friction of clutch discs.
μ	Viscous drag coefficient	*See the following model*
τ_{K}	Kinetic friction torque	*See the following model*
τ_{S}	Static friction torque limit	(static friction peak factor)·(kinetic friction torque for $\omega \to 0$) (*See the following model*)

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:

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

(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:

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

You specify the *kinetic friction coefficient* k_K as either a constant or a tabulated discrete function of relative angular velocity ω. 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 ω only.

The clutch applies a normal force from its piston as the product of the clutch friction capacity P_{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_{th}, the clutch applies no friction at all.

The *effective torque radius* r_{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:

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

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:

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

Static Friction

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

$$\tau_\mathrm{S} = k_\mathrm{S} \cdot D \cdot N \cdot r_\mathrm{eff} \cdot P_\mathrm{fric} \cdot A \ge 0 \ .$$

$k_\mathrm{S} > k_\mathrm{K}$, so that the torque τ 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:

$$\tau_\mathrm{S} \equiv \tau_\mathrm{S}{}^+ = -\tau_\mathrm{S}{}^- \ .$$

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 ($\tau_\mathrm{K} > 0$) by the positive clutch friction capacity condition ($P \ge P_\mathrm{th}$).

Power Dissipated by the Clutch

The power dissipated by the clutch is $|\omega \, \tau_\mathrm{K}|$. The clutch dissipates power only if it is both slipping ($\omega \ne 0$) and applying kinetic friction ($\tau_\mathrm{K} > 0$).

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

Related Examples   • "Gear Coupling Control with Clutches"

Purpose   Nonslip clutch based on positive engagement between interlocking teeth

Library   Clutches

Description   This block represents a nonslip clutch, a mechanical device that relies on the positive engagement of interlocking teeth to transfer torque between driveline shafts. The clutch contains three key components:
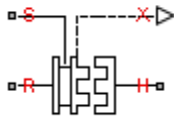
- Ring
- Hub
- Shift linkage

The ring and the hub are toothed components. The ring spins with the output shaft, sliding along its longitudinal axis to engage or disengage the coaxial hub. The hub, which sits on a bearing encircling the same shaft, can spin independently until engaged.

Shift Linkage (S)

Ring (R)   Hub (H)
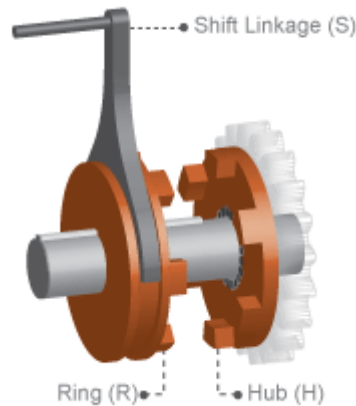
Engagement occurs when the toothed components interlock. Once engaged, the ring and the hub spin together as a unit. To control engagement, the dog clutch contains a shift linkage that governs the position of the ring with respect to the hub.

Dog Clutch

Moving the ring towards the hub so that their teeth interlock changes the clutch state to engaged. Tooth overlap must exceed a minimum value for engagement. Moving the ring in reverse so that the two sets of teeth no longer interlock changes the clutch state back to disengaged.

A physical signal, which you connect to port S, specifies the shift linkage position. When fully disengaged, the shift linkage position is zero. When fully engaged, the shift linkage position equals the sum of the tooth height and the ring-hub clearance when disengaged:

$$z = h + z_{Gap},$$

where:

- z is the shift linkage position.

- h is the tooth height.

- z_{Gap} is the ring-hub clearance when disengaged.

The figure shows side and front views of the dog clutch and some of its relevant variables.

T — Shift linkage translation   R — Mean clutch teeth radius
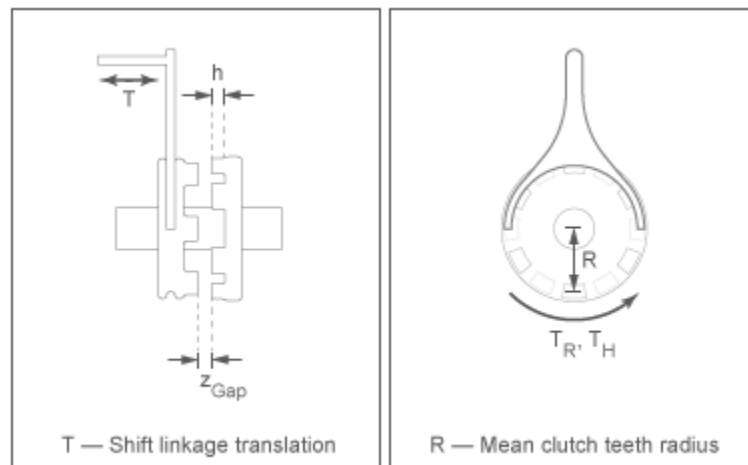
Torque Transmission Models

The Dog Clutch block provides a choice of two torque transmission models.

Friction Clutch Approximate Model

Treat clutch engagement as a friction phenomenon between the ring and the hub. This model ignores special effects such as backlash, an approximation that makes the block better suited for linearization, fixed-step simulation, and hardware-in-loop (HIL) simulation. The Fundamental Friction Clutch block provides the foundation for the model.

In the friction approximate model, the clutch has three possible configurations: disengaged, engaged, and locked. When disengaged, the contact force between the ring and the hub is zero. This force remains zero until the shift linkage reaches the minimum position for engagement.

When the ring-hub tooth overlap (h) exceeds the minimum value for engagement, the contact force between the two components begins to increase linearly with the shift linkage position (z).

At full engagement, the contact force reaches its maximum value, and the clutch state switched to locked. In this state, the ring and the hub spin as a unit without slip. To unlock the clutch, the transmitted torque must exceed the maximum allowed value that you specify.

Dynamical Model with Backlash

Capture clutch phenomena such as backlash, torsional compliance, and contact forces between ring and hub teeth. This model provides greater accuracy than the friction clutch approximation.

In the dynamical model, the clutch has two possible configurations: disengaged and engaged. When disengaged, the contact force between the ring and the hub is zero. This force remains zero until the shift linkage reaches the minimum position for engagement.

When the ring-hub tooth overlap (h) exceeds the minimum value for engagement, a contact force kicks in between the two components. This

Dog Clutch

force is the sum of torsional spring and damper components. Including backlash between the ring and hub teeth:

$$T_C = \begin{cases} -k_{RH}\left(\phi - \dfrac{\delta}{2}\right) - \mu_R \cdot \omega & \phi > \dfrac{\delta}{2} \\ \\ 0 & -\dfrac{\delta}{2} < \phi < \dfrac{\delta}{2}, \\ \\ -k_{RH}\left(\phi + \dfrac{\delta}{2}\right) - \mu_R \omega & \phi < -\dfrac{\delta}{2} \end{cases}$$

where:

- k_{RH} is the torsional stiffness of the ring-hub coupling.

- φ is the relative angle, about the common rotation axis, between the ring and the hub.

- δ is the backlash between ring and hub teeth.

- ω is the relative angular velocity between the ring and the hub. This variable describes how fast the two components slip past each other.

Compliant end stops limit the translational motion of the clutch shift linkage and the ring. The compliance model treats the end stops as linear spring-damper sets. The location of the end stops depends on the relative angle and angular velocity between the ring and hub teeth:

- If the teeth align and the relative angular velocity is smaller than the maximum value for clutch engagement, the end-stop location is the sum of the ring-hub clearance when fully disengaged and the tooth height. With the end stop at this location, the clutch can engage.

- If the teeth do not align or the relative angular velocity exceeds the maximum value for clutch engagement, the end-stop location is set to prevent the ring from engaging the hub. The clutch remains disengaged.

Translational friction opposes shift linkage and ring motion. This friction is the sum of Coulomb and viscous components:

$$F_Z = -k_K \cdot F_N \cdot \tanh\left(\frac{4v}{v_{th}}\right) - \mu_T v,$$

where:

- F_Z is the net translational friction force acting on the shift linkage and ring.

- k_K is the kinetic friction coefficient between ring and hub teeth.

- F_N is the normal force between ring and hub teeth.

- v is the translational velocity of the shift linkage and the ring.

- v_{th} is the translational velocity threshold. Below this threshold, a hyperbolic tangent function smooths the Coulomb friction force to zero as the shift linkage and ring velocity tends to zero.

- μ_T is the viscous damping coefficient acting on the shift linkage and the ring.

Clutch Engagement Conditions

The clutch engages when it satisfies a set of geometrical and dynamical conditions. These conditions specify the values that certain variables can take for clutch engagement to occur:

- The minimum position at which the ring and the hub can engage is

  $$z = h_0 + z_{Gap},$$

  where h_0 is the minimum tooth overlap for clutch engagement. Adjust this parameter to minimize engagement instability, i.e., the tendency of the clutch to switch rapidly between engaged and disengaged states

- The magnitude of the relative angular velocity between the ring and the hub must be smaller than the maximum engagement velocity:

$$|\omega| < |\omega_{\max}|,$$

where $\omega_{\max}$ is the maximum value of the relative angular velocity at which engagement can occur.

- If using the friction clutch approximate model, engagement occurs only if torque transfer between the ring and the hub remains smaller than the maximum transmitted torque that the clutch supports.

- If using the dynamical model with backlash, engagement occurs only if the relative angular position of the ring and hub teeth allows them to interlock.

Rotational Power Dissipation

When the clutch slips under an applied torque, it dissipates power. The power loss equals the product of the slip angular velocity and the contact torque between the ring and the hub:

$$P_{loss} = \omega \cdot T_C,$$

where:

- P_{loss} is the dissipated power due to slipping.

- T_C is the kinetic contact torque.

Dialog Box and Parameters

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.

Settings

Clutch | Shift Linkage | Engagement Conditions | Initial Conditions

Torque transmission model:	Friction clutch approximation - Suitable for HIL and linearization	▼
Maximum transmitted torque:	1000	N*m ▼
Clutch teeth mean radius:	50	mm ▼

OK   Cancel   Help   Apply

Dog Clutch

Clutch

Torque transmission model

> Computational framework with which to model the dynamic behavior of the dog clutch. The default setting is `Friction clutch approximation   Suitable for HIL and linearization`.

- `Friction clutch approximation - Suitable for HIL and linearization` — Model clutch engagement as a friction phenomenon between the ring and the hub. This model, based on the Fundamental Friction Clutch block, provides a computationally efficient approximation of the dog clutch.

  #### Friction Clutch Approximation

  Maximum transmitted torque

  > Largest torque that the clutch can transmit, corresponding to a nonslip engaged configuration. If the torque transmitted between the ring and the hub exceeds this value, the two components begin to slip with respect to each other. This torque determines the static friction limit in the friction clutch approximation. The default value is `1000`.

  Clutch teeth mean radius

  > Distance from the ring or hub center to the corresponding tooth center. The mean tooth radius determines the normal contact forces between ring and hub teeth given the transmission torque between the two components. The mean radius must be greater than zero. The default value is `50`.

- `Dynamic with backlash` — Model clutch engagement in detail, accounting for such phenomena as backlash, torsional compliance, and contact forces between ring and hub teeth. Selecting this option causes additional parameters to appear in the **Shift Linkage** and **Initial Conditions** tabs.

Dynamic with Backlash

Number of teeth

Total number of teeth in the ring or the hub. The two components have equal tooth numbers. The number of teeth must be greater than or equal to one. The default value is 6.

Rotational backlash

Allowable angular motion, or play, between the ring and hub teeth in the engaged clutch configuration. The backlash must be greater than zero. The default value is 10.

Clutch teeth mean radius

Distance from the ring or hub center to the corresponding tooth center. The mean tooth radius determines the normal contact forces between ring and hub teeth given the transmission torque between the two components. The mean radius must be greater than zero. The default value is 50.

Torsional stiffness

Linear torsional stiffness coefficient at the contact interface between the ring and hub teeth. This coefficient characterizes the restoring component of the contact force between the two sets of teeth. Greater stiffness values correspond to greater contact forces. The stiffness coefficient must be greater than zero. The default value is 1e+7 N*m/rad.

Torsional damping

Linear torsional damping coefficient at the contact interface between the ring and hub teeth. This coefficient characterizes the dissipative component of the contact force between the two sets of teeth. Greater damping

values correspond to greater energy dissipation during contact. The damping coefficient must be greater than zero. The default value is `100`.

Shift Linkage

Tooth height
> Distance between the base and crest of a tooth. Ring and hub teeth share the same height. The tooth height and the ring-hub clearance when fully disengaged determine the maximum travel span of the shift linkage. The tooth height must be greater than zero. The default value is `10` mm.

Hard stop at back of shift linkage
> Choice of translational motion limiting device. Select whether to stop the shift linkage when fully disengaged. The default value is `Hard stop when fully disengaged`.

Ring-hub clearance when disengaged
> Maximum open gap between the ring and hub tooth crests along the shift linkage translation axis. This gap corresponds to the fully disengaged clutch state. The tooth height and the ring-hub clearance when fully disengaged determine the maximum travel span of the shift linkage. The ring-hub clearance must be greater than zero. The default value is `3` mm.

Ring stop stiffness
> Linear stiffness coefficient of the ring end stop. This coefficient characterizes the restoring component of the contact force that resists translational motion past the end stops. Greater stiffness values correspond to greater contact forces and a smaller end stop compliance. The stiffness coefficient must be greater than zero. The default value is `1e+6` N/m.

Ring stop damping
> Linear damping coefficient of the ring end stop. This coefficient characterizes the dissipative component of the contact force that resists translational motion past the end stops. Greater damping values correspond to greater energy dissipation during contact.

The damping coefficient must be greater than or equal to zero. The default value is 1000 N/(m/s).

Shift linkage viscous friction coefficient

Linear damping coefficient acting on the shift linkage. This coefficient characterizes the dissipative force that resists shift linkage motion due to viscous damping. Greater coefficient values correspond to greater energy dissipation during shift linkage motion. The viscous friction coefficient must be greater than zero. The default value is 100 N/(m/s).

Tooth-tooth friction coefficient

Kinetic friction coefficient at the contact interface between ring and hub teeth. This coefficient characterizes the dissipative force that resists shift linkage motion due to tooth-tooth contact during clutch engagement/disengagement.

Greater coefficient values correspond to greater energy dissipation during shift linkage motion. The friction coefficient must be greater than zero. The default value is 0.05. This parameter appears only when you set **Torque transmission model** in the **Clutch** tab to Dynamic with backlash.

Engagement Conditions

Linkage travel direction

Direction the shift linkage must travel in to engage the clutch. Choices include positive and negative displacements. The default setting is Positive shift linkage displacement engages clutch.

Maximum engagement velocity

Relative angular velocity between the ring and the hub above which the clutch cannot engage. The maximum engagement velocity must be greater than zero. The default value is inf (infinity) rad/s.

Dog Clutch

Tooth overlap to engage

Overlap length between ring and hub teeth along the common longitudinal axis above which the clutch can engage. The clutch remains disengaged until the tooth overlap by at least this length. The tooth overlap to engage must be greater than zero. The default value is `3` mm.

Initial Conditions

Clutch Initial state

Clutch configuration at simulation time zero. Choices include:

- `Disengaged` — Clutch transmits zero torque between the ring and the hub.

- `Engaged` — Clutch transmits torque between the ring and the hub.

The default value is `Disengaged`.

Initial shift linkage position

Shift linkage position at simulation time zero. Values between zero and the sum of the ring-hub clearance and the tooth overlap to engage are consistent with a disengaged clutch. Larger values are consistent with an engaged clutch. The default value is `0` mm.

Initial ring-hub offset angle

Rotation angle between the ring and the hub at simulation time zero. This angle determines whether the ring and hub teeth can interlock, and hence whether the clutch can engage. The initial offset angle must satisfy these conditions:

- If the clutch initial state is disengaged, the initial offset angle must fall in the range

$$-\frac{180^\circ}{N} \leq \phi_0 \leq +\frac{180^\circ}{N},$$

where N is the number of teeth present in the ring or the hub. The two components contain the same number of teeth.

- If the clutch initial state is engaged, the initial offset angle must fall in the range

$$-\frac{\delta}{2} \le \phi_0 \le +\frac{\delta}{2},$$

where δ is the backlash angle between the ring and hub teeth.

The default value is 0 deg. This parameter appears only when you set **Torque transmission model** in the **Clutch** tab to Dynamic with backlash.

Ports   The table summarizes the ports that the block contains.

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

See Also   Fundamental Friction Clutch | Loaded-Contact Rotational Friction | Torsional Spring-Damper

Related Examples   • "Gear Coupling Control with Clutches"

Double-Pinion Planetary Gear

Purpose   Planetary gear train with two meshed planet gear sets

Library   Gears
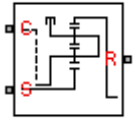
Description   This block represents a planetary gear train with two meshed planet gear sets between the sun gear and the ring gear. A single carrier holds the two planet gear sets at different radii from the sun gear centerline, while allowing the individual gears to rotate with respect to each other. The gear model includes power losses due to friction between meshing gear teeth and viscous damping of the spinning gear shafts.
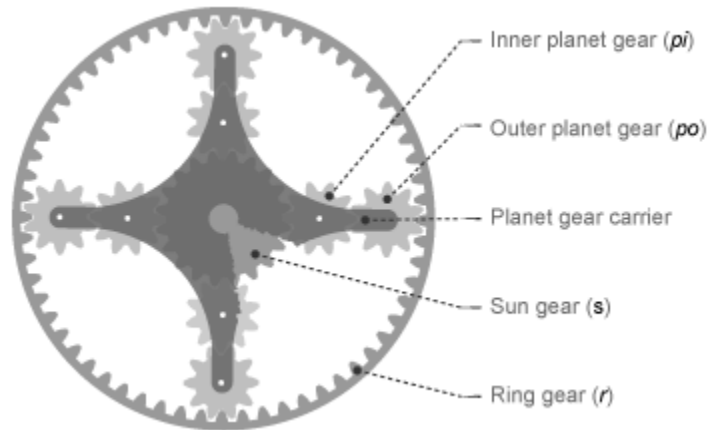
Structurally, the double-pinion planetary gear resembles a Ravigneaux gear without its second, large, sun gear. The inner planet gears mesh with the sun gear and the outer planet gears mesh with the ring gear. Because it contains two planet gear sets, the double-pinion planetary gear reverses the relative 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:

$$r_r = r_s + 2 \cdot r_{pi} + 2 \cdot r_{po},$$

where:

- r_r is the ring gear radius
- r_s is the sun gear radius
- r_{pi} is the inner planet gear radius
- r_{po} is the outer planet gear radius
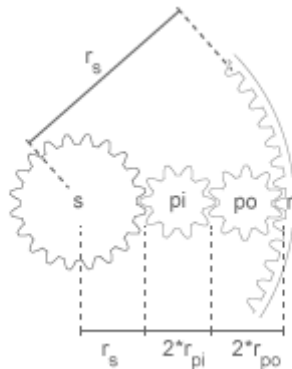
In terms of the ring-sun and outer planet-inner planet teeth ratios, the ring-outer planet teeth ratio is

$$\frac{r_r}{r_{po}} = 2 \cdot \frac{\dfrac{r_r}{r_s}}{\left(\dfrac{r_r}{r_s} - 1\right)} \cdot \frac{\left(\dfrac{r_{po}}{r_{pi}} + 1\right)}{\dfrac{r_{po}}{r_{pi}}},$$

The inner planet-sun teeth ratio is

Double-Pinion Planetary Gear

$$\frac{r_{pi}}{r_s} = \frac{\left(\dfrac{r_r}{r_s} - 1\right)}{2\left(\dfrac{r_{po}}{r_{pi}} + 1\right)},$$

The Differential block is a composite component. It contains three underlying blocks—Ring-Planet, Planet-Planet, and Sun-Planet—connected as shown in the figure. Each block connects to a separate drive shaft through a rotational conserving port.
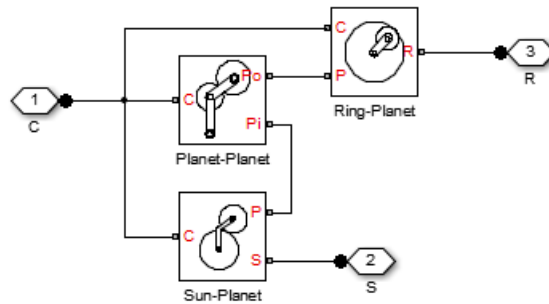
**Dialog
Box and
Parameters**
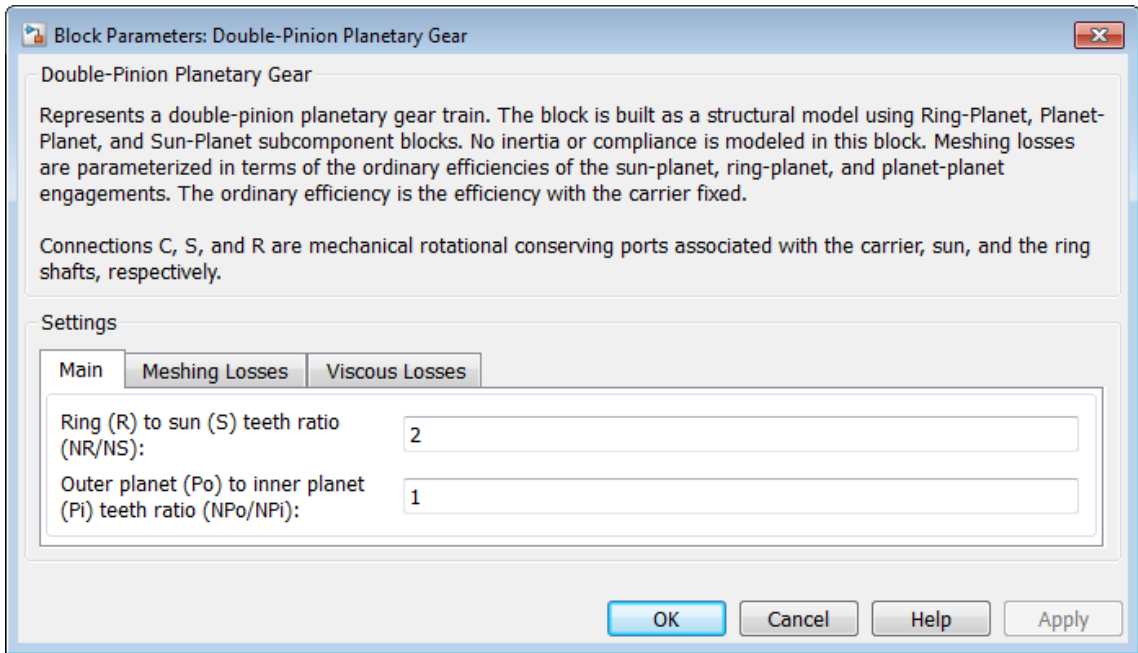
Main

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.

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).

Ports

Port	Description
C	Conserving rotational port that represents the planet gear carrier
S	Conserving rotational port that represents the sun gear
R	Conserving rotational port that represents the ring gear

See Also   Planetary Gear | Compound Planetary Gear | Ravigneaux Gear

Double-Shoe Brake

Purpose   Frictional brake with two pivoted shoes diametrically positioned about rotating drum

Library   Brakes & Detents/Rotational
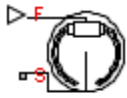
Description   The block represents a frictional brake with two pivoted rigid shoes that press against a rotating drum to produce a braking action. The rigid shoes sit inside or outside the rotating drum in a diametrically opposed configuration. A positive actuating force causes the rigid shoes to press 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.

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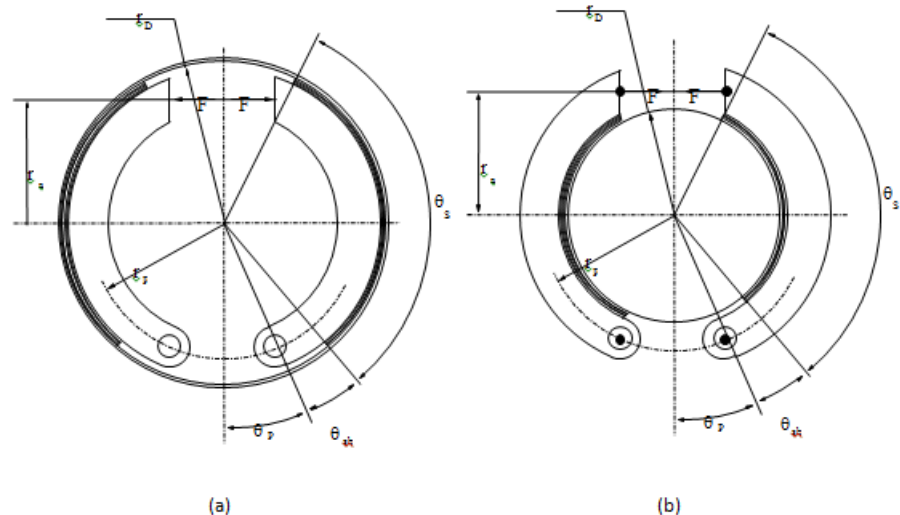
In this schematic, a) represents an internal double-shoe brake, and b) represents an external double-shoe brake. In both configurations, a

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° produce less accurate results. The following formulas provide the friction torque the leading and trailing shoes develop, respectively:

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

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

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
μ	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
θ_{sb}	Shoe beginning angle

Double-Shoe Brake

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

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:

$$\mu = \mu_{Coulomb} \cdot \tanh\left(\frac{4\omega_{shaft}}{\omega_{threshold}} \right)$$

In the formula, the parameters have the following meaning:

Parameter	Description
μ	Effective contact friction coefficient
$\mu_{Coulomb}$	Contact friction coefficient
ω_{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.

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

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

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

The parameters have the following meaning:

Parameter	Description
p_a	Maximum linear pressure at the shoe-drum contact surface
r_p	Pin location radius

Parameter	Description
θ_p	Hinge pin location angle
r_a	Actuator location radius

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 μ_{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

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.

Settings

| Geometry | Friction |

Drum radius:	150	mm ▼
Actuator location radius:	100	mm ▼
Pin location radius:	125	mm ▼
Pin location angle:	15	deg ▼
Shoe beginning angle:	5	deg ▼
Shoe span angle:	120	deg ▼

OK   Cancel   Help   Apply

Geometry

Drum radius

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

Double-Shoe Brake

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 $0 \leq \theta_{sb} \leq$ (π-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 $0 < \theta_{sb} \leq$ (π -pin location angle - shoe beginning angle). The default value is `130 deg`.

Friction

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 n*m/(rad/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 rad/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   Back-to-back dog-cone clutch pairs assembled symmetrically about a translational detent to provide smooth gear engagement

Library   Clutches

Description
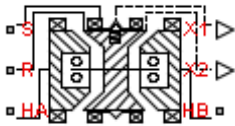
The block represents a double-sided synchronizer that contains two back-to-back dog clutches, two back-to-back cone clutches, and one translational detent. Shift linkage translation along the negative direction causes the clutches to engage the ring with hub A. Shift linkage translation along the positive 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.
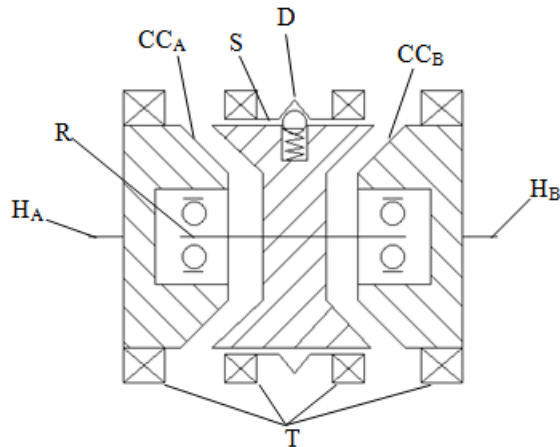
The schematic illustrates a double-sided synchronizer in the disengaged state. In this state, the ring (R) and hub (H_A and H_B) shafts can spin independently at different speeds. As the shift linkage (S) translates in the negative direction, the faces of cone clutch A (CC_A) come into contact. The friction 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).
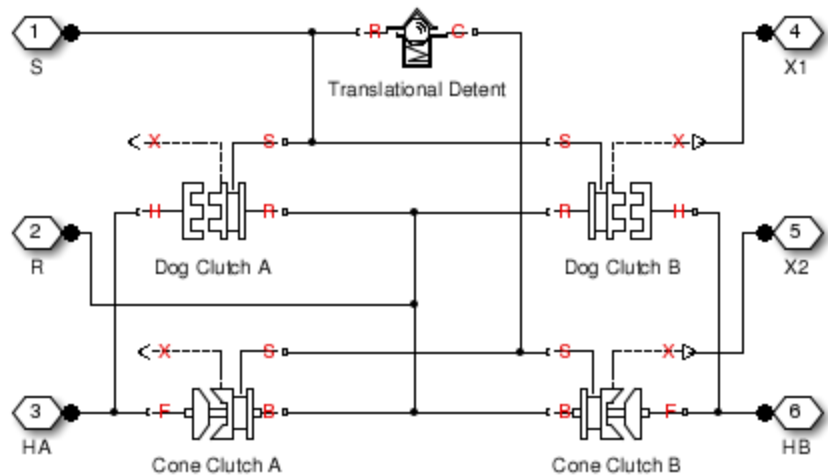
The model implements two Dog Clutch blocks, one Translational Detent, and two modified Dog Clutch blocks. Refer to each block reference page for more information on the corresponding block function.

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.

Dog Clutch State	X1
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

Cone Clutch State	X2
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.

Dialog Box and Parameters

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.

Settings

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

Torque transmission model:	Friction clutch approximation - Suitable for HIL and linearization ▼	
Maximum transmitted torque:	1000	N*m ▼
Clutch teeth mean radius:	200	mm ▼
Maximum engagement speed:	Inf	rad/s ▼

OK   Cancel   Help   Apply

Double-Sided Synchronizer

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`.

Cone Clutch

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`.

Friction model

Specification method for the kinetic friction coefficient between cone clutch disks in slipping conditions. Options include:

- `Fixed kinetic friction coefficient` — Specify a single value for the kinetic friction coefficient. This value is constant across all clutch relative velocities. Selecting this option exposes an additional parameter, **Kinetic friction coefficient**.

- `Table lookup kinetic friction coefficient` — Specify a lookup table with the kinetic friction coefficient values as a function of the clutch slip velocity. Selecting this option exposes additional parameters for the lookup table values and interpolation/extrapolation methods.

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`.

Kinetic friction coefficient relative velocity vector

Vector with the clutch slip velocities at which to define the kinetic friction coefficient.

Kinetic friction coefficient vector

Vector with the kinetic friction coefficient values corresponding to the clutch slip velocities in **Kinetic friction coefficient relative velocity vector**.

Friction coefficient interpolation method

Method for calculating kinetic friction coefficients between lookup-table data points. Options include `Linear`, `Cubic`, and `Spline`.

Friction coefficient extrapolation method

Method for calculating kinetic friction coefficient values outside of the lookup-table data range. Options include `From last 2 points` and `From last point`.

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

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`.

Double-Sided Synchronizer

Shift Linkages

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)`.

Initial Conditions

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 **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**
Dog clutch A Initially disengaged	Negative of the parameter value must be smaller than the sum of parameters **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**
Dog clutch B Initially engaged	Parameter value must be greater than the sum of parameters **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**
Dog clutch B Initially disengaged	Parameter value must be smaller than the sum of parameters **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**

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:

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

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 δ 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

Double-Sided Synchronizer

Port	Description
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

See Also   Synchronizer | Cone Clutch | Dog Clutch | Translational Detent

Purpose　　　　Torsionally-compliant driveshaft

Library　　　　Couplings & Drives

Description
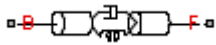
The Flexible Shaft block models the torsional dynamics of a flexible shaft. The shaft is represented by an approximate lumped parameter circuit consisting of *N* segments. *N* is a finite integer greater than zero.
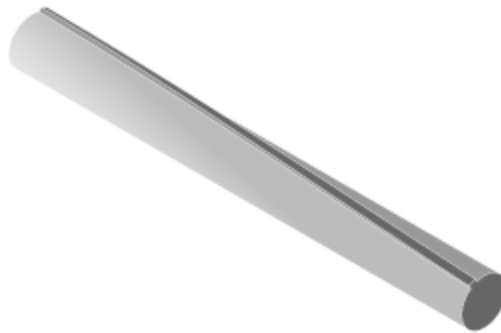
You can parameterize the block in either the overall stiffness and inertia or the dimensions and material properties of the shaft. You can also apply optional viscous friction at the base and follower ports to represent bearing losses at the shaft ends.

Ports and Conventions

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

Flexible Shaft
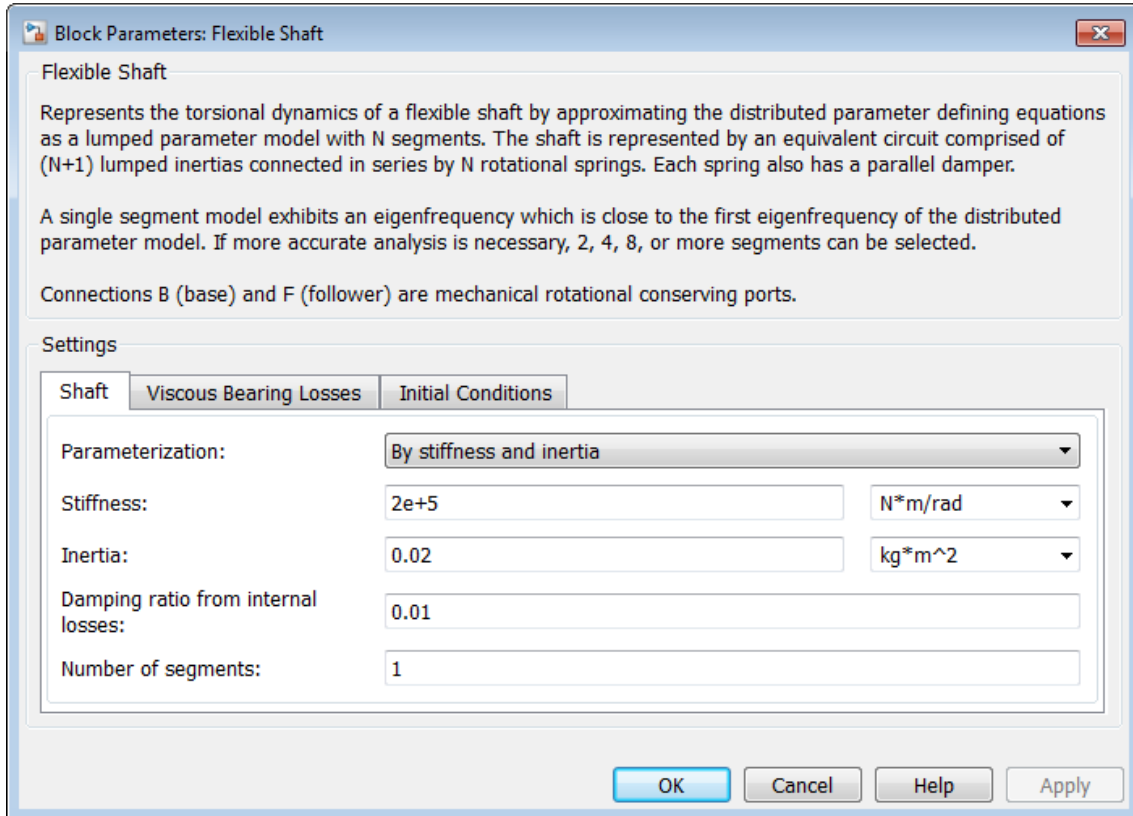
Shaft

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^2 (`kg-m^2`).

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

automate

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`).

Flexible Shaft

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

Annular

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`).

Material density

> Mass density ρ 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^3 (`kg/m^3`).

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 *N* of rigid segments into which the shaft is divided. The
default is 1.

Viscous Bearing Losses

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

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 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.
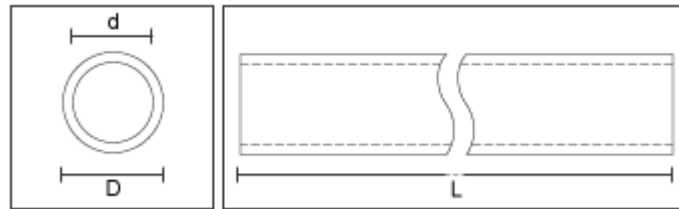
Shaft Characterized by Dimensions and Material Properties

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
ρ	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 $2c/\omega_N$. ω_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.

Flexible Shaft
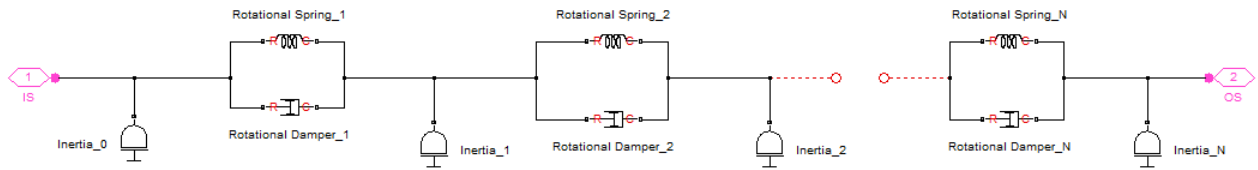
Limitations

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 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
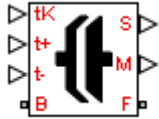
Description A friction clutch transfers motion and torque between two driveline axes by coupling them with friction. The Fundamental Friction Clutch block models a standard friction clutch with kinetic friction and static (locking) friction acting on the two axes. For model details, see "Fundamental Friction Clutch Model" on page 1-123.

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 τ_S^+ (port t+)
- Static friction lower limit torque τ_S^- (port t–)

The clutch generates two physical signal outputs:

- Clutch slip ω (port S)
- Clutch state or mode (port M)

Fundamental Friction Clutch

**Dialog
Box and
Parameters**
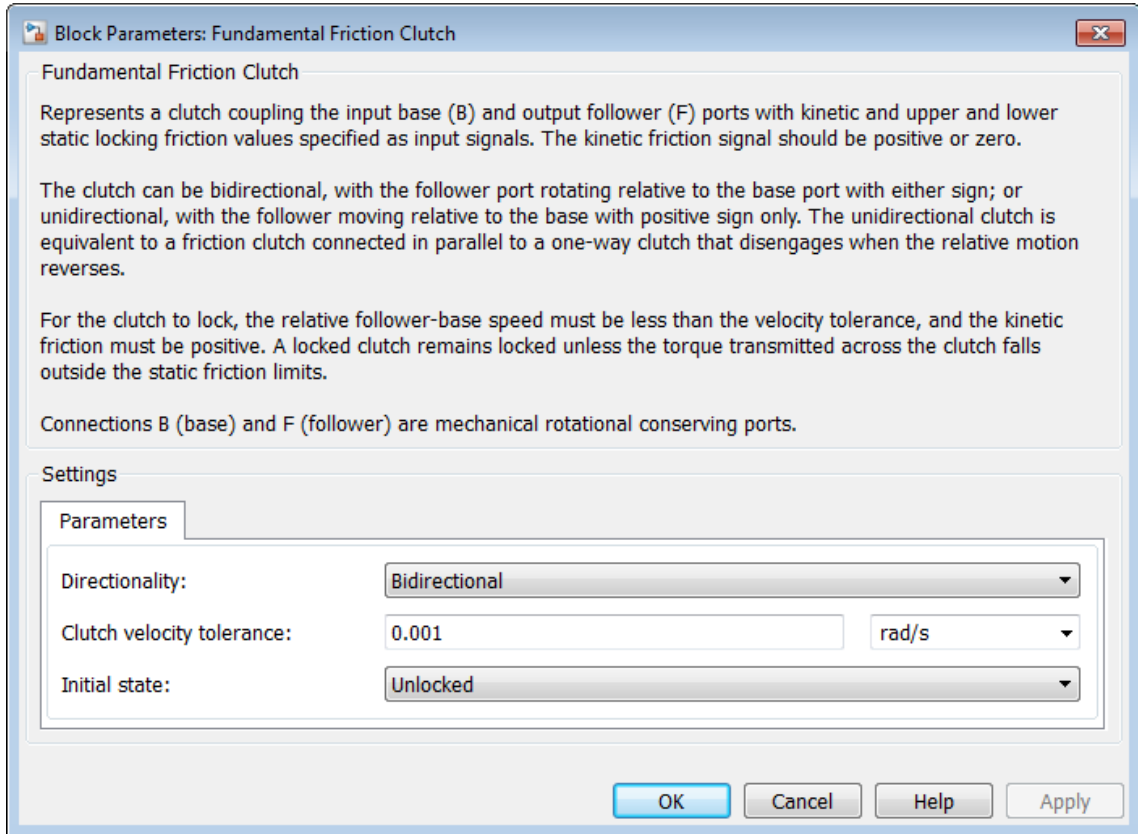
Directionality

>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 ω_{Tol} above which the clutch cannot lock. Below this speed, the clutch can lock. See the diagram in "Clutch State, Transition, and Variable Summary" on page 1-125. 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 ω_{Tol} for each clutch individually.

Fundamental Friction Clutch

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 following figure shows 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.
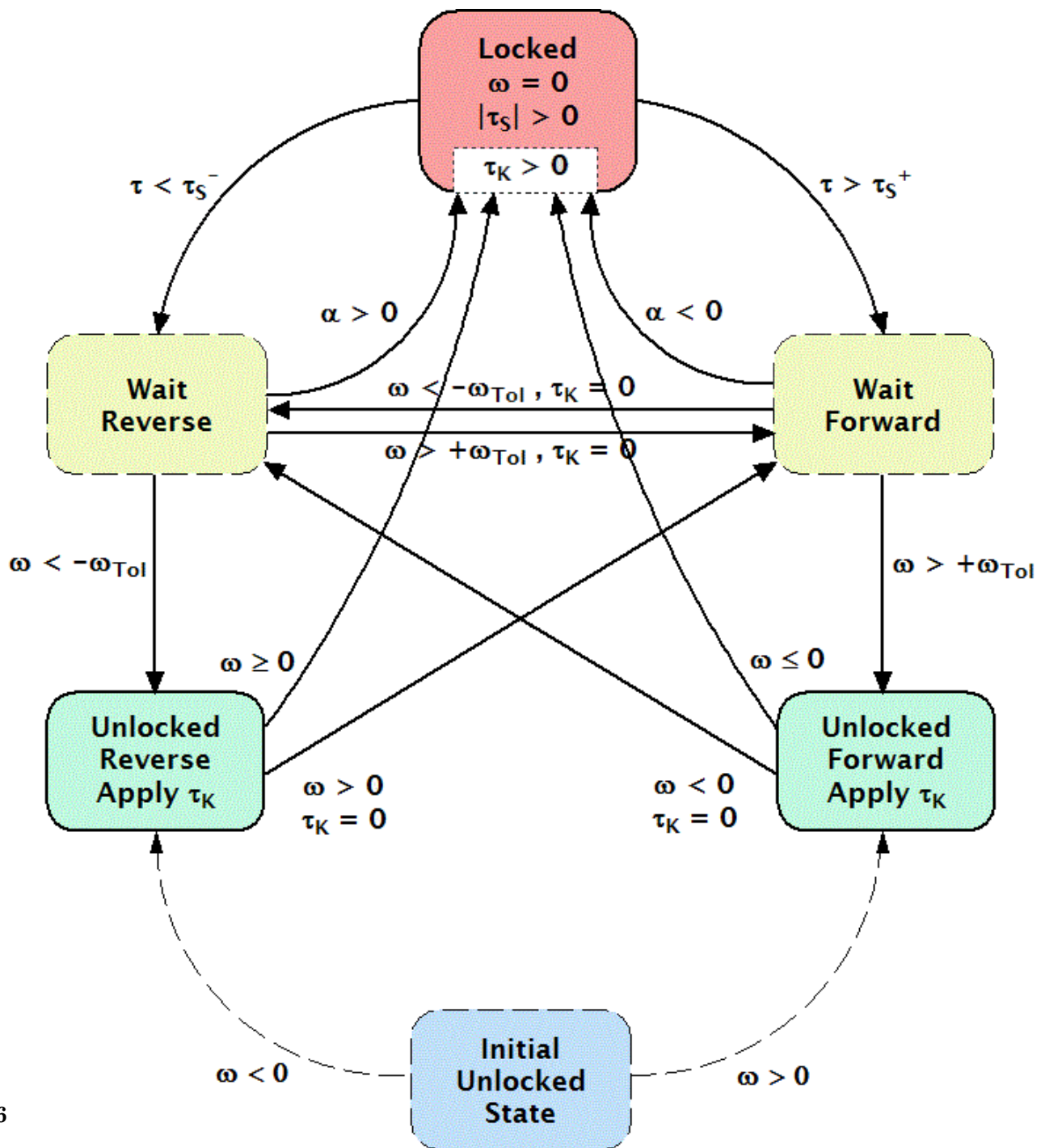
Fundamental Friction Clutch

The following diagram summarizes the physical differences between the locked and unlocked states.
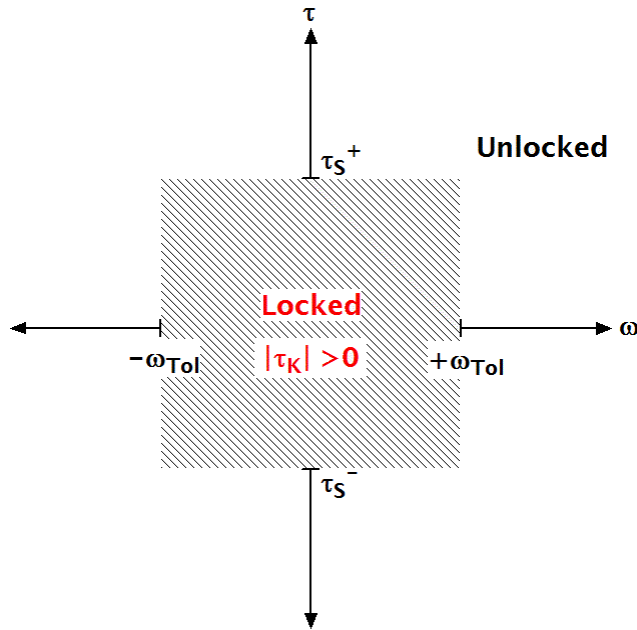
The following tables summarize the clutch variables, states, and modes.

Clutch Variables

Symbol	Definition	Input Signal	Significance
ω	Relative angular velocity (slip)		$\omega_F - \omega_B$
α	Relative angular acceleration		$d\omega/dt$
ω_{Tol}	Slip tolerance for clutch locking		First locking condition: $|\omega| \leq \omega_{Tol}$

Clutch Variables (Continued)

Symbol	Definition	Input Signal	Significance
τ_K	Kinetic friction torque	tK	Second locking condition: $\tau_K > 0$
$\tau_S^{\pm}$	Static friction torque limits	t±	Defines locked range
τ	Total torque transferred across clutch		Clutch remains locked if $\tau_S^- < \tau < \tau_S^+$.

Clutch States and Modes

State	Mode Signal
Forward *or* Wait Forward	+1
Locked	0
Reverse *or* 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 in "Clutch State, Transition, and Variable Summary" on page 1-125.

Clutch Locking and Unlocking

Locking requires that the:

- Relative speed (absolute slip) $|\omega|$ be smaller than a velocity threshold ω_{Tol}.

- Kinetic friction torque τ_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.

Fundamental Friction Clutch

Unlocked State: Kinetic Friction

The kinetic friction torque τ_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.

Locked State: Static Friction

Once the friction clutch locks, it remains locked as long as the total torque τ 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. τ_S^+ and τ_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:

  - $|\omega| \leq \omega_{Tol}$

  - τ changes sign while the clutch is unlocked

If the ω 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 τ_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 α is negative.

- The whole driveline requires the axes to turn in the relative reverse direction, but α 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 ω_{Tol}. If the result is less than ω_{Tol}, the clutch returns to the start of the Wait state and the relative acceleration check. If the result exceeds ω_{Tol}, the clutch unlocks.

3 In the Unlocked state, the clutch begins applying kinetic friction
again.

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

See Also   Differential | Disk Friction Clutch | Torque Converter | Variable
Ratio Transmission

**Related
Examples**
- "Gear Coupling Control with Clutches"

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

Library   Engines

Description
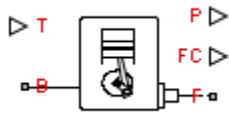
The block represents a general internal combustion engine. Engine types include spark-ignition and diesel. Speed-power and speed-torque parameterizations are provided. A throttle physical signal input specifies the normalized engine torque. Optional dynamic parameters include crankshaft inertia and response 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 1-144.

Generic Engine

Dialog Box and Parameters
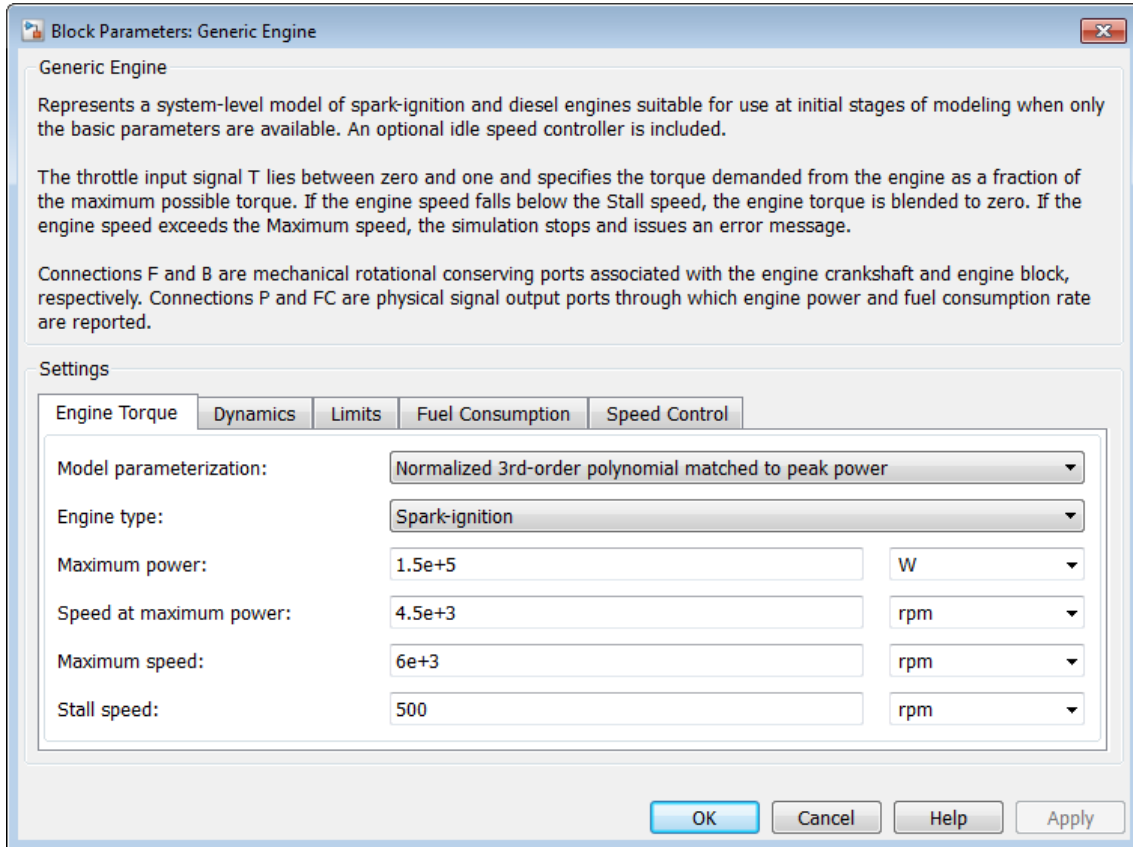
Engine Torque

Model parameterization
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 Ω_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 Ω_{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 Ω exceeds this maximum, the simulation stops with an error.

Note The engine maximum speed Ω_{max} cannot exceed the engine speed at which the engine power becomes negative. For more information, see "Generic Engine Model" on page 1-144.

Stall speed

Minimum speed Ω_{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 Ω 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

Speed vector

Vector of values of the engine function's independent variable, the speed Ω.

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 τ.

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

Speed vector

Vector of values of the engine function's independent variable, the speed Ω.

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 (kg*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`.

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 Ω approaches the stall speed. The default is `100`.

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

Generic Engine

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.

Fuel consumption by speed and torque

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.

Brake specific fuel consumption by speed and torque

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.

Brake specific fuel consumption by speed and brake mean effective pressure

Displaced volume
Enter the volume displaced by a piston stroke. The default value is 400. The default unit is cm^3.

Generic Engine

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:

$$BMEP = T \left[\frac{2\pi \; n_c}{V_d} \right]$$

where:

- T — Output torque

- n_c — Number of cycles per revolution

- V_d — Cylinder displaced volume

Brake specific fuel consumption table

Enter the 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, 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 1-146.

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 1-146. 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

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 Ω. 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 Ω_0 such that $P_{max} = g(\Omega_0)$. Define $w \equiv \Omega/\Omega_0$ and $g(\Omega) \equiv P_{max} \cdot p(w)$. 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:

$$w_\pm = \frac{1}{2}\left(-p_2 \pm \sqrt{p_2^2 + 4p_1 p_3}\right)$$
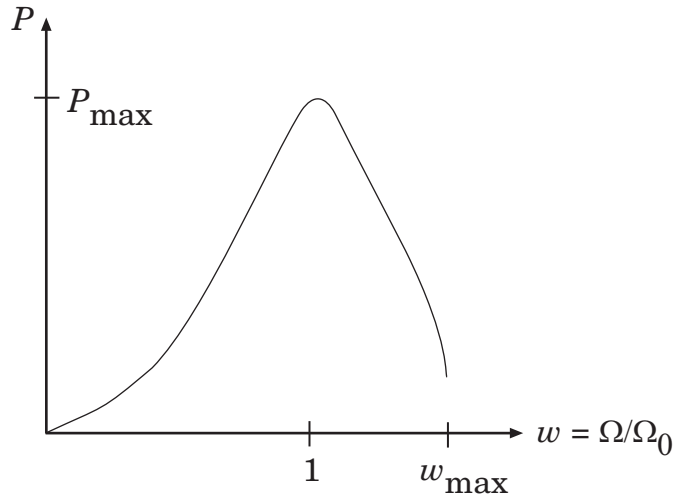
Typical Engine Power Demand Function

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: $0 \le w_{min} \le w \le w_{max}$.

- The engine power at minimum speed must be nonnegative: $p(w_{min}) \ge 0$. If you use the polynomial form, this condition is a restriction on the p_i:

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

- The engine power at maximum speed must be nonnegative: $p(w_{\max})$ ≥ 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:

$$\Pi = \max(\Pi_i, \Pi_c)$$

$$\frac{d(\Pi_c)}{dt} = \frac{0.5 \cdot \left(1 - \tanh\left(4 \cdot \dfrac{\omega - \omega_r}{\omega_t}\right)\right) - \Pi_c}{\tau}$$

where:

- Π — Engine throttle
- Π_i — Input throttle (port T)
- Π_c — Controller throttle
- ω — Engine speed

- ω_e — Idle speed reference
- ω_t — Controller speed threshold
- τ — 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

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

Generic Engine

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

Related Examples

- "Complete Vehicle Model"
- "Adjust Model Fidelity"

Purpose   High-ratio speed reducer based on elastic deformation of an elliptical gear

Library   Gears

Description   This block represents a compact, high-ratio, speed reduction mechanism that contains three key components:
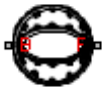
- Strain wave generator
- Elliptical gear
- Circular ring gear

The strain wave generator comprises an elliptical plug mated to a raced ball bearing. It sits inside an elastic metal gear, deforming it into a slight elliptical pattern. Rotation of the elliptical pattern in the body of the gear constitutes a strain wave.

The elliptically deformed gear engages the internal teeth of a fixed circular ring gear of only slightly larger diameter. Meshing occurs concurrently at the two elongated ends of the elliptical gear. This design doubles the teeth in mesh, boosting the torque capacity of the drive system.
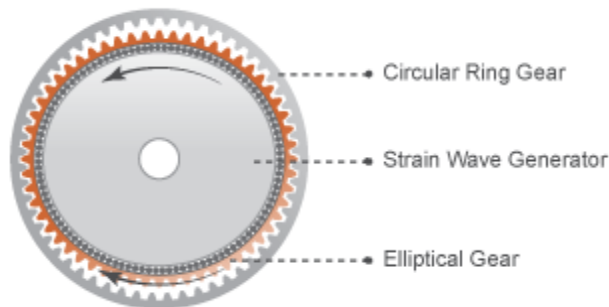
Harmonic Drive

During normal operation, the base shaft drives the strain wave generator. The elliptical plug spins freely inside the elastic metal gear, propagating the strain wave about the gear rotation axis. This strain wave causes the elliptical gear teeth to progressively engage the internal teeth of the circular ring gear.

The internal meshing between the two gears causes the elliptical gear axis to spin counter to the elliptical strain wave. For every clockwise rotation that the strain wave generator completes, the elliptical gear axis rotates counterclockwise by a small amount.

Large reduction ratios arise from the near-equal gear tooth numbers. The effective gear reduction ratio is:

$$r = \frac{n_C - n_E}{n_E},$$

where:

- r is the gear reduction ratio.
- n_C is the tooth number of the circular ring gear.
- n_E is the tooth number of the deformable elliptical gear.

Optional parameters account for power losses due to gear meshing and viscous friction. The Simple Gear block provides the foundation for this block. For more information, see Simple Gear.

Dialog Box and Parameters
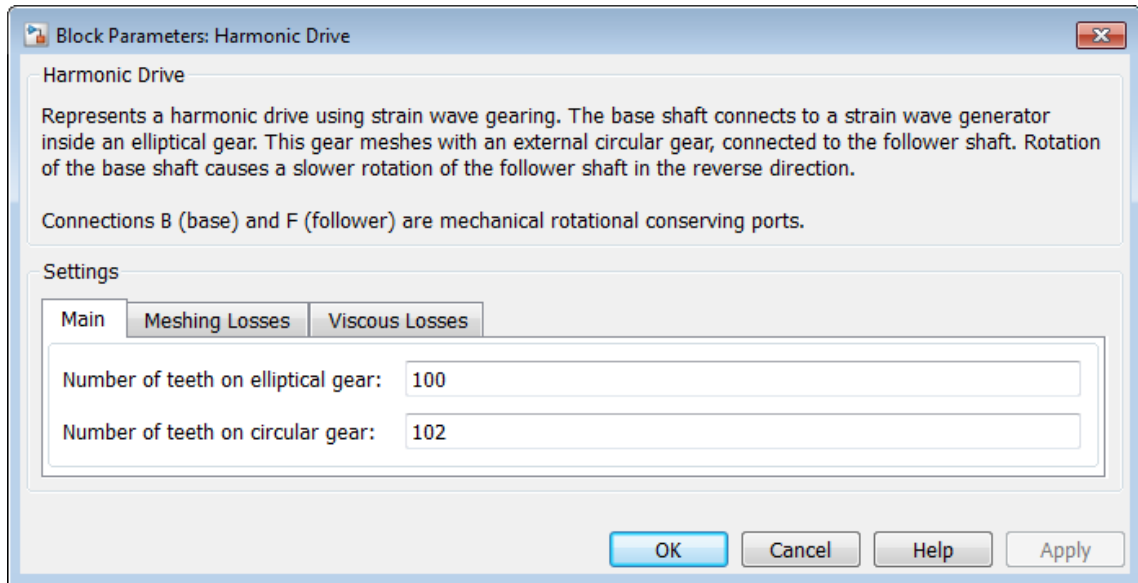
Main

Number of teeth on elliptical gear

> Total number of teeth protruding outward from the elliptical gear perimeter. This number should be slightly smaller than the number of teeth on the circular ring gear. The ratio of the two gear tooth numbers defines the relative angular velocities of the base and follower shafts. The default value is 100.

Number of teeth on circular gear

> Number of teeth protruding inward from the circular ring gear perimeter. This number should be slightly larger than the number of teeth on the elliptical gear. The ratio of the two gear

tooth numbers defines the relative angular velocities of the base and follower shafts. The default value is 102.

Meshing Losses

Friction model

List of friction models at various precision levels for estimating power losses due to meshing.

- No meshing losses - Suitable for HIL simulation — Neglect friction between gear cogs. Meshing is ideal.

- Constant efficiency — Reduce torque transfer by a constant efficiency factor. This factor falls in the range $0 < \eta \leq 1$ and is independent of load. Selecting this option exposes additional parameters.

Constant efficiency

Efficiency

Torque transfer efficiency (η) between base and follower shafts. This parameter is inversely proportional to the meshing power losses. Values can range from 0 to 1. The default value is 0.95.

- Load-dependent efficiency — Reduce torque transfer by a variable efficiency factor. This factor falls in the range $0 < \eta < 1$ and varies with the torque load. Selecting this option exposes additional parameters.

Load-Dependent Efficiency

Input shaft torque at no load

Net torque (τ_{idle}) acting on the input shaft in idle mode, e.g., when torque transfer to the output shaft equals zero. For nonzero values, the power input in idle mode

completely dissipates due to meshing losses. The default value is `0.1` N*m.

Nominal output torque

Output torque (τ_F) at which to normalize the load-dependent efficiency. The default value is `5` N*m.

Efficiency at nominal output torque

Torque transfer efficiency (η) at the nominal output torque. Efficiency values must fall in the interval [0, 1]. Larger efficiency values correspond to greater torque transfer between the input and output shafts. The default value is `0.95`.

Follower angular velocity threshold

Absolute value of the follower shaft angular velocity above which the full efficiency factor is in effect (ω_F). Below this value, a hyperbolic tangent function smooths the efficiency factor to one, lowering the efficiency losses to zero when at rest.

As a guideline, the angular velocity threshold should be lower than the expected angular velocity during simulation. Higher values might cause the block to underestimate efficiency losses. Very low values might, however, raise the computational cost of simulation.

The default value is `0.01` rad/s.

Viscous Losses

Viscous friction coefficients at base (B) and follower (F)
Two-element array with the viscous friction coefficients in effect at the base and follower shafts. The default array, `[0 0]`, corresponds to zero viscous losses.

Harmonic Drive

Ports

Port	Description
B	Conserving rotational port representing the base shaft
F	Conserving rotational port representing the follower shaft

See Also   Cycloidal Drive | Simple Gear | Simple Gear with Variable Efficiency

Purpose   Leadscrew gear set of threaded rotating screw and translating nut, with adjustable thread and friction losses

Library   Gears/Rotational-Translational
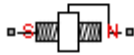
Description   The Leadscrew block represents a threaded rotational-translational gear that constrains the two connected driveline axes, screw (S) and nut(N), to, respectively, rotate and translate together in a fixed ratio that you specify. You can choose whether the nut axis translates in a positive or negative direction, as the screw rotates in a positive right-handed direction. If the screw helix is right-handed, ω_S and v_N have the same sign. If the screw helix is left-handed, ω_S and v_N have opposite signs. For model details, see "Leadscrew Gear Model" on page 1-159.

Ports

S is a rotational conserving port. N is a translational conserving port. They represent the screw and the nut, respectively.

Leadscrew

Dialog Box and Parameters
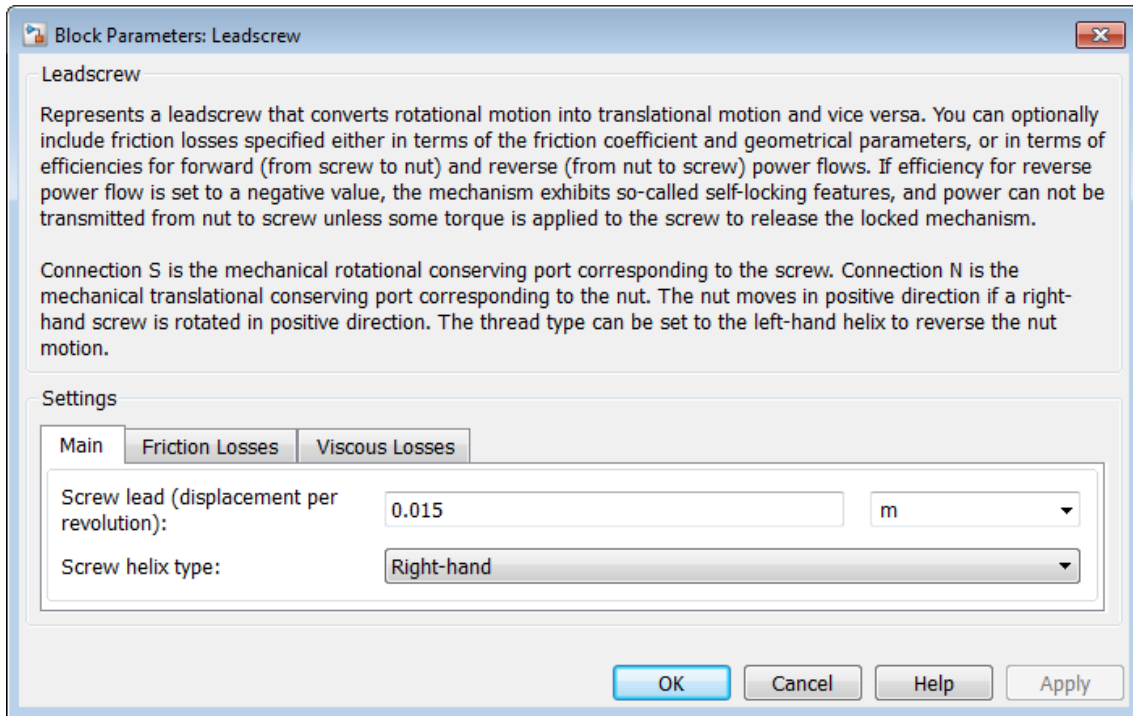
Main

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

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

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 a in the normal plane. If a square thread is simulated, $a = 0$. 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

Screw-nut efficiency

> Efficiency η_{SN} of the power transfer from screw to nut.

Nut-screw efficiency

> Efficiency η_{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

Viscous friction coefficient
Viscous friction coefficient μ_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:

$$R_{NS}\tau_S + F_N - F_{loss} = 0 ,$$

with $F_{loss} = 0$ in the ideal case.

Nonideal Gear Constraint and Losses

In the nonideal case, $F_{loss} \neq 0$. 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:

Leadscrew

- Coulomb friction between thread surfaces on S and N, characterized by friction coefficient k or constant efficiencies (η_{SN}, η_{NS}]

- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficient μ

The loss force has the general form:

$$F_{\mathrm{loss}} = F_{\mathrm{Coul}} \tanh(4v_{\mathrm{N}}/v_{\mathrm{th}}) + \mu\omega_{\mathrm{S}}/R_{\mathrm{NS}} \ .$$

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 $F_{\mathbf{Coul}}$
Forward	$\omega_{\mathrm{S}}\tau_{\mathrm{S}} > F_{\mathrm{N}}v_{\mathrm{N}}$	Nut, v_{N}	$R_{\mathrm{NS}}|\tau_{\mathrm{S}}| \cdot (1 - \eta_{\mathrm{SN}})$
Reverse	$\omega_{\mathrm{S}}\tau_{\mathrm{S}} < F_{\mathrm{N}}v_{\mathrm{N}}$	Screw, ω_{S}	$|F_{\mathrm{N}}| \cdot (1 - \eta_{\mathrm{NS}})$

Geometric Surface Contact Friction

In the contact friction case, η_{SN} and η_{NS} are determined by:

- The screw-nut threading geometry, specified by lead angle λ and acme thread half-angle a.

- The surface contact friction coefficient k.

$$\eta_{\mathrm{SN}} = (\cos a - k \cdot \tan a)/(\cos a + k/\tan\lambda) \ ,$$

$$\eta_{\mathrm{NS}} = (\cos a - k/\tan\lambda)/(\cos a + k \cdot \tan a) \ .$$

Constant Efficiencies

In the constant efficiency case, you specify η_{SN} and η_{NS}, independently of geometric details.

Self-Locking and Negative Efficiency

η_{NS} has two distinct regimes, depending on lead angle λ, separated by the *self-locking point* at which $\eta_{\mathrm{NS}} = 0$ and $\cos a = k/\tan\lambda$.

- In the *overhauling regime*, $\eta_{NS} > 0$. The force acting on the nut can rotate the screw.

- In the *self-locking regime*, $\eta_{NS} < 0$. An external torque must be applied to the screw to release an otherwise locked mechanism. The more negative is η_{NS}, the larger the torque must be to release the mechanism.

η_{SN} is conventionally positive.

Meshing Efficiency

The efficiencies η 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 μ controls the viscous friction torque experienced by the screw from lubricated, nonideal gear threads. The viscous friction torque on a screw driveline axis is $-\mu_S \omega_S$. ω_S is the angular velocity of the screw with respect to its mounting.

Limitations
- Gear inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. 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

Related Examples
- "Represent and Transfer Driveline Motion and Torque"

Loaded-Contact Rotational Friction

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.

Dialog Box and Parameters
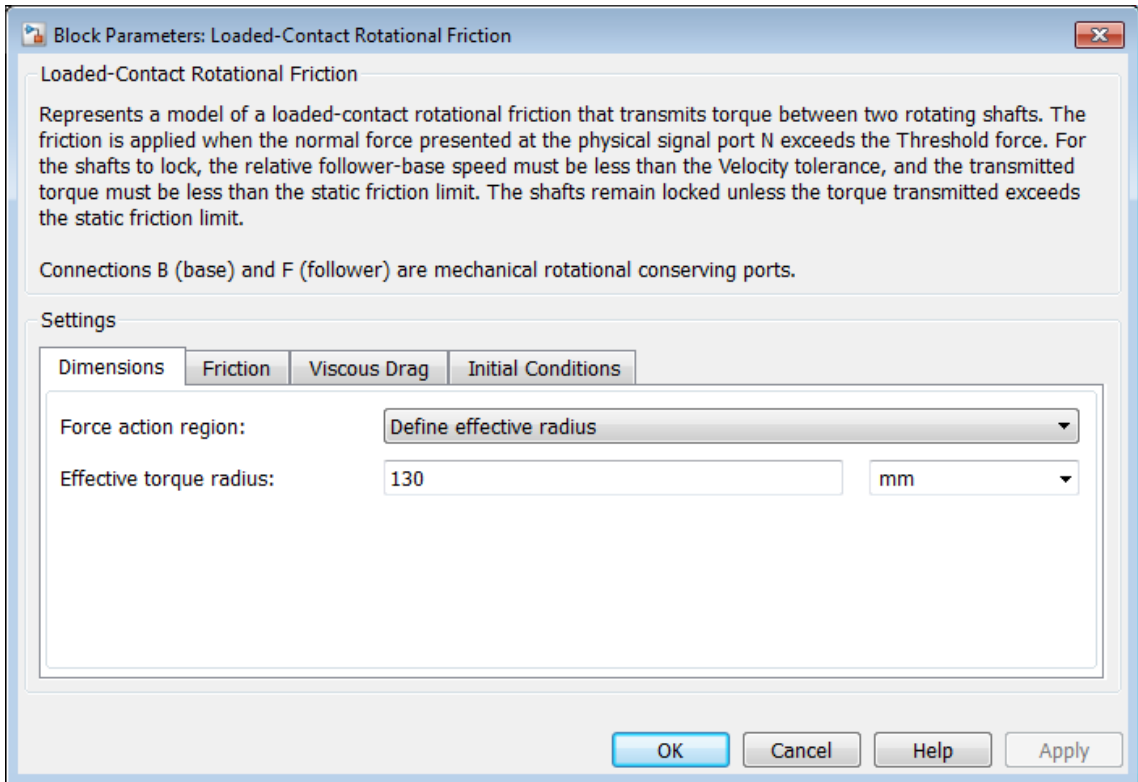
Dimensions

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_{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.

Define annular region

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

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

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 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

Viscous drag torque coefficient

Viscous drag coefficient μ_{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

Initial state

Select the initial condition applied to the Fundamental Friction Clutch state inside the block:

- `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:

$$\tau_{\text{fr}} = N \cdot \mu \cdot r_{\text{eff}} \cdot \text{sign}(\omega) + \tau_{\text{visc}} \ ,$$

$$r_{\text{eff}} = \frac{2}{3} \cdot \frac{r_{\text{o}}^3 - r_{\text{i}}^3}{r_{\text{o}}^2 - r_{\text{i}}^2} \ ,$$

$$\tau_{\text{visc}} = \mu_{\text{visc}} \cdot \omega \ ,$$

where:

τ_{fr}	Friction torque
N	Normal force
μ	Friction coefficient
r_{eff}	Effective radius
r_{o}	Disk outside radius
r_{i}	Disk inside radius
ω	Relative angular velocity
τ_{visc}	Viscous drag torque
μ_{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

Related Examples

- "Control Gear Couplings with Clutches"

Loaded-Contact Translational Friction

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.

Dialog Box and Parameters

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

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]`.

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.

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

Viscous drag force coefficient

Viscous drag coefficient μ_{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`.

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:

$$F_{\text{fr}} = N \cdot \mu \cdot \text{sign}(v_{\text{rel}}) + F_{\text{visc}} \ ,$$

$$F_{\text{visc}} = \mu_{\text{visc}} \cdot v_{rel} \ ,$$

where:

F_{fr}	Friction force
N	Normal force
μ	Friction coefficient
v_{rel}	Relative velocity
F_{visc}	Viscous drag force
μ_{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 Translational Friction

Related Examples

• "Control Gear Couplings with Clutches"

Nonlinear Rotational Damper

Purpose   Rotational damper based on polynomial or lookup-table parameterizations

Library   Couplings & Drives/Springs & Dampers
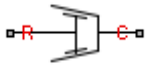
Description The block represents a nonlinear rotational damper. Polynomial and lookup-table parameterizations define the nonlinear relationship between damping torque and relative angular velocity. The damping torque can be symmetric or asymmetric about the zero velocity point. The block applies equal 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:

$$T = b_1\omega + sign(\omega) \cdot b_2\omega^2 + b_3\omega^3 + sign(\omega) \cdot b_4\omega^4 + b_5\omega^5,$$

where:

- T — Damping torque

- b_1, b_2, ..., b_5 — Damping coefficients

- ω — Relative angular velocity between ports R and C, $\omega = \omega_R - \omega_C$

- ω_R — Absolute angular velocity associated with port R

- ω_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:

$$T = \begin{cases} b_{1p}\omega + b_{2p}\omega^2 + b_{3p}\omega^3 + b_{4p}\omega^4 + b_{5p}\omega^5, & \omega \geq 0 \\ b_{1n}\omega + b_{2n}\omega^2 + b_{3n}\omega^3 + b_{4n}\omega^4 + b_{5n}\omega^5, & \omega < 0 \end{cases},$$

where:

- b_{1p}, b_{2p}, ..., b_{5p} — Damping coefficients for positive relative velocities
- b_{1n}, b_{2n}, ..., 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.

Nonlinear Rotational Damper

Block Parameters: Nonlinear Rotational Damper

Nonlinear Rotational Damper

Models a nonlinear rotational damper.

If Parameterization is set to By polynomial, torque, T, is calculated as $T = b1 * w + sign(w) * b2 * w^2 \ldots + b5 * w^5$, where b1 through b5 are the first through fifth-order coefficients, and w is the rotational velocity of port R minus the rotational velocity of port C. The unit specified is the unit of b1.

If Parameterization is set to By lookup table, torque is calculated using the tabulated values of torque and angular velocity. The torque must be zero when the angular velocity is zero.

Connections R and C are mechanical rotational conserving ports.

Settings

Parameters

Parameterization:	By polynomial
Symmetry:	Symmetric
Vector of damping coefficients:	[1e-4 0 1e-5 0 1e-6] N*m/(rad/s)

OK   Cancel   Help   Apply

Parameterization

Select damping parameterization. Options are By polynomial and By lookup table.

By polynomial

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].

Nonlinear Rotational Damper

By lookup table

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 light passing through the last data point.

Ports

Port	Description
C	Rotational conserving port
R	Rotational conserving port

See Also   Variable Rotational Damper | Nonlinear Translational damper | Variable Translational Damper

Nonlinear Rotational Spring

Purpose   Torsional spring based on polynomial or lookup table parameterizations

Library   Couplings & Drives/Springs & Dampers
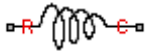
Description   The block represents a torsional spring with nonlinear
torque-displacement curve. The spring torque magnitude is a general
function of displacement. It need not satisfy Hooke's law. Polynomial
and lookup-table parameterizations provide two ways to specify the
torque-displacement relationship. The spring torque can be symmetric
or asymmetric with respect to zero deformation.

The symmetric polynomial parameterization defines spring torque
according to the expression:

$$T = k_1\theta + sign(\theta) \cdot k_2\theta^2 + k_3\theta^3 + sign(\theta) \cdot k_4\theta^4 + k_5\theta^5,$$

where:

- T — Spring force

- k_1, k_2, ...,k_5 — Spring coefficients

- θ — Relative displacement between ports R and C, $\theta = \theta_{init} + \theta_R - \theta_C$

- θ_{init} — Initial spring deformation

- θ_R — Absolute angular position of port R

- θ_C — Absolute angular position of port C

At simulation start (t=0), θ_R and θ_C are zero, making θ equal to θ_{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:

$$T = \begin{cases} k_{1t}\theta + k_{2t}\theta^2 + k_{3t}\theta^3 + k_{4t}\theta^4 + k_{5t}\theta^5, & \theta \geq 0 \\ k_{1c}\theta - k_{2c}\theta^2 + k_{3c}\theta^3 - k_{4c}\theta^4 + k_{5c}\theta^5, & \theta < 0 \end{cases},$$

where:

- k_{1t}, k_{2t}, ..., k_{5t} — Spring tension coefficients
- k_{1c}, k_{2c}, ..., 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).

Nonlinear Rotational Spring

Dialog Box and Parameters

Block Parameters: Nonlinear Rotational Spring

Nonlinear Rotational Spring

Models a nonlinear rotational spring.

If Parameterization is set to By polynomial, torque, T, is calculated as T = k1 * theta + sign(theta) * k2 * theta^2 ... + k5 * theta^5, where k1 through k5 are the first through fifth-order coefficients, and theta is the current deformation. The unit specified is the unit of k1.

If Parameterization is set to By lookup table, torque is calculated using the tabulated values of torque and deformation. The torque must be zero when the deformation is zero.

Connections R and C are mechanical rotational conserving ports.

Settings

Parameters

Parameterization:	By polynomial
Symmetry:	Symmetric
Vector of spring coefficients:	[1 0 0.1 0 0.01] N*m/rad
Initial deformation:	0 rad

OK   Cancel   Help   Apply

Parameterization

Select spring parameterization. Options are By polynomial and By lookup table.

By Polynomial
Specify nonlinear spring function in terms of polynomial coefficients.

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.

Nonlinear Rotational Spring

By lookup table

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 `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 light passing through the last data point.

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

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.
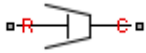
The symmetric polynomial parameterization defines the damping force for both positive and negative relative velocities according to the expression:

$$F = b_1 v + sign(v) \cdot b_2 v^2 + b_3 v^3 + sign(v) \cdot b_4 v^4 + b_5 v^5,$$

where:

- F — Damping force
- b_1, b_2, ..., 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:

$$F = \begin{cases} b_{1e}v + b_{2e}v^2 + b_{3e}v^3 + b_{4e}v^4 + b_{5e}v^5, & v \geq 0 \\ b_{1c}v - b_{2c}v^2 + b_{3c}v^3 - b_{4c}v^4 + b_{5c}v^5, & v < 0 \end{cases},$$

where:

- b_{1e}, b_{2e}, ..., b_{5e} — Damping coefficients for positive relative velocities

- b_{1c}, b_{2c}, ..., 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.

Nonlinear Translational Damper
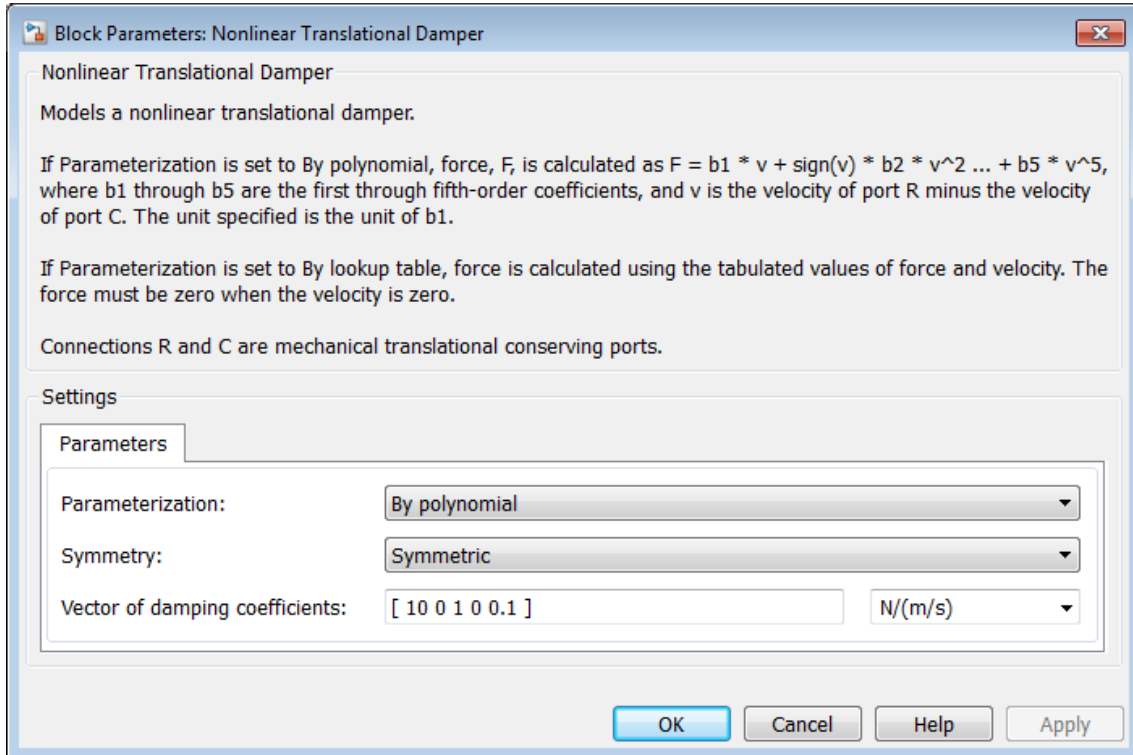
Parameterization

> 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.

- 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).

Nonlinear Translational Damper

By lookup table

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 light passing through the last lookup-table data point.

Ports

Port	Description
C	Rotational conserving port
R	Rotational conserving port

See Also Variable Translational Damper | Nonlinear Rotational Damper | Variable Rotational Damper

Nonlinear Translational Spring

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

$$F = k_1 x + sign(x) \cdot k_2 x^2 + k_3 x^3 + sign(x) \cdot k_4 x^4 + k_5 x^5,$$

where:

- F — Spring force

- k_1, k_2, ..., 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:

$$F = \begin{cases} k_{1t}x + k_{2t}x^2 + k_{3t}x^3 + k_{4t}x^4 + k_{5t}x^5, & x \geq 0 \\ k_{1c}x + k_{2c}x^2 + k_{3c}x^3 + k_{4c}x^4 + k_{5c}x^5, & x < 0 \end{cases},$$

where:

- k_1, k_2, ..., k_5 — Spring tension coefficients
- k_1, k_2, ..., 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).

Nonlinear Translational Spring

Dialog Box and Parameters

Block Parameters: Nonlinear Translational Spring

Nonlinear Translational Spring

Models a nonlinear translational spring.

If Parameterization is set to By polynomial, force, F, is calculated as F = k1 * x + sign(x) * k2 * x^2 ... + k5 * x^5, where k1 through k5 are the first through fifth-order coefficients, and x is the current deformation. The unit specified is the unit of k1.

If Parameterization is set to By lookup table, force is calculated using the tabulated values of force and deformation. The force must be zero when the deformation is zero.

Connections R and C are mechanical translational conserving ports.

Settings

Parameters

Parameterization:	By polynomial	
Symmetry:	Symmetric	
Vector of spring coefficients:	[100 0 10 0 1]	N/m
Initial deformation:	0	m

OK   Cancel   Help   Apply

Parameterization

Select spring parameterization. Options are By polynomial and By lookup table.

By polynomial

Specify coefficients of polynomial spring function.

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.

Nonlinear Translational Spring

By lookup table

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 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 light 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

Planetary Gear

Purpose   High-ratio gear reduction mechanism with sun, planet, and ring gears

Library   Gears

Description   This block represents a high-ratio gear reduction mechanism with four key components:
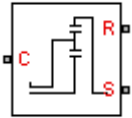
- Sun gear
- Planet Gear Set
- Planet Gear Carrier
- Ring Gear

The centrally located sun gear engages the planet gear set, which in turn engages the ring gear. A carrier holds the planet gear set. Each of these components, with the exception of the planet gear set, connects to a drive shaft.

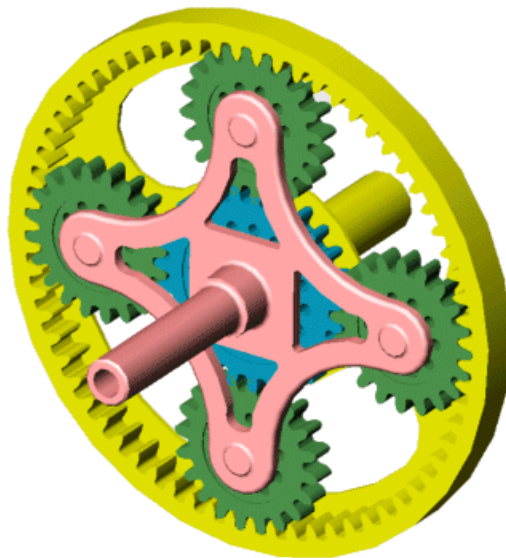Depending on which shaft is driving, driven, or fixed, the planetary gear train can achieve a variety of speed reduction ratios. These ratios are a function of the sun and ring radii, and therefore of their tooth numbers. You specify the tooth numbers directly in the block dialog box.

This block is a composite component with two underlying blocks:

- Sun-Planet

- Ring-Planet

The figure shows the connections between the two blocks.
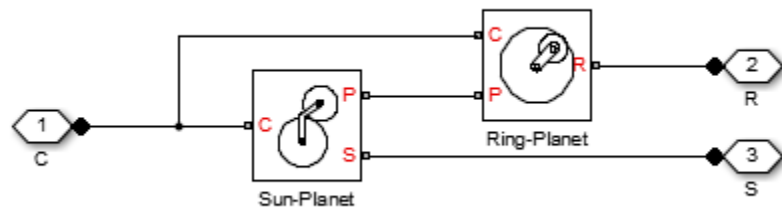
Planetary Gear

Dialog Box and Parameters
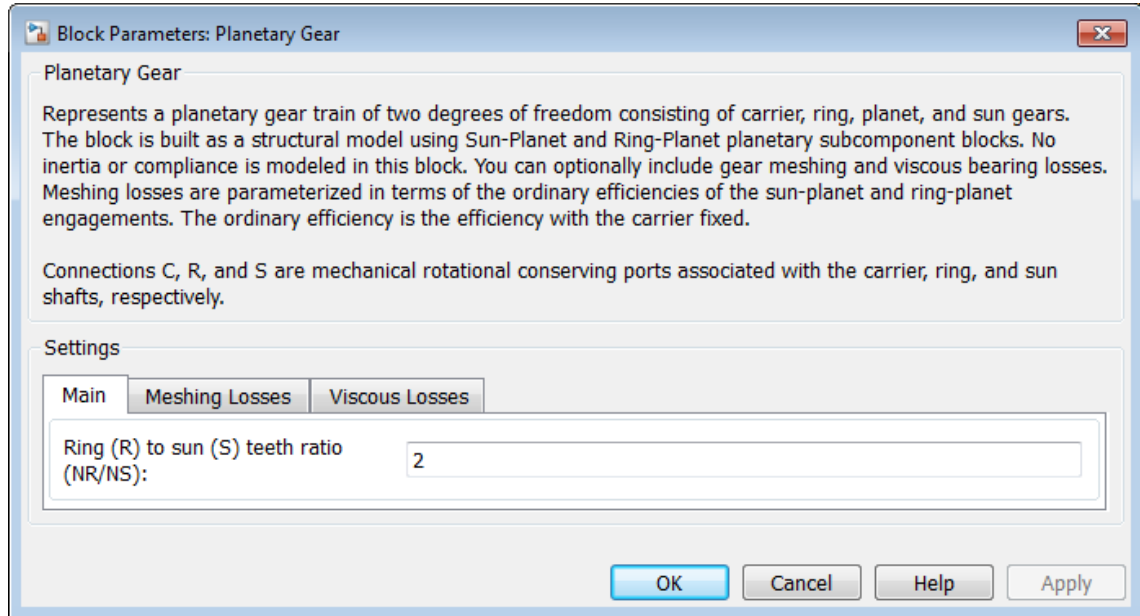
Main

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

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.

Sun-carrier and planet-carrier velocity thresholds

Vector of absolute angular velocity thresholds ω_{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

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):

$$r_C\omega_C = r_S\omega_S + r_P\omega_P \, , \; r_C = r_S + r_P \, ,$$

Planetary Gear

$$r_R\omega_R = r_C\omega_C + r_P\omega_P \ , \ r_R = r_C + r_P \ .$$

The ring-sun gear ratio $g_{RS} = r_R/r_S = N_R/N_S$. 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:

$$g_{RS}\tau_S + \tau_R - \tau_{loss} = 0 \ ,$$

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 inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

Ports

Port	Description
C	Conserving rotational port that represents the planet gear carrier
R	Conserving rotational port that represents the ring gear
S	Conserving rotational port that represents the sun gear

See Also   Double-Pinion Planetary Gear | Compound Planetary Gear |
                    Ravigneaux Gear

**Related
Examples**   • "Represent and Transfer Driveline Motion and Torque"

Planet-Planet

Purpose   Planetary gear set of carrier, inner planet, and outer planet wheels
with adjustable gear ratio and friction losses

Library   Gears/Planetary Subcomponents
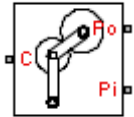
Description   The Planet-Planet gear block represents a set of carrier, inner planet,
and outer planet gear wheels. Both planetary gears are connected to
and rotate with respect to the carrier. The planets corotate with a fixed
gear ratio that you specify. For model details, see "Planet-Planet Gear
Model" on page 1-208.
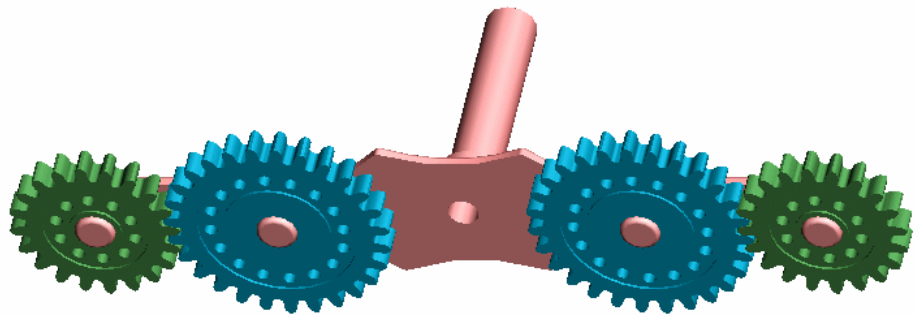
Ports

C, Po, and Pi are rotational conserving ports representing, respectively,
the carrier, outer planet, and inner planet gear wheels.

Dialog Box and Parameters
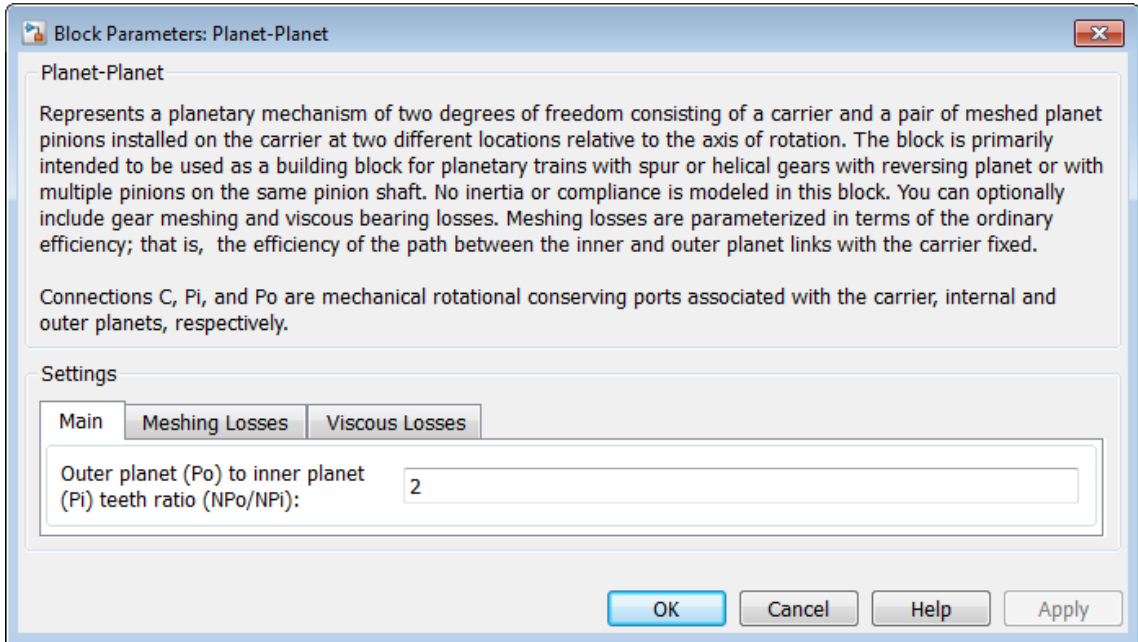
Main

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

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

Ordinary efficiency

Torque transfer efficiency η_{PP} for planet-planet gear
wheel pair meshing.

Inner planet-carrier velocity threshold

Absolute angular velocity threshold ω_{th} above which full
efficiency loss is applied, for inner planet-carrier velocity
ω_{Pi}.

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

Viscous Losses

Inner planet-carrier viscous friction coefficient

Viscous friction coefficient μ_{Pi} 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:

$$r_{\mathrm{C}}\omega_{\mathrm{C}} = r_{\mathrm{Po}}\omega_{\mathrm{Po}} + r_{\mathrm{Pi}}\omega_{\mathrm{Pi}} \, , \; r_{\mathrm{C}} = r_{\mathrm{Po}} + r_{\mathrm{Pi}} \, .$$

The outer planet-to-inner planet gear ratio $g_{oi} = r_{Po}/r_{Pi} = N_{Po}/N_{Pi}$. N is the number of teeth on each gear. In terms of this ratio, the key kinematic constraint is:

$$(1 + g_{oi})\omega_C = \omega_{Pi} + g_{oi}\omega_{Po} .$$

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 inertia is negligible. It does not impact gear dynamics.

- Gears are rigid. They do not deform.

- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

See Also

Planetary Gear | Ravigneaux Gear | Ring-Planet | Sun-Planet | Sun-Planet Bevel

Related Examples

- "Represent and Transfer Driveline Motion and Torque"

Rack & Pinion

Purpose   Rack and pinion gear coupling translational and rotational motion, with adjustable pinion radius and friction losses

Library   Gears/Rotational-Translational
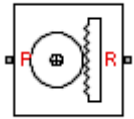
Description   The Rack & Pinion block represents rack and pinion gear that converts between translational and rotational motion. The rotational-translational gear constrains the pinion (P) and rack (R) to, respectively, rotate and translate together in a fixed ratio that you specify. You can choose whether 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.

Dialog Box and Parameters
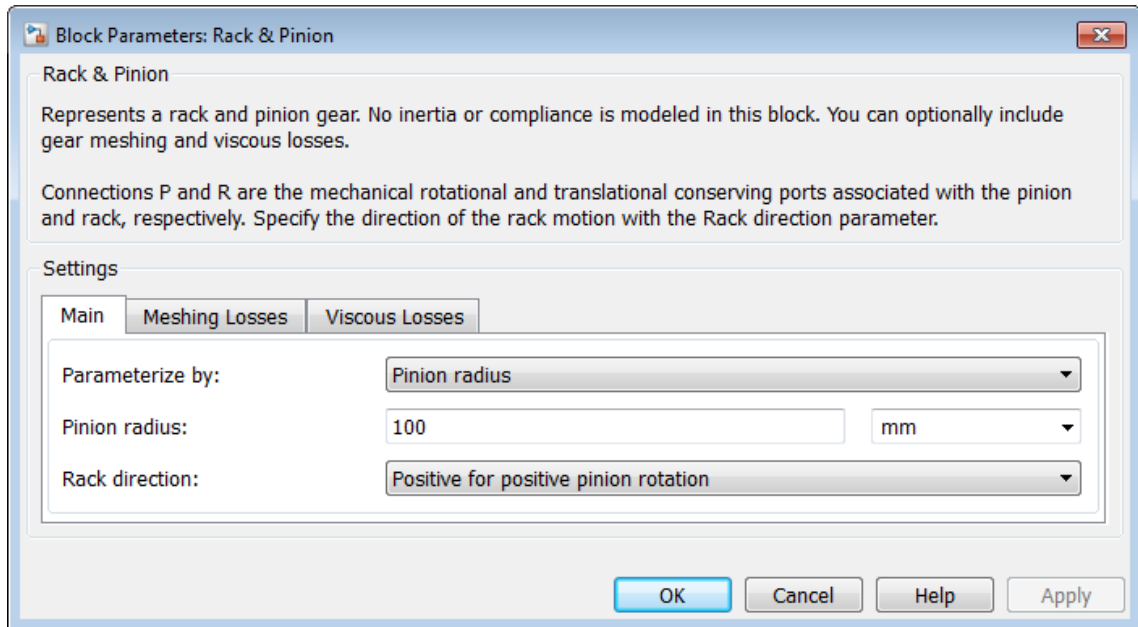
Main

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

  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.

  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

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

Efficiency

Torque transfer efficiency η 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

Pinion rotational viscous friction coefficient
Viscous friction coefficient μ_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 μ_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)`).

Gear Model

Model Variables

R_{RP} Rack-pinion gear ratio

ω_P Angular velocity of the pinion shaft

Rack & Pinion

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
τ_P	Pinion shaft torque
F_R	Rack force
F_{loss}	Total loss force
F_{Coul}	Friction force
η	Torque transfer efficiency
v_{th}	Absolute translational velocity threshold
μ_P	Viscous friction coefficient for the pinion shaft
μ_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:

$$R_{RP} = 1 / r_P = \omega_P / v_N = \pm 2\pi / N_P v_R .$$

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:

$$R_{RP} \tau_P + F_R - F_{loss} = 0 ,$$

with $F_{loss} = 0$ 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 η
- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients μ

The loss force has the general form:

$$F_{\mathrm{loss}} = F_{\mathrm{Coul}} \cdot \tanh(4 v_{\mathrm{R}} / v_{\mathrm{th}}) + \mu_{\mathrm{P}} \omega_{\mathrm{P}} R_{\mathrm{RP}} + \mu_{\mathrm{R}} v_{\mathrm{R}}.$$

The hyperbolic tangent regularizes the sign change in the Coulomb friction force when the rack velocity changes sign.

Power Flow	Power Loss Condition	Output Driveshaft	Coulomb Friction Force $F_{\mathbf{Coul}}$
Forward	$\omega_{\mathrm{P}} \tau_{\mathrm{P}} > F_{\mathrm{R}} v_{\mathrm{R}}$	Rack, v_{R}	$R_{\mathrm{RP}} \cdot \|\tau_{\mathrm{P}}\| \cdot (1 - \eta)$
Reverse	$\omega_{\mathrm{P}} \tau_{\mathrm{P}} \leq F_{\mathrm{R}} v_{\mathrm{R}}$	Pinion, ω_{P}	$R_{\mathrm{RP}} \cdot \|\tau_{\mathrm{P}}\| \cdot (1 - \eta) / \eta$

Meshing Efficiency

The efficiency η 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_{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.

Viscous Friction Force

The viscous friction coefficients μ_{P} and μ_{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 $-\mu_R v_R$.

Limitations

- Gear inertia is negligible. It does not impact gear dynamics.

- Gears are rigid. They do not deform.

- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

See Also   Leadscrew | Simple Gear

Related Examples

- "Represent and Transfer Driveline Motion and Torque"

Purpose   Planetary gear train with dual sun and planet gear sets

Library   Gears

Description
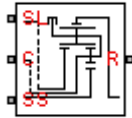
This block represents a planetary gear train with dual sun and planet gear sets. The two sun gears are centrally located and separated longitudinally along a common rotation axis. The smaller of these gears engages an inner planet gear set, which in turn engages an outer planet gear set. The outer planet gear set, whose length spans the distance between the two sun gears, engages both the larger sun gear and the ring gear.

A carrier holds the planet gear sets in place at different radii. The carrier, which rigidly connects to a drive shaft, can spin as a unit with respect to the sun and ring gears. Revolute joints, each located between a planet gear and the carrier, enable the gears to spin about their individual longitudinal axes.

The relative angular velocities of the sun, planet, and ring gears follow from the kinematic constraints between them. For more information, see "Ravigneaux Gear Model" on page 1-222.

Ravigneaux Gear
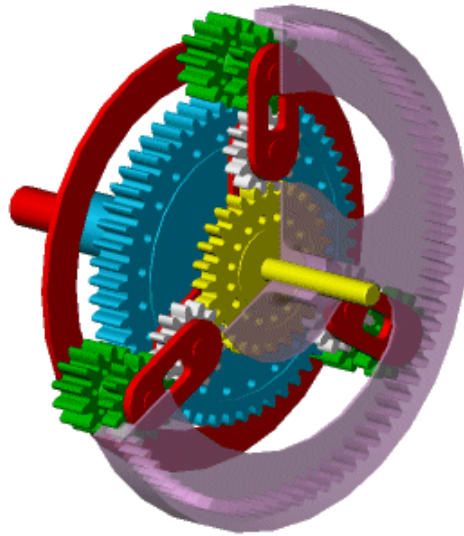
This block is a composite component with four underlying blocks:

- Ring-Planet
- Planet-Planet
- Sun-Planet (two)

The figure shows the connections between the blocks.

Ravigneaux Gear

Dialog Box and Parameters

Main

Ring (R) to large sun (SL) teeth ratio (NR/NSL)

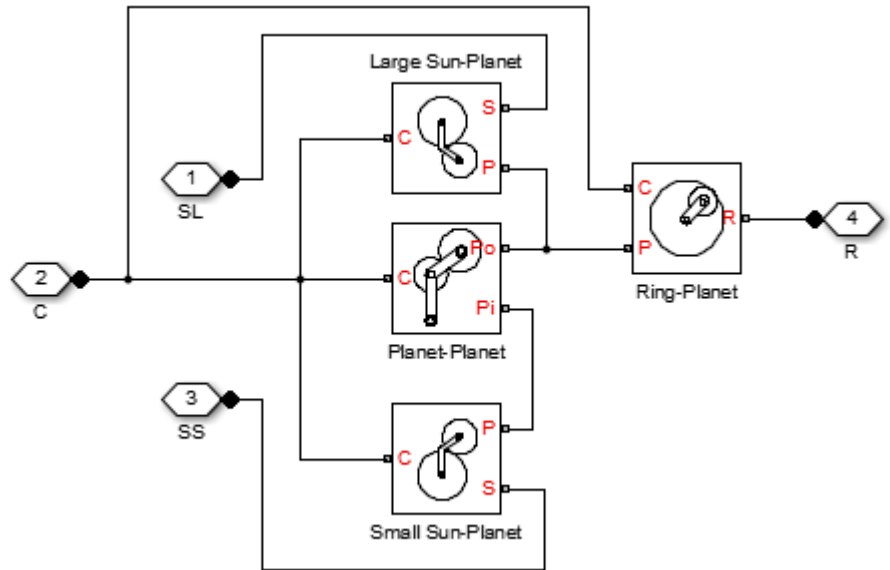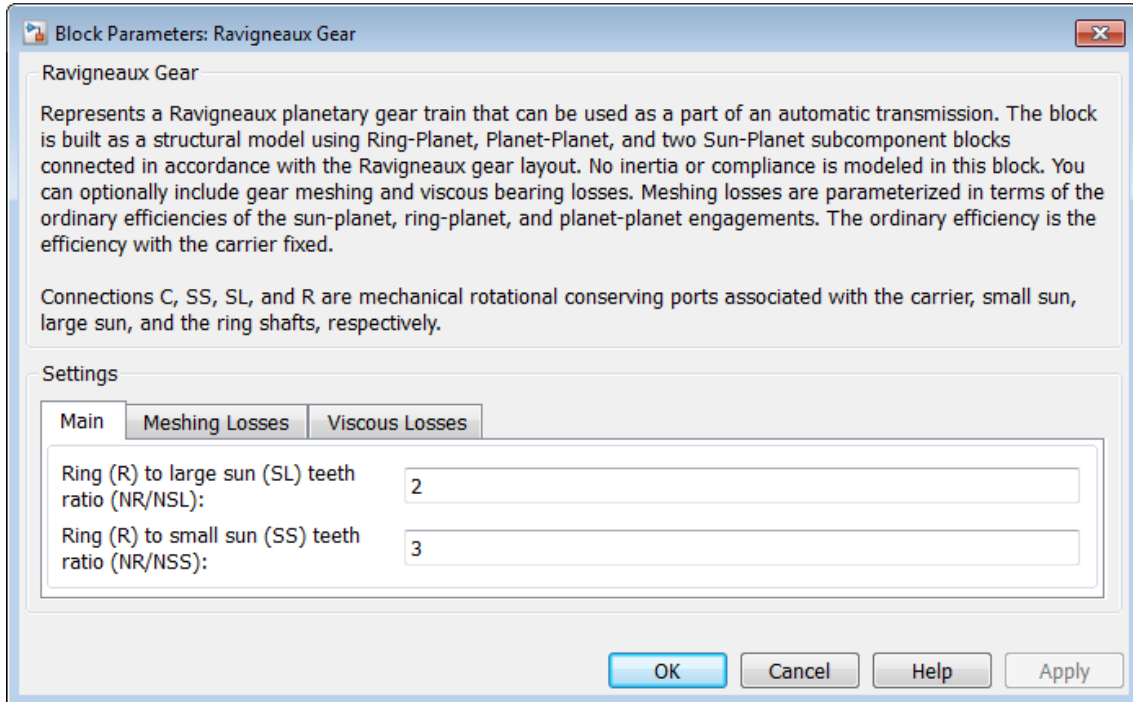Ratio g_{RSL} of the ring gear wheel radius to the large sun gear wheel radius. This gear ratio must be strictly greater than 1. The default is 2.

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

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 ω_{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).

Ravigneaux Gear

Viscous Losses

Large sun-carrier, small sun-carrier, large sun planet-carrier, and small sun planet-carrier viscous friction coefficients

> Vector of viscous friction coefficients $[\mu_{LS} \; \mu_{SS} \; \mu_{LSP} \; \mu_{SSP}]$ 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):

$$r_{Ci}\omega_C = r_{SS}\omega_{SS} + r_{Pi}\omega_{Pi} \;,\; r_{Ci} = r_{SS} + r_{Pi} \;,$$

$$r_{Co}\omega_C = r_{SL}\omega_{SL} + r_{Po}\omega_{Po} \;,\; r_{Co} = r_{SL} + r_{Po} \;,$$

$$(r_{Co} - r_{Ci})\omega_C = r_{Pi}\omega_{Pi} + r_{Po}\omega_{Po} \;,\; r_{Co} - r_{Ci} = r_{Po} + r_{Pi} \;,$$

$$r_R\omega_R = r_{Co}\omega_C + r_{Po}\omega_{Po} \;,\; r_R = r_{Co} + r_{Po} \;.$$

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 inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

Example
The sdl_Ravigneaux_4_speed example model uses a Ravigneaux gear as the core of its transmission.

Ports
- C — Rotational conserving port representing the planet gear carrier.
- R — Rotational conserving port representing the ring gear.
- SL — Rotational conserving port representing the large sun gear.
- SS — Rotational conserving port representing the small sun gear.

See Also
Compound Planetary Gear | Planet-Planet | Planetary Gear | Ring-Planet | Sun-Planet

Related Examples
- "Represent and Transfer Driveline Motion and Torque"

Ring-Planet

Purpose   Planetary gear set of carrier, planet, and ring wheels with adjustable gear ratio and friction losses

Library   Gears/Planetary Subcomponents
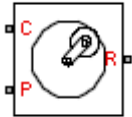
Description   The Ring-Planet gear block represents a set of carrier, planet, and ring gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and ring corotate with a fixed gear ratio that you specify. A ring-planet and a sun-planet gear are basic elements of a planetary gear set. For model details, see "Ring-Planet Gear Model" on page 1-226.

Ports

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

Dialog Box and Parameters
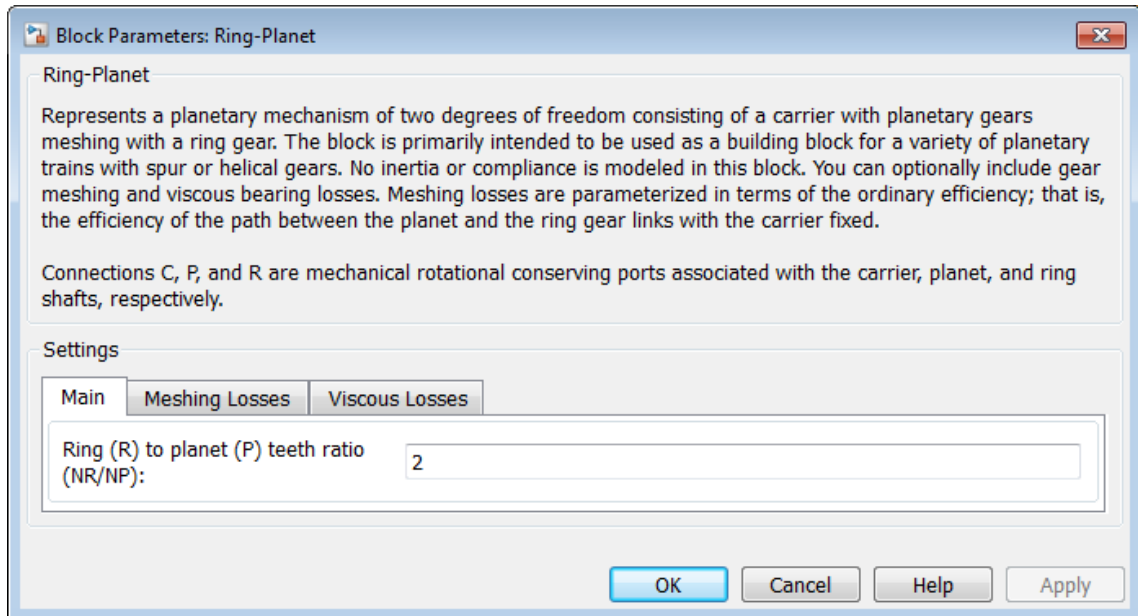
Main

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

Ordinary efficiency

Torque transfer efficiency η_{RP} for ring-planet gear wheel pair meshing.

Planet-carrier velocity threshold

Absolute angular velocity threshold ω_{th} above which full efficiency loss is applied, for planet-carrier velocity ω_P.

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

Viscous Losses

Planet-carrier viscous friction coefficient

Viscous friction coefficient μ_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:

$$r_R \omega_R = r_C \omega_C + r_P \omega_P \, , \; r_R = r_C + r_P \, .$$

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:

$$g_{\mathrm{RP}}\omega_{\mathrm{R}} = \omega_{\mathrm{P}} + (g_{\mathrm{RP}} - 1)\omega_{\mathrm{C}} \ .$$

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:

$$g_{\mathrm{RP}}\tau_{\mathrm{P}} + \tau_{\mathrm{R}} - \tau_{\mathrm{loss}} = 0 \ ,$$

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

Nonideal Gear Constraints and Losses

In the nonideal case, $\tau_{\mathrm{loss}} \neq 0$. See "Model Gears with Losses".

Limitations
- Gear inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. 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

Related Examples
- "Represent and Transfer Driveline Motion and Torque"
- "Adjust Model Fidelity"

Rolling Resistance

Purpose   Model rolling resistance

Library   Tires & Vehicles/Tire Subcomponents
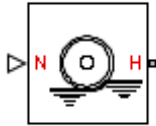
Description   The block models the resistance force that acts on the wheel hub due to the rolling resistance at the road-wheel contact surface. The model can use a constant resistance coefficient or the pressure and velocity dependence of the SAE J2452 standard. The resistance force is zero when the 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
- μ — Rolling resistance coefficient

The rolling resistance coefficient has a hyperbolic form that eliminates discontinuity at $v_{hub} = 0$:

$$\mu = \mu_0 \tanh(4 v_{hub}/v_{threshold})$$

where parameters represent the following quantities:

- μ_0 — Asymptotic rolling resistance coefficient
- v_{hub} — Hub velocity
- $v_{threshold}$ — Threshold velocity

Pressure and Velocity Dependent Model

The pressure- and velocity-dependent model uses the following formula:

$$F = \left(\frac{P}{P_0}\right)^{\alpha} \left(\frac{N}{N_0}\right)^{\beta} N_0 \cdot \left(A + B|v_{hub}| + Cv_{hub}{}^2\right)$$

where parameters represent the following quantities:

- P — Tire pressure
- v_{hub} — Hub velocity
- α, β, 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.

Rolling Resistance

Dialog Box and Parameters

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.

Settings

Parameters

Resistance model:	Constant coefficient
Constant coefficient:	0.015
Velocity threshold:	0.001 m/s

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`.

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 1-229. 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`.

Rolling Resistance

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^2/m^2`.

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)

Purpose   Power transmission system with tightly wound rope around cylindrical drum

Library   Couplings & Drives

Description   This block represents a power transmission system with two key components:
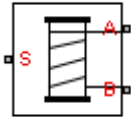
• Rope

• Drum

The rope is tightly wound around the cylindrical drum, at a sufficient tension level to prevent slipping between the two components. The rope can be wound such that its ends point in equal or opposite directions. Depending on the rope configuration, the two rope ends can move in equal or opposite directions. These are the rope windup types that you specify in the block dialog box:

• If the rope ends point in equal directions, the two translate in opposite directions.

• If the rope ends point in opposite directions, the two translate in equal directions.

Rope Drum

Crossed Rope Configuration   Open Rope Configuration
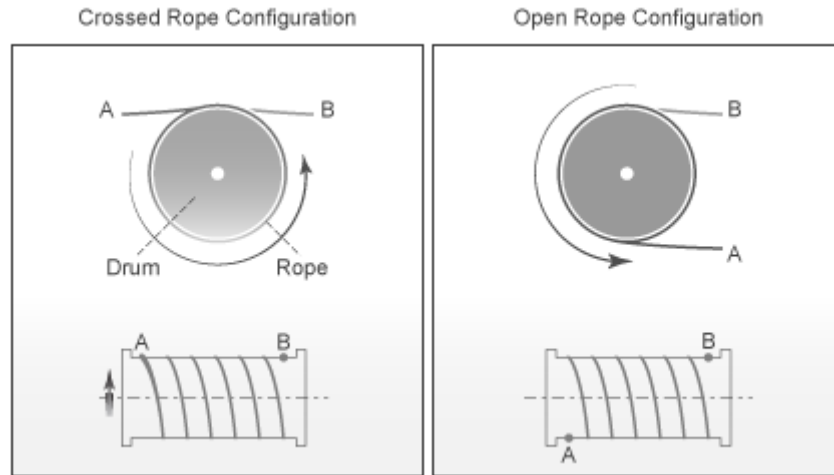
During normal operation, a driving torque causes the cylindrical drum to rotate about its symmetry axis. The rotating drum transmits a tensile force to the rope ends, which translate with respect to the drum center line. The relative direction of translation between the two rope ends depends on the rope windup type. However, a positive drum rotation always corresponds to a positive translation at rope end A.

The rope drum model assumes that each rope end remains taut during simulation. This assumption is important since slack rope ends do not transmit force. An optional tension warning indicates if this assumption fails. The warning, which appears at the MATLAB command prompt, should prompt careful analysis of the simulation results.

Optional effects in the rope drum model include:

- Drum inertia
- Viscous friction at the drum bearing

The net torque acting on the cylinder satisfies the force balance equation

$$T = F_B \cdot R \cdot \delta - F_A \cdot R + \mu \cdot \omega,$$

where:

- T is the net torque acting on the drum.

- F_A and F_B are the external forces pulling on rope ends A and B.

- R is the drum radius.

- μ is the viscous friction coefficient at the drum bearings.

- ω is the drum angular velocity.

- δ is the rope windup type according to the table below.

Rope Windup Type	δ
Ends move in the same direction	-1
Ends move in opposite directions	+1

The figure shows the equation variables.

Crossed Rope Configuration   Open Rope Configuration
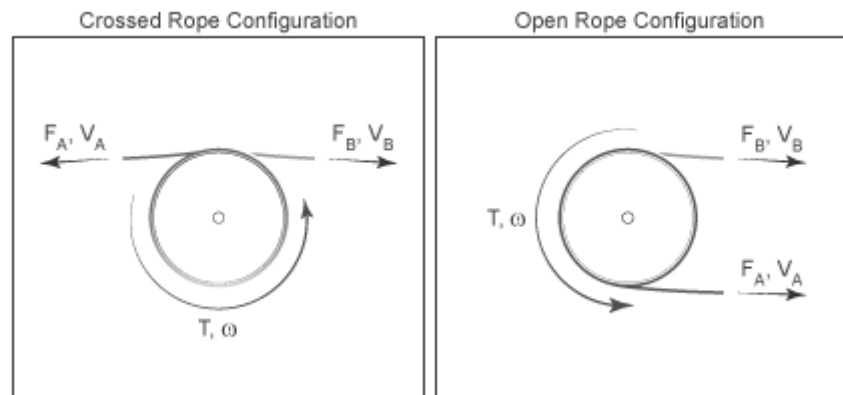
The translational velocities of the two rope ends are functions of the drum radius and angular velocity. Each translational velocity is equal

to the tangential velocity of a point at the drum periphery according to the expressions:

$$V_A = -\omega \cdot R,$$
$$V_B = +\omega \cdot R,$$

where:

- V_A is the translational velocity of rope end A.

- V_B is the translational velocity of rope end B.

Assumptions and Limitations

- Slip does not occur at the rope-drum contact surface.

- Rope compliance is ignored.

**Dialog
Box and
Parameters**
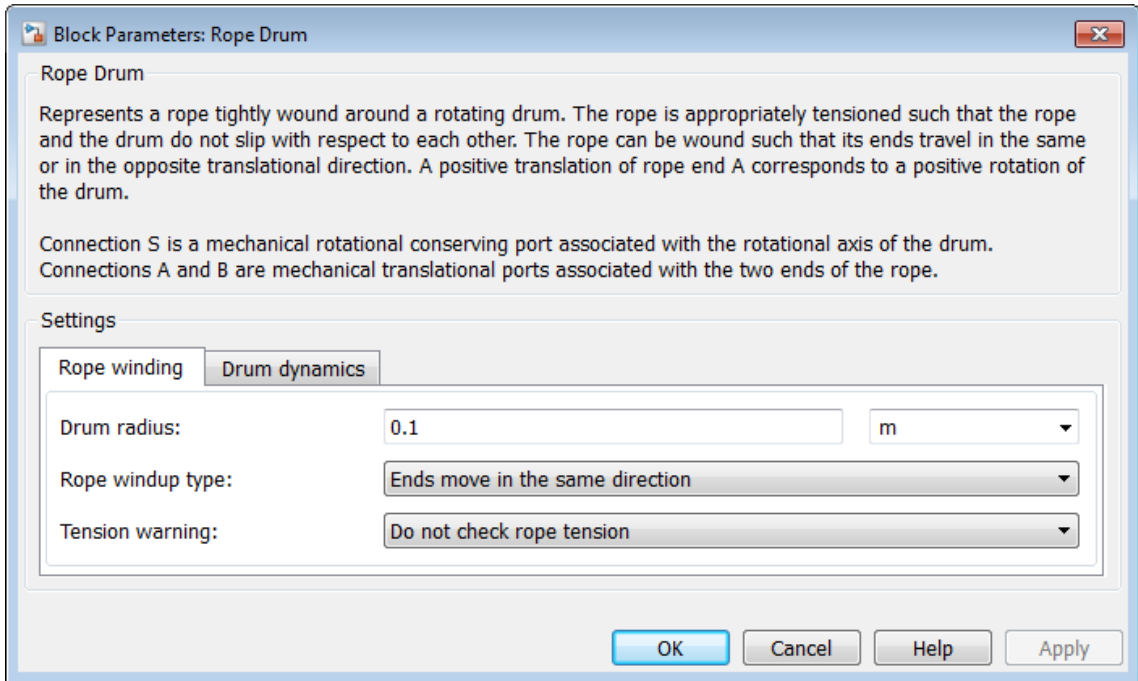
Rope Winding

Drum radius
> Distance between the drum center and its periphery. The drum periphery coincides with the contact surface between the drum and the rope. The default value is 0.1 m.

Rope windup type
> Relative direction of the translation motion of the two rope ends, A and B. Options include:

> - The ends move in the same direction

- The ends move in opposite direction

Tension warning

Optional warning at the MATLAB command line indicating that at least one rope end has become slack. A rope end becomes slack when the net tensile forces responsible for keeping it taut fall below zero. Since slack cables do not transmit force, you must interpret simulation results carefully if a tension warning appears at the MATLAB command line. Warning options include:

- Do not check rope tension
- Warn if either rope end loses tension

Drum Dynamics

Bearing viscous friction coefficient

Linear damping coefficient in effect at the drum bearing. At a given drum angular velocity, this coefficient determines the power losses due to viscous friction. The default value is 0.001 (N*m)/(rad/s).

Inertia

Optional parameter accounting for drum inertia, its resistance to angular acceleration. Options include:

- No inertia — Default mode. Keep this mode if drum inertia has a negligible impact on driveline dynamics. Selecting this mode sets drum inertia to zero.

- Specify drum inertia and initial velocity — Select this mode if drum inertia has a significant impact on driveline dynamics. Selecting this mode exposes additional parameters enabling you to specify the drum inertia and initial angular velocity.

Drum inertia

Moment of inertia of the drum about its rotation axis. In the simple case of a solid cylindrical drum, the moment of inertia is

$$I = \frac{1}{2}MR^2,$$

where M is the drum mass and R is the drum radius.

The default value of the drum inertia is `0.01` kg*m^2.

Drum initial velocity

Angular velocity of the drum at simulation time zero. The default value is `0` rad/s.

	Port	Description
Ports	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

Related Examples   • "Basic Motion, Torque, and Force Modeling"

Rotational Detent

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.

Dialog Box and Parameters

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.

Settings

| Detent Characteristics | Spacing | Friction |

Parameterization: By peak torque and detent width

Peak torque: 0.25 N*m

Detent width: 5 deg

Initial relative rotation: 0 deg

OK   Cancel   Help   Apply

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.

Rotational Detent

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.

Lookup Table

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.

For more information on interpolation algorithms, see the PS Lookup Table (1D) block reference page.

Spacing

Detent spacing

Select how to specify the detent spacing. The default is Regularly spaced.

- Regularly spaced — Specify detent spacing by regular intervals.

Rotational Detent

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

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

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) (N*m/(rad/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 τ 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.

Rotational Detent
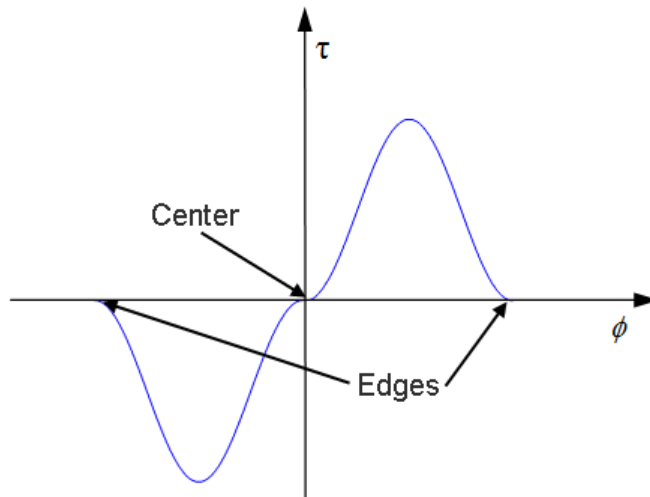
Lookup Table

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 | Tire
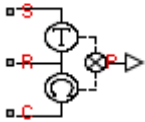(Friction Parameterized)

Concepts   • "Clutches, Clutch-Like Elements, and Coulomb Friction"

Rotational Power Sensor

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.

Dialog Box and Parameters
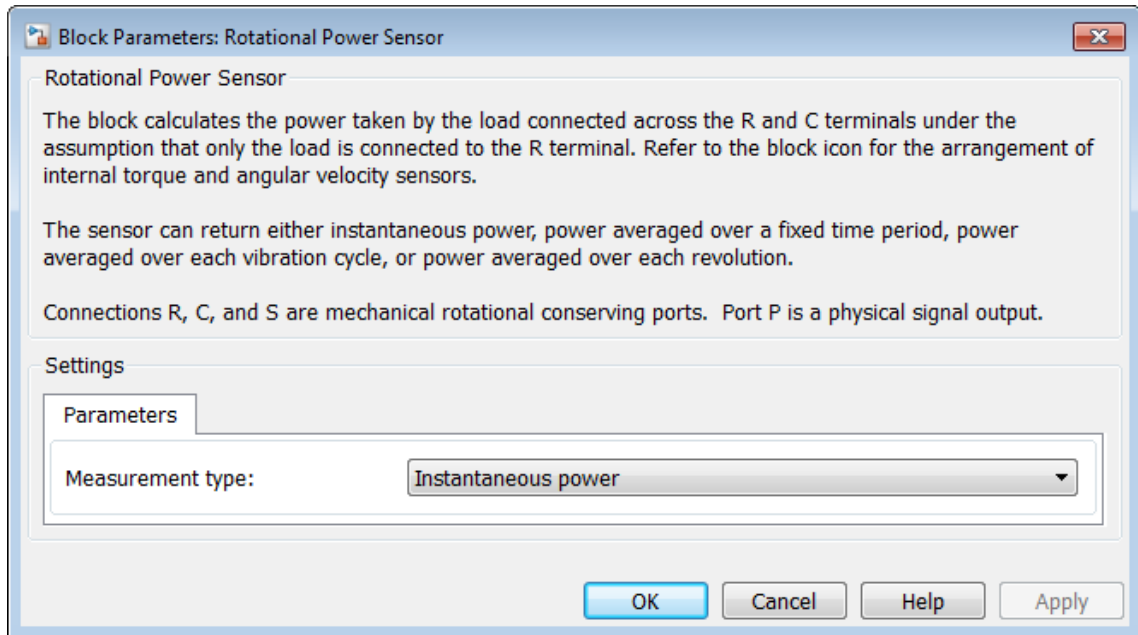
Measurement Type

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.

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.

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.

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

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

Shock Absorber

Purpose   Translational spring-damper system

Library   Couplings & Drives
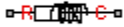
Description   The block represents a spring-damper system commonly used to dampen
vibration in mechanical systems. An internal force acts between ports
R and C. This force is the sum of spring stiffness, damping, Coulomb
friction, and hard-stop contributions. All force contributions are
optional. The net internal force satisfies the expression

$$F = F_{spr} + F_d + F_c + F_{HS},$$

where:

- F — Net internal force

- F_{spr} — Spring force, $F_k = -k \cdot x$

- F_d — Damping force, $F_d = -b \cdot v$

- F_c — Coulomb friction force,

$$F_c = \begin{cases} -F_k \cdot \text{sign}(v), & \text{when unlocked} \\ -F_s, & \text{when locked} \end{cases},$$

- F_{HS} — Hard stop force,

$$F_{HS} = \begin{cases} -k_{HS} \cdot (x - UB) - b_{HS} \cdot v, & x \geq UB \\ 0 & UB \geq x \geq LB, \\ -k_{HS} \cdot (x - LB) + b_{HS} \cdot v, & LB > x \end{cases}$$

- k — Restoring spring stiffness

- b — Viscous damping coefficient

- x — Relative displacement between ports R and C

- v — Relative velocity between R and C

- 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.

Shock Absorber

Dialog Box and Parameters
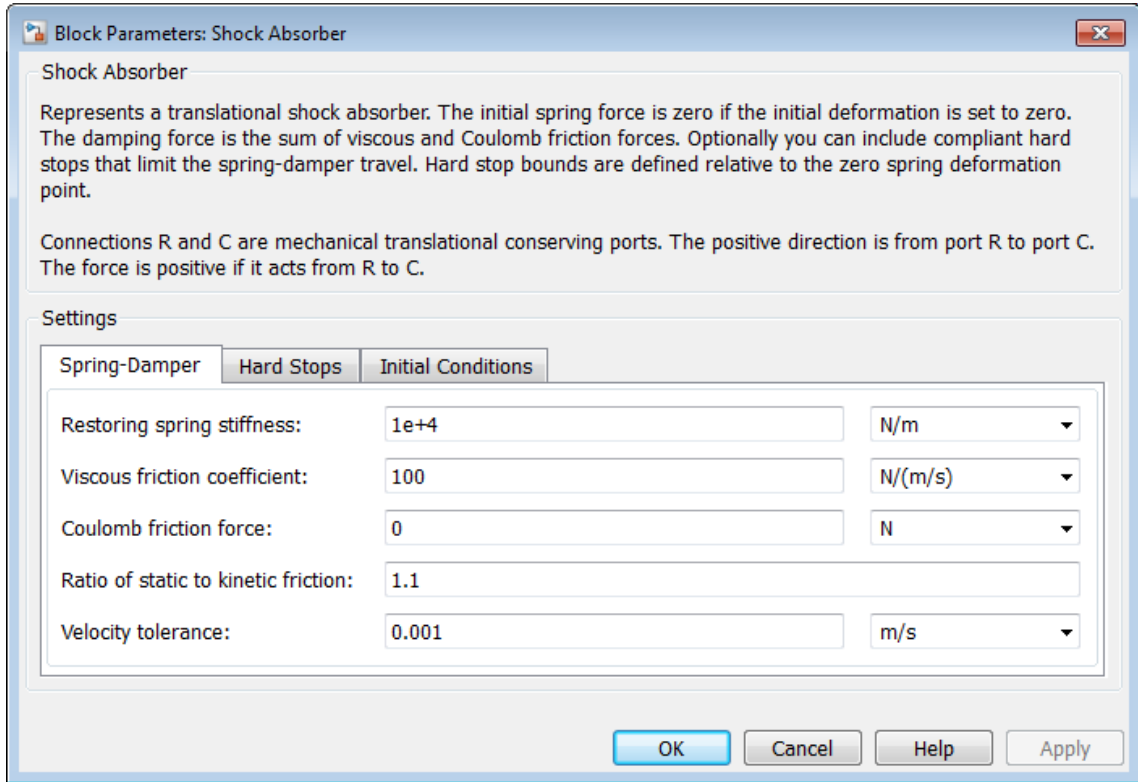
Spring-Damper

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.

The default setting is No hard stops   Suitable for HIL simulation.

Shock Absorber

Compliant hard stops

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)`.

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

Purpose
Simple gear of base and follower wheels with adjustable gear ratio and friction losses

Library
Gears

Description
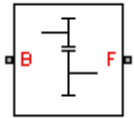
The Simple Gear block represents a gearbox that constrains the two connected driveline axes, base (B) and follower (F), to corotate with a fixed ratio that you specify. You can choose whether the follower axis rotates in the same or opposite direction as the base axis. If they rotate in the same direction, ω_F and ω_B have the same sign. If they rotate in opposite directions, ω_F and ω_B have opposite signs. For model details, see "Simple Gear Model" on page 1-261.

Ports

B and F are rotational conserving ports representing, respectively, the base and follower gear wheels.

Dialog Box and Parameters
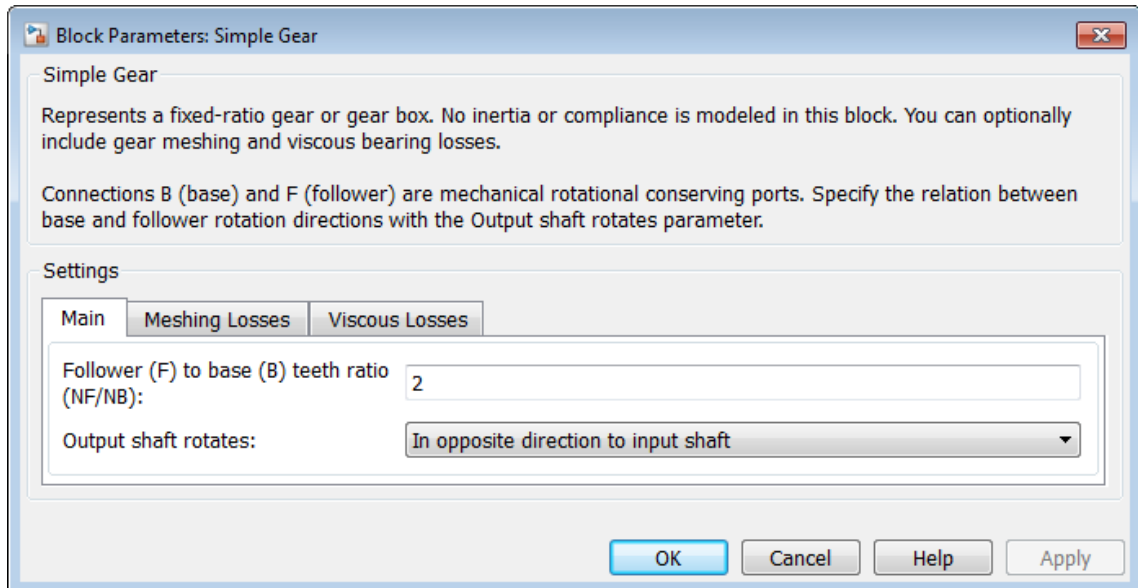
Main

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.

Output shaft rotates

> 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.

Simple Gear

Meshing Losses

Friction model

List of friction models at various precision levels for estimating power losses due to meshing.

- `No meshing losses - Suitable for HIL simulation` — Neglect friction between gear cogs. Meshing is ideal.

- `Constant efficiency` — Reduce torque transfer by a constant efficiency factor. This factor falls in the range $0 < \eta \leq 1$ and is independent of load. Selecting this option exposes additional parameters.

Constant efficiency

Efficiency

Torque transfer efficiency (η) between base and follower shafts. This parameter is inversely proportional to the meshing power losses. Values can range from 0 to 1. The default value is 0.95.

- `Load-dependent efficiency` — Reduce torque transfer by a variable efficiency factor. This factor falls in the range $0 < \eta < 1$ and varies with the torque load. Selecting this option exposes additional parameters.

Load-Dependent Efficiency

Input shaft torque at no load

Net torque (τ_{idle}) acting on the input shaft in idle mode, e.g., when torque transfer to the output shaft equals zero. For nonzero values, the power input in idle mode completely dissipates due to meshing losses. The default value is 0.1 N*m.

Nominal output torque

Output torque (τ_F) at which to normalize the load-dependent efficiency. The default value is 5 N*m.

Efficiency at nominal output torque

Torque transfer efficiency (η) at the nominal output torque. Efficiency values must fall in the interval [0, 1]. Larger efficiency values correspond to greater torque transfer between the input and output shafts. The default value is 0.95.

Follower angular velocity threshold

Absolute value of the follower shaft angular velocity above which the full efficiency factor is in effect (ω_F). Below this value, a hyperbolic tangent function smooths the efficiency factor to one, lowering the efficiency losses to zero when at rest.

As a guideline, the angular velocity threshold should be lower than the expected angular velocity during simulation. Higher values might cause the block to underestimate efficiency losses. Very low values might, however, raise the computational cost of simulation.

The default value is 0.01 rad/s.

Viscous Losses

Viscous friction coefficients at base (B) and follower (F)
Two-element array with the viscous friction coefficients in effect at the base and follower shafts. The default array, [0 0], corresponds to zero viscous losses.

Simple Gear Model

Ideal Gear Constraint and Gear Ratio

Simple Gear imposes one kinematic constraint on the two connected axes:

Simple Gear

$r_F \omega_F = r_B \omega_B$.

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:

$g_{FB} \tau_B + \tau_F - \tau_{loss} = 0$,

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

Nonideal Gear Constraint and Losses

In the nonideal case, $\tau_{loss} \neq 0$. 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 η

- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients μ

$\tau_{loss} = \tau_{Coul} \cdot \tanh(4\omega_{out}/\omega_{th}) + \mu\omega_{out}$, $\tau_{Coul} = |\tau_F| \cdot (1 - \eta)$.

The hyperbolic tangent regularizes the sign change in the Coulomb friction torque when the angular velocity changes sign.

Power Flow	Power Loss Condition	Output Driveshaft out
Forward	$\omega_B \tau_B > \omega_F \tau_F$	Follower, ω_F
Reverse	$\omega_B \tau_B < \omega_F \tau_F$	Base, ω_B

Constant Efficiency

In the constant efficiency case, η is constant, independent of load or power transferred.

Load-Dependent Efficiency

In the load-dependent efficiency case, η 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. η is related to τ_{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 inertia is negligible. It does not impact gear dynamics.

- Gears are rigid. They do not deform.

- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

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

Related Examples
- "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.
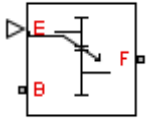
Dialog Box and Parameters

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 inertia or 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.

Settings

| Main | Meshing Losses | Viscous Losses |

Follower (F) to base (B) teeth ratio (NF/NB): 2

Output shaft rotates: In opposite direction to input shaft

OK   Cancel   Help   Apply

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. Options include equal or opposite directions. The default setting is In opposite direction to input shaft.

Meshing Losses

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.

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

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).

Simple Gear with Variable Efficiency

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:

$$g_{FB} \tau_B + \tau_F - \tau_{loss} = 0 ,$$

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

Nonideal Gear Constraint and Losses

In the nonideal case, $\tau_{loss} \neq 0$. 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 η

- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients μ

$$\tau_{loss} = \tau_{Coul} \cdot \tanh(4\omega_{out}/\omega_{th}) + \mu\omega_{out} , \quad \tau_{Coul} = |\tau_F| \cdot (1 - \eta) .$$

The hyperbolic tangent regularizes the sign change in the Coulomb friction torque when the angular velocity changes sign.

Power Flow	Power Loss Condition	Output Driveshaft out
Forward	$\omega_B \tau_B > \omega_F \tau_F$	Follower, ω_F
Reverse	$\omega_B \tau_B < \omega_F \tau_F$	Base, ω_B

Limitations

- Gear inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

Ports

Port	Description
B	Rotational conserving port representing the base shaft
F	Rotational Conserving port representing the follower shaft

See Also

Simple Gear | Variable Ratio Transmission

Related Examples

- "Represent and Transfer Driveline Motion and Torque"

Sun-Planet

Purpose   Planetary gear set of carrier, planet, and sun wheels with adjustable gear ratio and friction losses

Library   Gears/Planetary Subcomponents
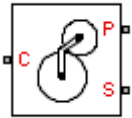
Description   The Sun-Planet gear block represents a set of carrier, planet, and sun gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and sun corotate with a fixed gear ratio that you specify and in the same direction with respect to the carrier. A sun-planet and a ring-planet gear are basic elements of a planetary gear set. For model details, see "Sun-Planet Gear Model" on page 1-270.

Ports

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

**Dialog
Box and
Parameters**
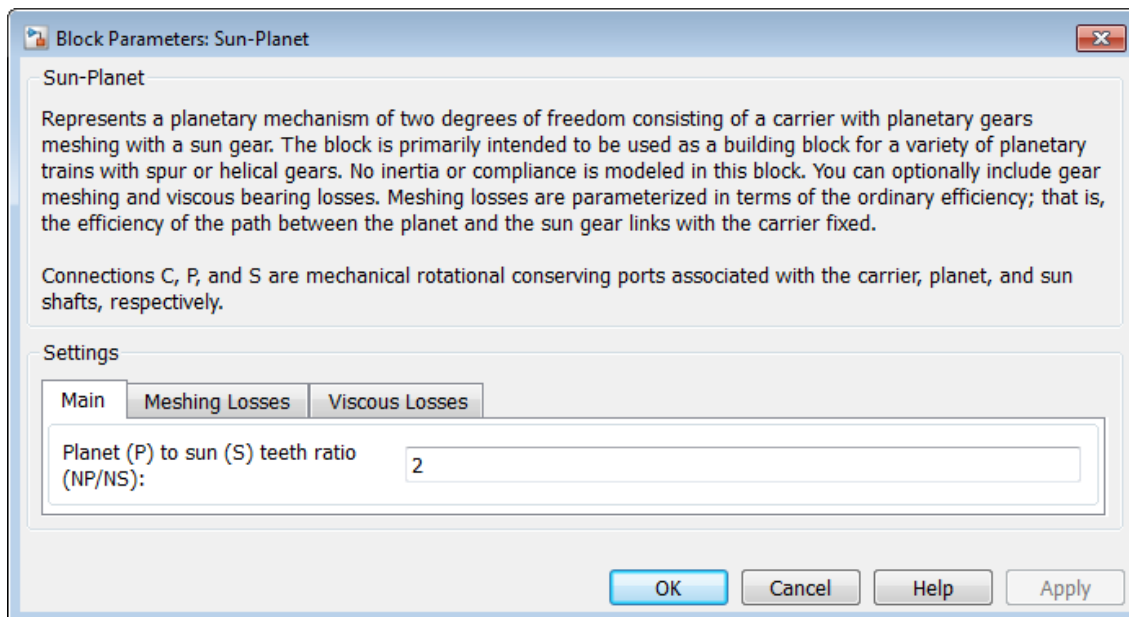
Main

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

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 η satisfying $0 < \eta$
  ≤ 1. If you select this option, the panel changes from its default.

Constant Efficiency

Ordinary efficiency

Torque transfer efficiency η_{SP} for sun-planet gear wheel
pair meshing.

Sun-carrier velocity threshold

Absolute angular velocity threshold ω_{th} above which full
efficiency loss is applied, for sun-carrier velocity ω_S.

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

Viscous Losses

Sun-carrier viscous friction coefficient

Viscous friction coefficient μ_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:

$$r_C \omega_C = r_S \omega_S + r_P \omega_P , \; r_C = r_P + r_S .$$

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:

$$\omega_{\mathrm{S}} = -g_{\mathrm{PS}}\omega_{\mathrm{P}} + (1 + g_{\mathrm{PS}})\omega_{\mathrm{C}} .$$

The three degrees of freedom reduce to two independent degrees of freedom. The gear pair is $(1,2) = (\mathrm{S},\mathrm{P})$.

Warning

The planet-sun gear ratio g_{PS} must be strictly greater than one.

The torque transfer is:

$$g_{\mathrm{PS}}\tau_{\mathrm{S}} + \tau_{\mathrm{P}} - \tau_{\mathrm{loss}} = 0 ,$$

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

Nonideal Gear Constraints and Losses

In the nonideal case, $\tau_{\mathrm{loss}} \neq 0$. See "Model Gears with Losses".

Limitations
- Gear inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

See Also
Compound Planetary Gear | Planet-Planet | Planetary Gear | Ring-Planet | Sun-Planet Bevel

Related Examples
- "Represent and Transfer Driveline Motion and Torque"

Sun-Planet Bevel

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
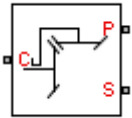
The Sun-Planet Bevel gear block represents a set of carrier, planet, and sun gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and sun corotate with a fixed gear ratio and in a direction that you specify. You control the direction by setting the assembly orientation, 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 1-275.

Ports

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

Dialog Box and Parameters
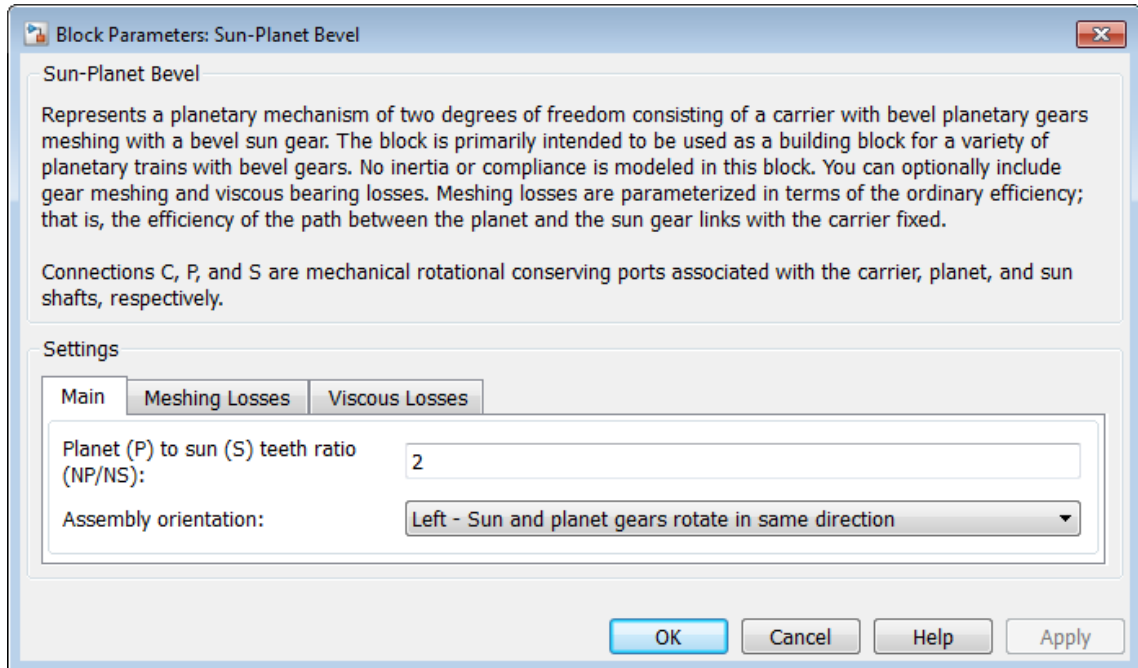
Main

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

Ordinary efficiency

Torque transfer efficiency η_{SP} for sun-planet gear wheel pair meshing.

Sun-carrier velocity threshold

Absolute angular velocity threshold ω_{th} above which full efficiency loss is applied, for sun-carrier velocity ω_S.

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

Viscous Losses

Sun-carrier viscous friction coefficient

Viscous friction coefficient μ_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 Gear Model

Ideal Gear Constraints and Gear Ratios

Sun-Planet Bevel imposes one kinematic and one geometric constraint on the three connected axes:

$$r_C \omega_C = r_S \omega_S \pm r_P \omega_P \ , \ r_C = r_S \pm r_P \ .$$

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:

$$g_{PS} \tau_S + \tau_P - \tau_{loss} = 0 \ ,$$

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 inertia is negligible. It does not impact gear dynamics.

- Gears are rigid. They do not deform.

- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

Examples

These SimDriveline example models use the Sun-Planet Bevel gear to create custom gear sets:

- sdl_limited_slip_differential

Sun-Planet Bevel

- sdl_custom_planetary_gear

See Also
Differential | Planet-Planet | Planetary Gear | Ring-Planet |
Sun-Planet

Related Examples
- "Represent and Transfer Driveline Motion and Torque"

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
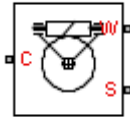
Description   The Sun-Planet Worm Gear block represents a two-degree-of-freedom planetary gear built from carrier, sun and planet gears. By type, the sun and planet gears are crossed helical spur gears arranged as a worm-gear transmission, in which the planet gear is a worm. Such transmissions are used 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
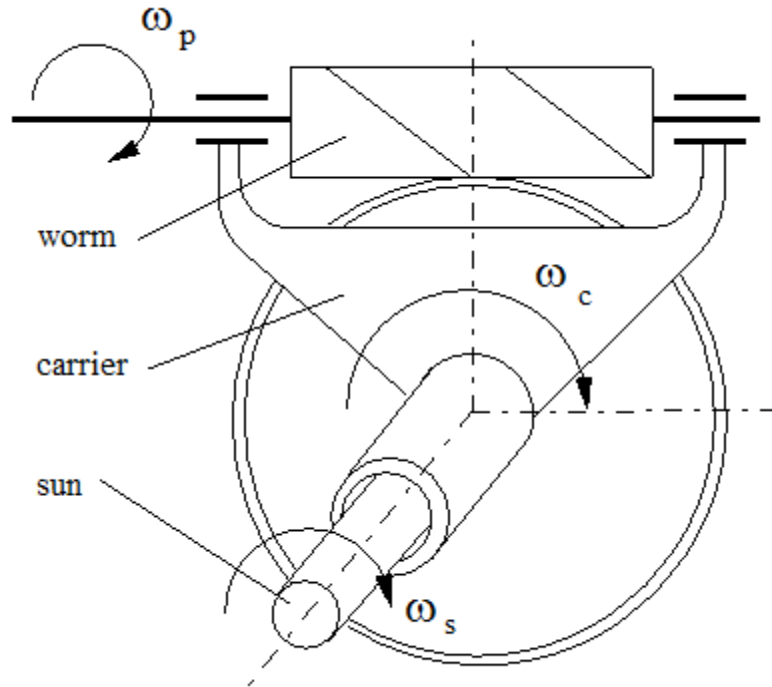
You specify a fixed gear ratio, which is determined as the ratio of the worm angular velocity to the sun gear angular velocity. You control the direction by setting the worm thread type, left-handed or right-handed. Rotation of the right-handed worm in positive direction causes the sun gear to rotate 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.

Dialog Box and Parameters

Block Parameters: Sun-Planet Worm Gear

Sun-Planet Worm Gear

Represents a two degree-of-freedom planetary mechanism consisting of a carrier with planetary gears meshing with a sun gear. The sun and planet gears are crossed helical spur gears arranged as a worm-gear transmission in which the planet gear is the worm. No inertia or compliance is modeled in this block.

You can optionally include gear meshing and viscous bearing losses. Meshing losses are parameterized in terms of the ordinary efficiency, which is the efficiency of the path between the worm (planet) and the sun gear with the carrier fixed. The efficiency depends on the direction of power flow. You can either specify the efficiency directly or derive it from geometrical parameters. Reverse efficiency (from sun to worm) can be negative, which results in self-locking.

Connections C, W, and S are the mechanical rotational conserving ports associated with the carrier, worm (planet), and sun shafts, respectively. If the Worm thread type is set to Right-handed, a positive worm rotation gives rise to a positive sun gear rotation.

Settings

| Main | Friction Losses | Viscous Losses |

Gear ratio: 25

Worm thread type: Right-handed

OK   Cancel   Help   Apply

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.

Sun-Planet Worm Gear

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

> 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.

Constant Efficiency

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 α 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

Worm-gear efficiency

> Efficiency η_{WG} of the power transfer from worm to gear.

Gear-worm efficiency

> Efficiency η_{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
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.
ω_S	Angular velocity of the sun gear
ω_P	Planet (that is, worm) angular velocity
ω_C	Carrier angular velocity
ω_{SC}	Angular velocity of the sun with respect to the carrier
a	Normal pressure angle
λ	Worm lead angle
L	Worm lead
d	Worm pitch diameter
τ_S	Torque applied to the sun shaft
τ_P	Torque applied to the planet shaft
τ_C	Torque applied to the carrier shaft

τ_{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.
τ_{instfr}	Instantaneous value of the friction torque added to the model to simulate friction losses
τ_{fr}	Steady-state value of the friction torque
k	Friction coefficient
η_{WG}	Efficiency for worm-gear power transfer
η_{GW}	Efficiency for gear-worm power transfer
ω_{th}	Absolute angular velocity threshold
μ_{SC}	Sun-carrier viscous friction coefficient
μ_{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:

$$\omega_S = \omega_P/R_{WG} + \omega_C \ .$$

The gear has two independent degrees of freedom. The gear pair is $(1,2) = (S,P)$.

The torque transfer is:

$$R_{WG}\tau_P + \tau_S - \tau_{loss} = 0 \ ,$$

$$\tau_C = -\ \tau_S,$$

with $\tau_{loss} = 0$ 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 μ_{SC} and μ_{WC}

The torque transfer for nonideal gear has the general form:

$$\tau_S = - R_{WG}(\tau_P - \mu_{WC}\omega_P) + \tau_{instfr} ,$$

$$\tau_{instfr} = \tau_{fr} \cdot \tanh(4\omega_{SC}/\omega_{th}) + \mu_{SC}\omega_{SC} .$$

The hyperbolic tangent regularizes the sign change in the friction torque when the sun gear velocity changes sign.

Condition	Friction Torque τ_{fr}
$\omega_{SC}\tau_S > 0$	$\|\tau_S\| \cdot (1 - \eta_{GW})$
$\omega_{SC}\tau_C \leq 0$	$\|\tau_S\| \cdot (1 - \eta_{WG})/\eta_{WG}$

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	η_{WG}	η_{WG}
Sun	η_{GW}	n/a	No loss
Carrier	η_{GW}	No loss	n/a

Geometric Surface Contact Friction

In the contact friction case, η_{WG} and η_{GW} are determined by:

- The worm-gear threading geometry, specified by lead angle λ and normal pressure angle a.

- The surface contact friction coefficient k.

    $\eta_{\mathrm{WG}} = (\cos a - k \tan\lambda)/(\cos a + k/\tan\lambda)$,

    $\eta_{\mathrm{GW}} = (\cos a - k/\tan\lambda)/(\cos a + k \tan a)$.

Constant Efficiencies

In the constant efficiency case, you specify η_{WG} and η_{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 η 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 1-284 section.

Sun-Planet Worm Gear

Limitations
- Gear inertia is negligible. It does not impact gear dynamics.

- Gears are rigid. They do not deform.

- Coulomb friction slows down simulation. 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

Related Examples
- "Represent and Transfer Driveline Motion and Torque"
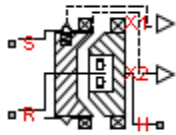
**Purpose**   Cone clutch, dog clutch, and translational detent assembled to provide smooth gear engagement

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.
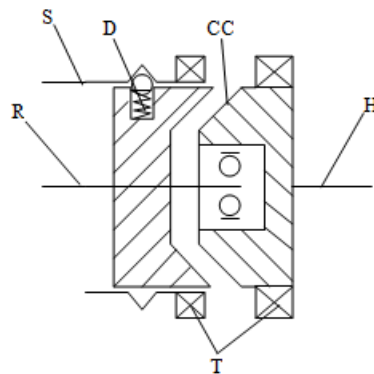
The schematic illustrates a synchronizer in the disengaged state. In this state, the ring (R) and hub (H) shafts can spin independently at different speeds. To synchronize ring and hub shaft speeds, the shift linkage (S) translates toward the hub shaft to engage the cone clutch. The friction surfaces 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
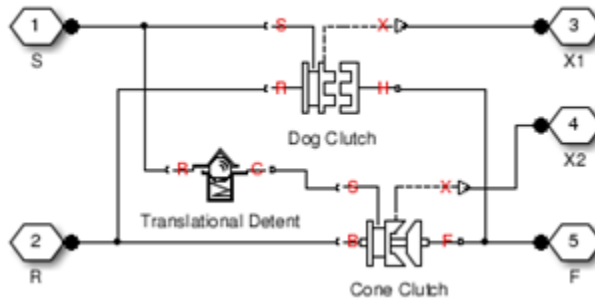
The model implements Dog Clutch, Translational Detent and a modified version of Cone Clutch blocks. Refer to each block reference page for more information on the corresponding block function. You can use a similar approach to model customized versions of the synchronizer. One example is the Gearbox (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

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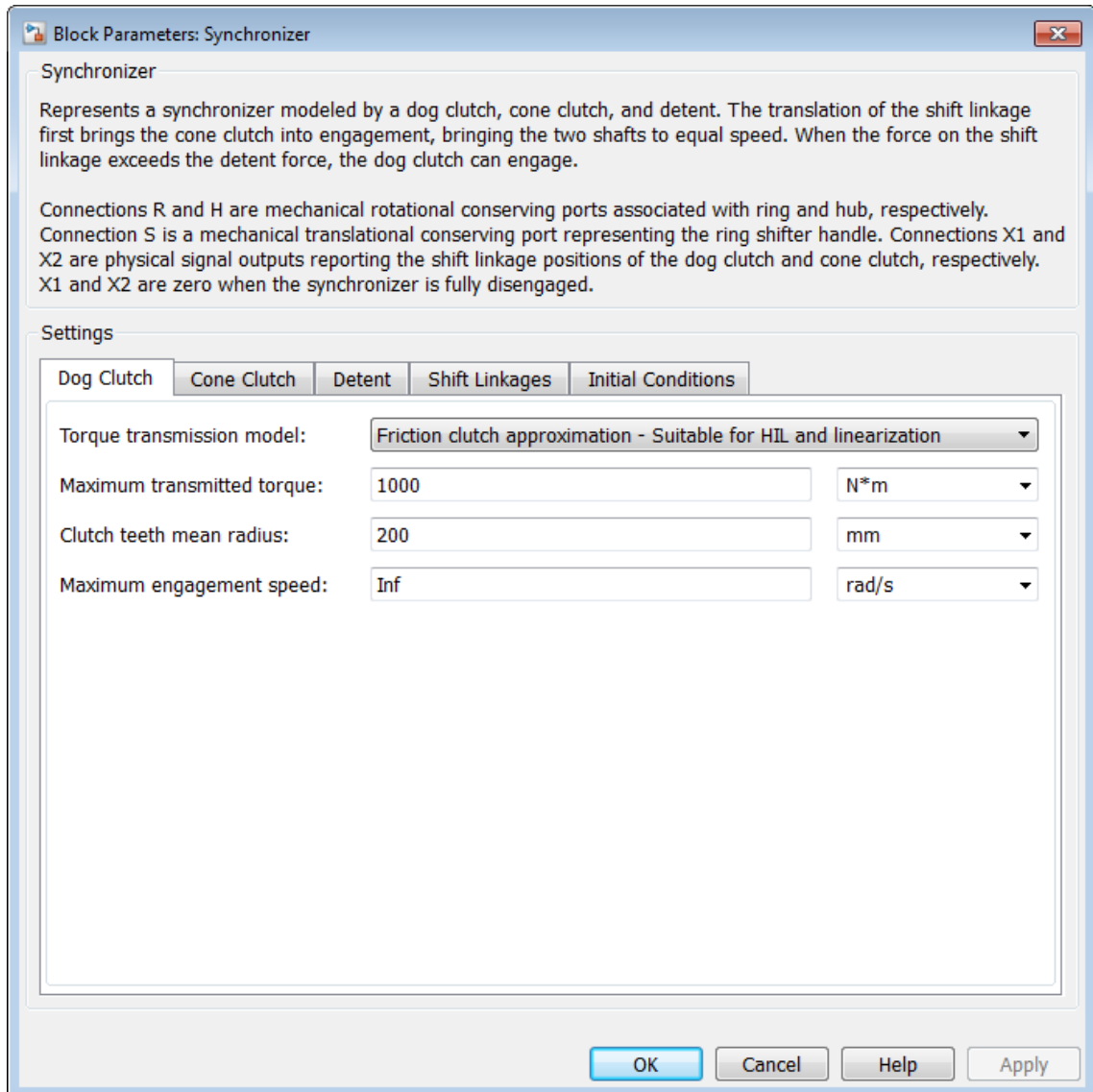
Connections R and H are mechanical rotational conserving ports associated with ring and hub, respectively. Connection S is a mechanical translational conserving port representing the ring shifter handle. Connections X1 and X2 are physical signal outputs reporting the shift linkage positions of the dog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

Settings

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

Torque transmission model:	Friction clutch approximation - Suitable for HIL and linearization ▼
Maximum transmitted torque:	1000	N*m ▼
Clutch teeth mean radius:	200	mm ▼
Maximum engagement speed:	Inf	rad/s ▼

OK   Cancel   Help   Apply

Synchronizer

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`.

Cone Clutch

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`.

Friction model

Specification method for the kinetic friction coefficient between cone clutch disks in slipping conditions. Options include:

- `Fixed kinetic friction coefficient` — Specify a single value for the kinetic friction coefficient. This value is constant across all clutch relative velocities. Selecting this option exposes an additional parameter, **Kinetic friction coefficient**.

- `Table lookup kinetic friction coefficient` — Specify a lookup table with the kinetic friction coefficient values as a function of the clutch slip velocity. Selecting this option exposes additional parameters for the lookup table values and interpolation/extrapolation methods.

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`.

Kinetic friction coefficient relative velocity vector

Vector with the clutch slip velocities at which to define the kinetic friction coefficient.

Kinetic friction coefficient vector

Vector with the kinetic friction coefficient values corresponding to the clutch slip velocities in **Kinetic friction coefficient relative velocity vector**.

Friction coefficient interpolation method

Method for calculating kinetic friction coefficients between lookup-table data points. Options include `Linear`, `Cubic`, and `Spline`.

Friction coefficient extrapolation method

Method for calculating kinetic friction coefficient values outside of the lookup-table data range. Options include `From last 2 points` and `From last point`.

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

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

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

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 **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**
	Initially disengaged	Parameter must be smaller than the sum of parameters **Ring-hub clearance when dog clutch**

Linkage Travel Direction	Dog Clutch State	Parameter Restriction
		disengaged and **Tooth overlap to engage**
Negative shift linkage displacement engages clutch	Initially engaged	Negative of the parameter must be greater than the sum of parameters **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**
	Initially disengaged	Negative of the parameter must be smaller than the sum of parameters **Ring-hub clearance when dog clutch disengaged** and **Tooth overlap to engage**

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:

Linkage Travel Direction	Dog Clutch State	Parameter Restriction
Positive shift linkage displacement engages clutch	Initially engaged	Parameter must be greater than the value of **Ring-hub clearance when cone clutch disengaged**
	Initially disengaged	Parameter must be smaller than the value of **Ring-hub clearance when cone clutch disengaged**
Negative shift linkage displacement engages clutch	Initially engaged	Negative of the parameter must be greater than the value of **Ring-hub clearance when dog cone disengaged**
	Initially disengaged	Negative of the parameter must be smaller than the value of **Ring-hub clearance when dog cone disengaged**

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 -π/N ≤ θ ≤ +π/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 -δ/2 ≤ θ ≤ +δ/2, where *δ* 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)

Purpose   Model a tire using kinetic and static friction coefficients

Library   Tires & Vehicles
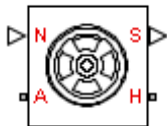
Description   The block represents a model of a tire that is parameterized with static and dynamic friction coefficients. When the static friction limit is exceeded, the tire loses traction and begins to slip. The kinetic friction coefficient determines the ability of the tire to transmit torque when 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.
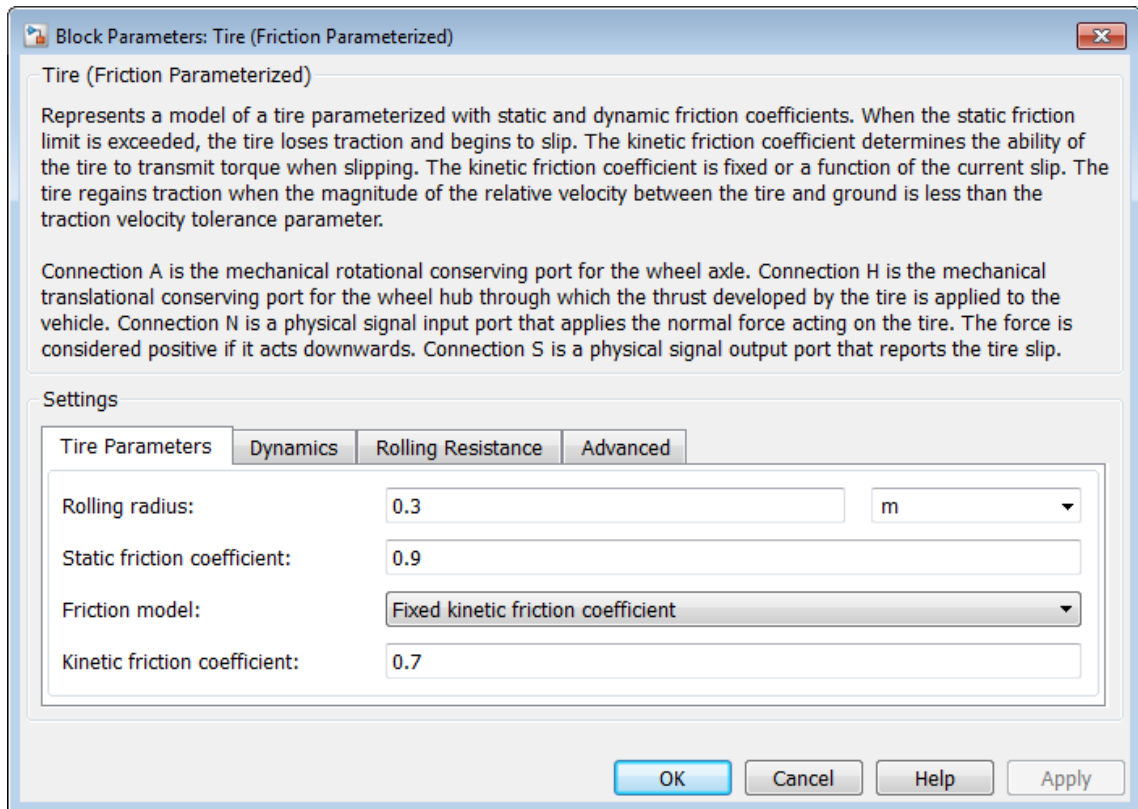
The traction or slip state model for the Tire (Friction Parameterized) block is based on that of the Fundamental Friction Clutch block. For more information on the state machine, see "Clutch State, Transition, and Variable Summary".

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

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.

Settings

| Tire Parameters | Dynamics | Rolling Resistance | Advanced |

Rolling radius: 0.3   m

Static friction coefficient: 0.9

Friction model: Fixed kinetic friction coefficient

Kinetic friction coefficient: 0.7

OK   Cancel   Help   Apply

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 (μ_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 (μ_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].

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.

The **Extrapolation method** parameter appears if you set the **Friction model** parameter to `Table lookup kinetic friction coefficient`.

Dynamics

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 `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^2`.

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`.

Rolling Resistance

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`.

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`.

Tire (Friction Parameterized)

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`.

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 1-229. 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^2/m^2`.

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.

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.

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

Tire (Friction Parameterized)

Port	Description
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 | Tire (Simple) | Tire (Magic Formula) | Tire-Road Interaction (Magic Formula)

Concepts   • "Clutch State, Transition, and Variable Summary"

Purpose   Tire with longitudinal dynamics and motion approximated by Magic
                   Formula

Library   Tires & Vehicles
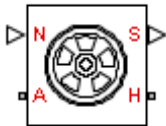
Description   The Tire (Magic Formula) block models the longitudinal dynamics of
                   a vehicle axle-wheel-tire combination, with road contact represented
                   by the Magic Formula and optional deformation compliance. The
                   convention for the vertical load 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 1-316.

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 κ, 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.

Tire (Magic Formula)

Dialog Box and Parameters
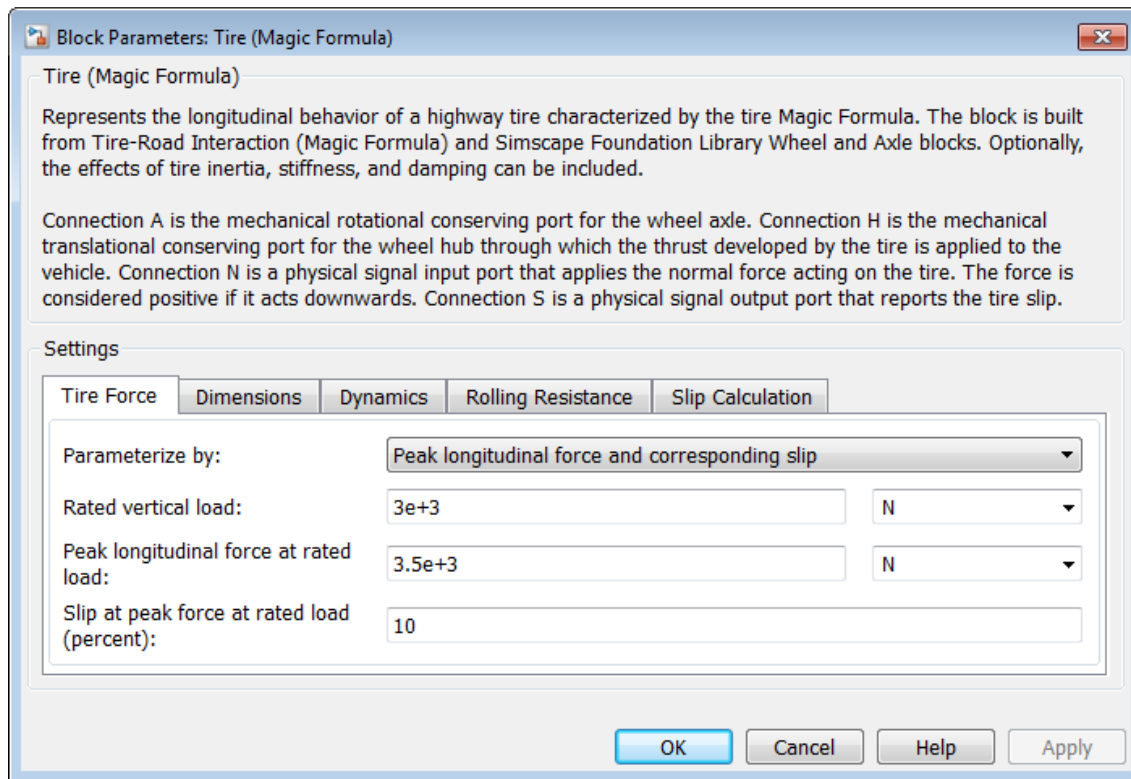
Tire Force

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.

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 κ'_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:

Tire (Magic Formula)

B	C	D	E
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.

Load-Dependent Magic Formula Coefficients

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`).

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 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 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 (kg*m^2).

Initial velocity

Initial angular velocity $\Omega(0)$ of the tire. The default is 0.

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

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.

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`.

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 1-229. 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`.

Tire (Magic Formula)

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^2/m^2`.

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`.

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 shows the forces acting on the tire. The table defines the tire model variables.

Tire Model Variables

Symbol	Description and Unit
r_w	Wheel radius
V_x	Wheel hub longitudinal velocity
u	Tire longitudinal deformation
Ω	Wheel angular velocity
Ω'	Contact point angular velocity = Ω 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 = κ if $u = 0$

Tire Model Variables (Continued)

Symbol	Description and Unit
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
$b_{Fx} = (\partial F_x/\partial \mathring{u})_0$	Tire longitudinal damping under deformation
I_w	Wheel-tire inertia; effective mass $= I_w/r_w^2$
τ_{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 Ω' from the wheel Ω and requires, instead of the wheel slip, the *contact point* or *contact patch slip κ'*. 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
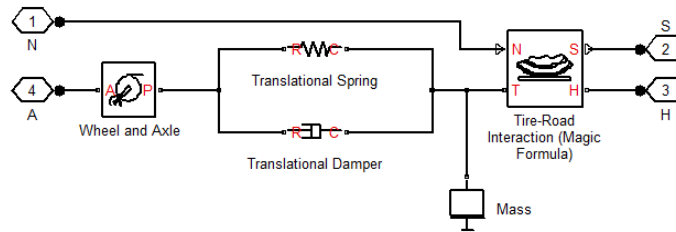
The full tire model is equivalent to this Simscape-SimDriveline component diagram. It simulates both transient and steady-state behavior and correctly represents starting from, and coming to, a stop. The Translational Spring and Translational Damper are equivalent to the tire stiffness C_{Fx} and damping b_{Fx}. Tire-Road Interaction (Magic Formula) models the longitudinal force F_x on the tire as a function of F_z and κ' using the Magic Formula, with κ' 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).

Tire (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

Related Examples

- "Complete Vehicle Model"

Purpose   Tire tread in contact with road surface approximated by Magic Formula

Library   Tires & Vehicles/Tire Subcomponents
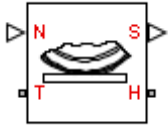
Description   The Tire-Road Interaction (Magic Formula) block models the longitudinal tire-road contact force using the Magic Formula. Use this block as a component in tire models.

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 1-325.

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 κ, 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)

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.

Peak Force and Corresponding Slip

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 κ_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.

Constant Magic Formula Coefficients

Magic formula coefficients

Dimensionless Magic Formula *B*, *C*, *D*, and *E* coefficients. The defaults are:

B	C	D	E
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.

Load-Dependent Magic Formula Coefficients

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
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 Contact Variables

Symbol	Description and Unit
Ω	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

Tire-Road Interaction (Magic Formula)

Tire-Road Contact Variables (Continued)

Symbol	Description and Unit
$\kappa = V_{sx}/\lvert V_x \rvert$	Wheel slip
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 κ

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

$$df_z = (F_z - F_{z0})/F_z \ ,$$

$$\kappa_x = \kappa + S_{Hx} \ ,$$

$$C_x = \text{p\_Cx1} \ ,$$

$$D_x = \mu_x \cdot F_z \ ,$$

$$\mu_x = \text{p\_Dx1} + \text{p\_Dx2} \cdot df_z \ ,$$

$$E_x = (\text{p\_Ex1} + \text{p\_Ex2} \cdot df_z + \text{p\_Ex3} \cdot df_z^2)[1 - \text{p\_Ex4} \cdot \text{sgn}(\kappa_x)] \ ,$$

$$K_{x\kappa} = F_z \cdot (\text{p\_Kx1} + \text{p\_Kx2} \cdot df_z) \cdot \exp(\text{p\_Kx3} \cdot df_z) \ ,$$

$$B_x = K_{x\kappa}/(C_x D_x + \varepsilon_x) \ ,$$

$$S_{Hx} = \text{p\_Hx1} + \text{p\_Hx2} \cdot df_z \ ,$$

$$S_{Vx} = F_z \cdot (\text{p\_Vx1} + \text{p\_Vx2} \cdot df_z) \ .$$

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. μ_x is the longitudinal load-dependent friction coefficient. ε_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 κ_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) |

Related Examples

- "Complete Vehicle Model"

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.

Tire (Simple)
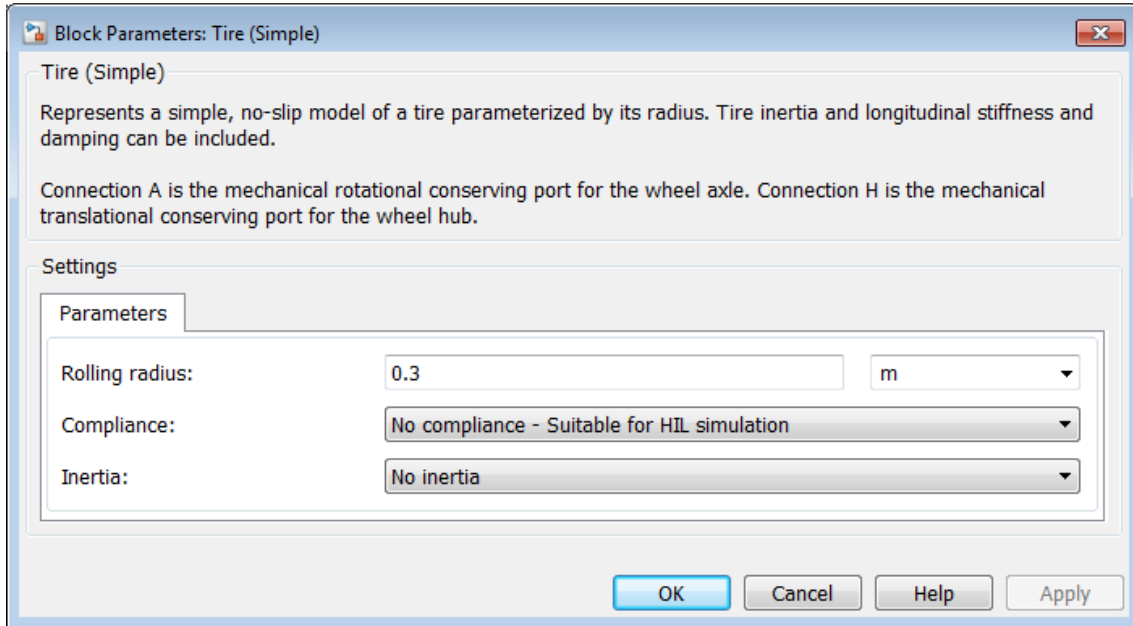
Rolling radius

>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^2`.

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`.

Tire (Simple)

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)
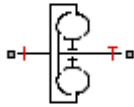
Related Examples
- "Complete Vehicle Model"

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 1-336.

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

Block Parameters: Torque Converter

Torque Converter

Represents a three-part torque converter consisting of impeller, turbine, and stator. The model is only valid for positive impeller speeds, but does support simulation of both driving (power flowing from impeller to turbine) and coasting (power flowing from turbine to impeller) modes.

Connections I and T are mechanical rotational conserving ports associated with the impeller and turbine, respectively. The speed ratio is (Output angular speed)/(Input angular speed). The torque ratio is (Output torque)/(Input torque). The capacity factor can either be parameterized as (Input speed)/sqrt(Input torque) or as (Input torque)/(Input speed)^2. The torque is positive if it flows from port I to port T.

Settings

| Torque Characteristics | Dynamics |

Speed ratio vector:	[0 0.50 0.60 0.70 0.80 0.87 0.92 0.94 0.96 0.97]
Torque ratio vector:	2.2320 1.5462 1.4058 1.2746 1.1528 1.0732 1.0192 0.9983 0.9983 0.9983]
Capacity factor parameterization:	Ratio of speed to square root of impeller torque ▾
Capacity factor reference speed:	Always impeller speed ▾
Capacity factor vector:	4.6163 16.2675 19.3503 22.1046 29.9986 50.00] (rad/s)/(N*m)^0.5 ▾
Interpolation method:	Linear ▾
Extrapolation method:	From last 2 points ▾

OK   Cancel   Help   Apply

Torque Characteristics

Speed ratio vector

Vector of values of the independent variable, the dimensionless speed ratio R_ω. 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_τ. Each torque ratio value corresponds to a speed ratio value.

Capacity factor parameterization

Definition of the capacity factor, either K (ratio of impeller speed ω_I to square root of impeller torque τ_I) or K^* (ratio of τ_I to ω^2_I). The default is K.

Capacity factor reference speed

Choice of speed in the capacity factor definition, depending on speed ratio R_ω. Select either:

- Impeller speed ω_I for all values of R_ω.

- Impeller speed ω_I for $R_\omega < 1$, and turbine speed ω_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/√(newton-meters) (`rad/s/(N*m)^0.5`).

- If you choose the alternative capacity factor definition K^*, the default units are newton-meters/(radians/second)^2 (`N*m/(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`.

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 τ_T to impeller torque τ_I at the start of the simulation.

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 $R_\omega \to 0$ or ∞, and shrinks as $R_\omega \to 1$. Because R_ω 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_ω is the turbine angular speed divided by the impeller angular speed:

    $R_\omega = \omega_T/\omega_I$.

- The torque ratio R_τ 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:

    $K^* = \tau_I \,/\, \omega^2{}_I$.

- The capacity factor reference speed for $R_\omega > 1$ is ω_I by default. That is, the input speed ω_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 ω_I by the output speed ω_T in this defining ratio.

The two dependent variables, R_τ and K, are functions of the independent variable R_ω. 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_ω falls outside the specified range during simulation, the torque ratio and capacity ratio functions are extrapolated. The extrapolation must satisfy the requirement that τ_{out} be zero when R_ω is 1. To meet this requirement, the block uses a mathematical extension that starts with the last R_τ value below 1, which is taken to be the locking point. τ_I is extrapolated with:

$\tau_I = \mathrm{sgn}(1 - \omega_T/\omega_I) \cdot [\omega_I \,/\, K(\mathrm{L})]^2 \cdot \sqrt{[\,|\,1 - R_\omega\,|\,/(1 - R_\omega(\mathrm{L}))]}$.

R_ω(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 τ_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_τ 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:

$$\tau_I = \text{sgn}(1 - \omega_T/\omega_I) \cdot [\omega_I / K]^2 \, , \, \tau_T = \tau_I \cdot R_\tau \, ,$$

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 τ_T and τ_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 τ_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 Converter

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

Related Examples
- "Complete Vehicle Model"

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 1-345.

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.

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
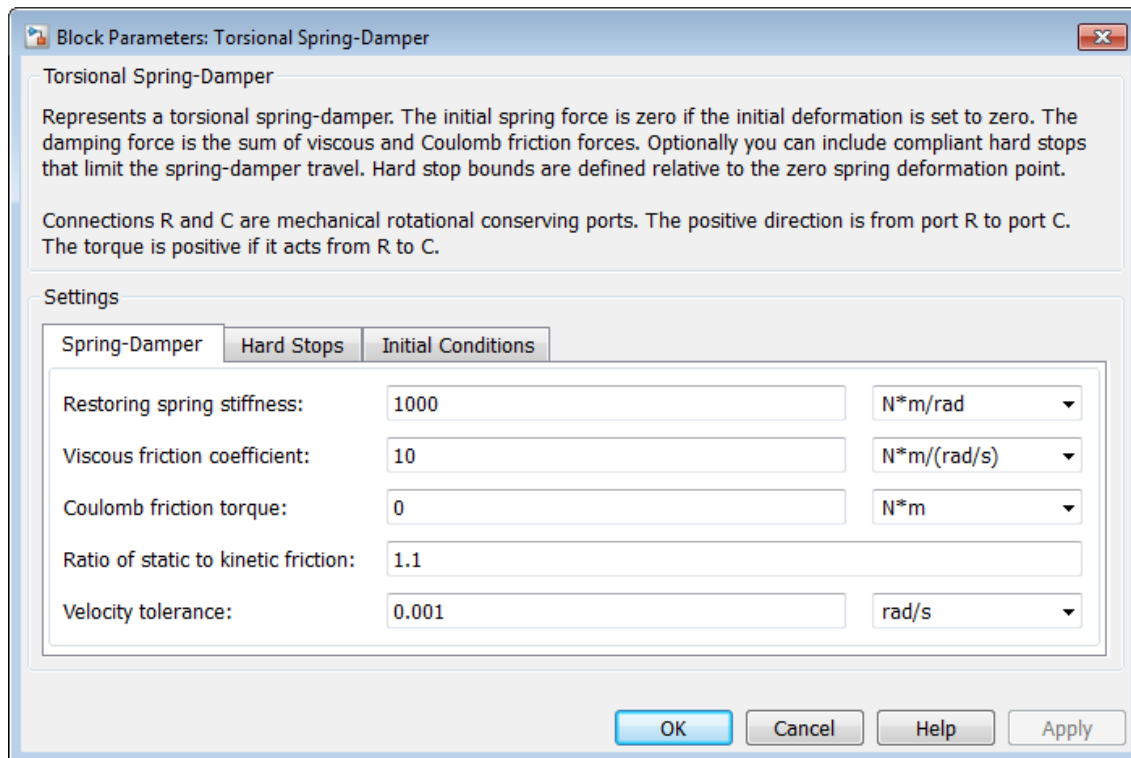
Spring-Damper

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 (N*m/rad).

Viscous friction coefficient

Torsional damping μ 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) (N*m/(rad/s)).

Coulomb friction torque

Constant kinetic friction torque τ_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 (N*m).

Ratio of static to kinetic friction

Constant ratio R of static Coulomb friction torque τ_S to kinetic Coulomb friction torque τ_K acting between connected driveshafts. Must be greater than one. The default is 1.1.

Velocity tolerance

Minimum relative angular speed ω_{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 (rad/s).

Hard Stops

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.

- Compliant hard stops — Model friction geometry in terms of annulus dimensions. If you select this option, the panel changes from its default.

Torsional Spring-Damper

Compliant hard stops

Upper bound

Upper hard stop angular displacement δ_+ from the zero-torque reference angle $\varphi = 0$. Must be greater than δ_-. The default is 10.

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

Lower bound

Lower hard stop angular displacement δ_- from the zero spring force reference angle $\varphi = 0$. Must be less than δ_+. 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 φ 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 μ_{HS} applied if the relative angle φ 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

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 τ 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
φ	Relative angle between ring and hub	Relative angular position of ring and hub
ω	Relative angular velocity	$\omega = \omega_R - \omega_C$
k	Torsional stiffness	*See the following model*
μ	Torsional damping	*See the following model*
δ_+, δ_-	Upper and lower hard stop angular displacements	*See the following model*
k_{HS}	Contact stiffness applied in hard stop regions	*See the following model*
μ_{HS}	Contact damping applied in hard stop regions	*See the following model*
τ_K	Kinetic friction	Constant sliding Coulomb friction
τ_S	Static friction	Constant locking Coulomb friction

Torsional Spring-Damper Variables (Continued)

Symbol	Definition	Significance
R	τ_S/τ_K	Ratio of static to kinetic Coulomb friction
ω_{Tol}	Maximum relative speed for clutch locking	*See the following model*

Stiffness and Damping

The stiff-damping torque is a simple linear spring-damping:

$$\tau_{\text{SD}} = -k\varphi - \mu\omega \ .$$

Hard Stops

The hard stop torque is applied if φ 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 τ_K if ω is nonzero (unlocked), and a constant τ_S if ω 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 τ_S.

If the clutch locks, ω is reset to zero. If the torque across the torsional spring-damper exceeds τ_S, the driveshafts unlock from one another, and ω becomes nonzero.

See Also   Dog Clutch | Loaded-Contact Rotational Friction | Rotational Damper | Rotational Hard Stop | Rotational Spring

Related Examples

- "Gear Coupling Control with Clutches"

Translational Detent

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.
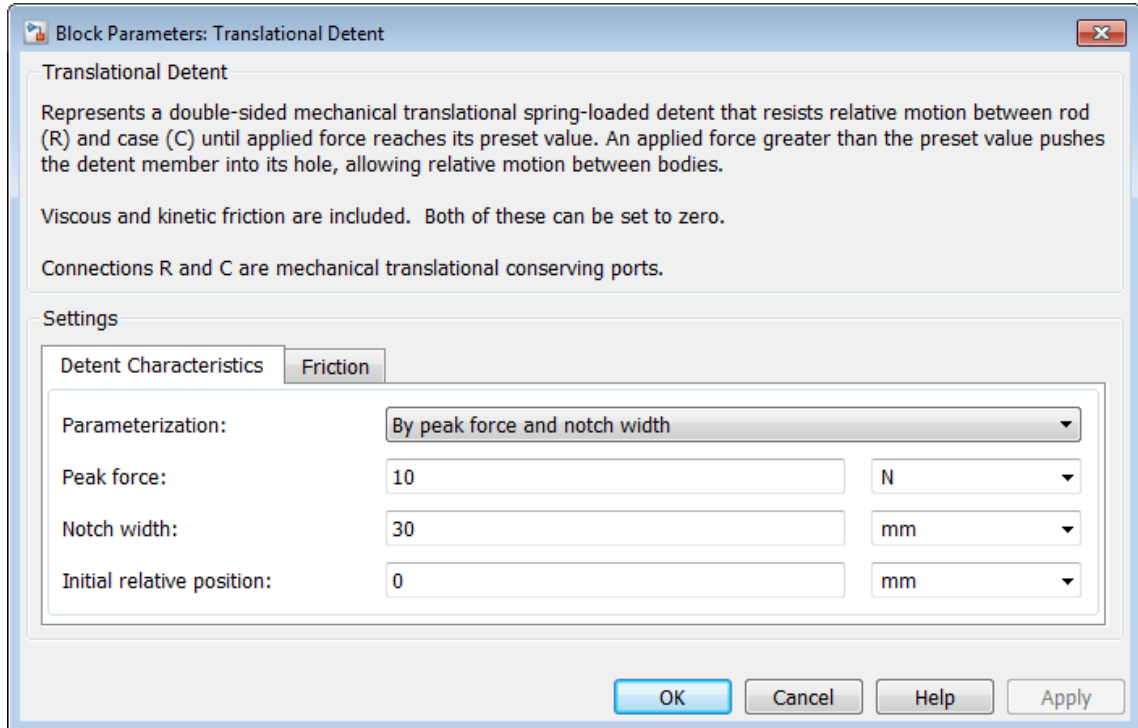
Dialog Box and Parameters

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.

Translational Detent

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

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.

Geometric and Spring Parameters

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 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.

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

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.

Translational Detent

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.
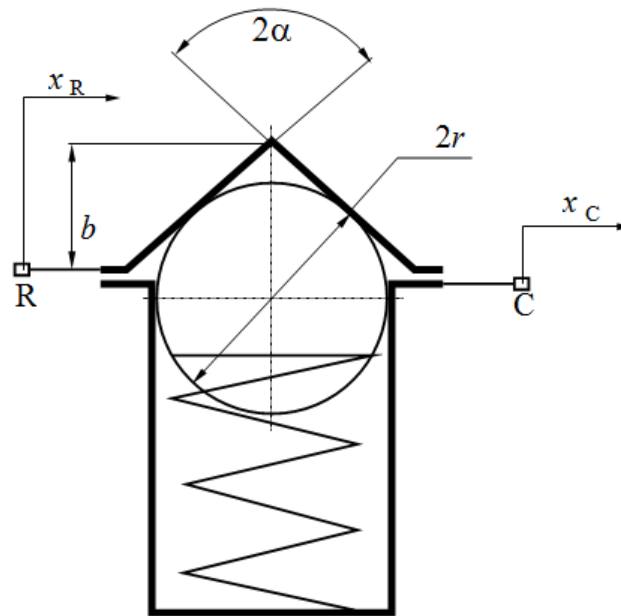
b	Notch depth
r	Ball radius
a	Notch half angle

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.

Geometric and Spring Parameters

The geometric and spring parameterization uses four regions to define the shear force. The following figure displays these regions.
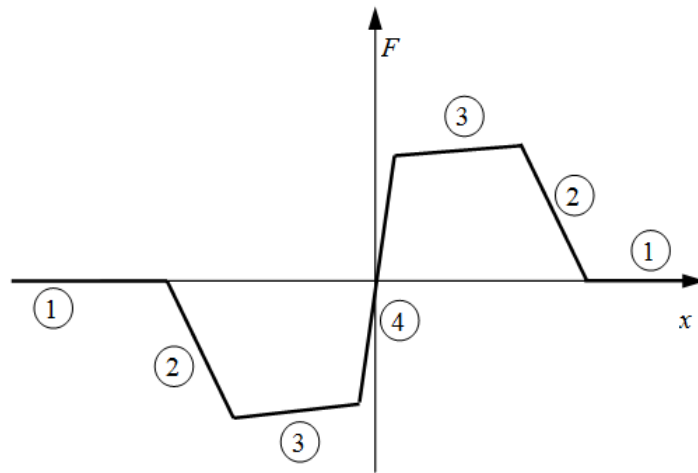
Because the ball is spherical and the notch is symmetric, the horizontal force versus displacement function is symmetric about the origin. (See "Peak Force and Notch Width" on page 1-356.)

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.

Region	Ball Position and Contact Angle	Shear Force
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.
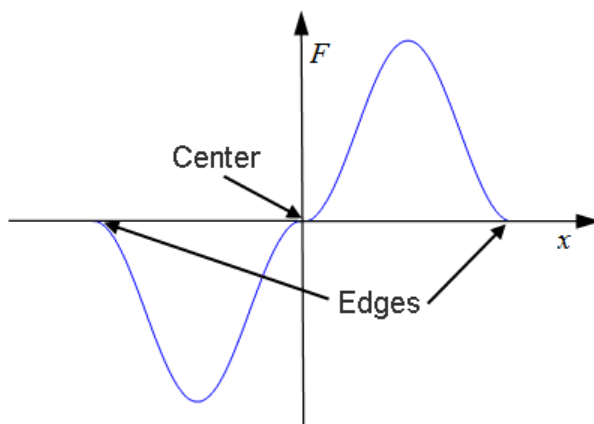
Lookup Table

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.

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

Related Examples   • "Clutches, Clutch-Like Elements, and Coulomb Friction"

Translational Power Sensor

Purpose   Mechanical sensor used to measure average or instantaneous translational power

Library   Sensors
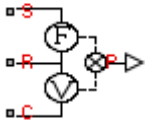
Description The block represents a sensor that you can use to measure the translational power change due to one or more driveline components connected between ports R and C. The power output can be an instantaneous or average value. Two averaging options are available: period and vibration averaging.

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**
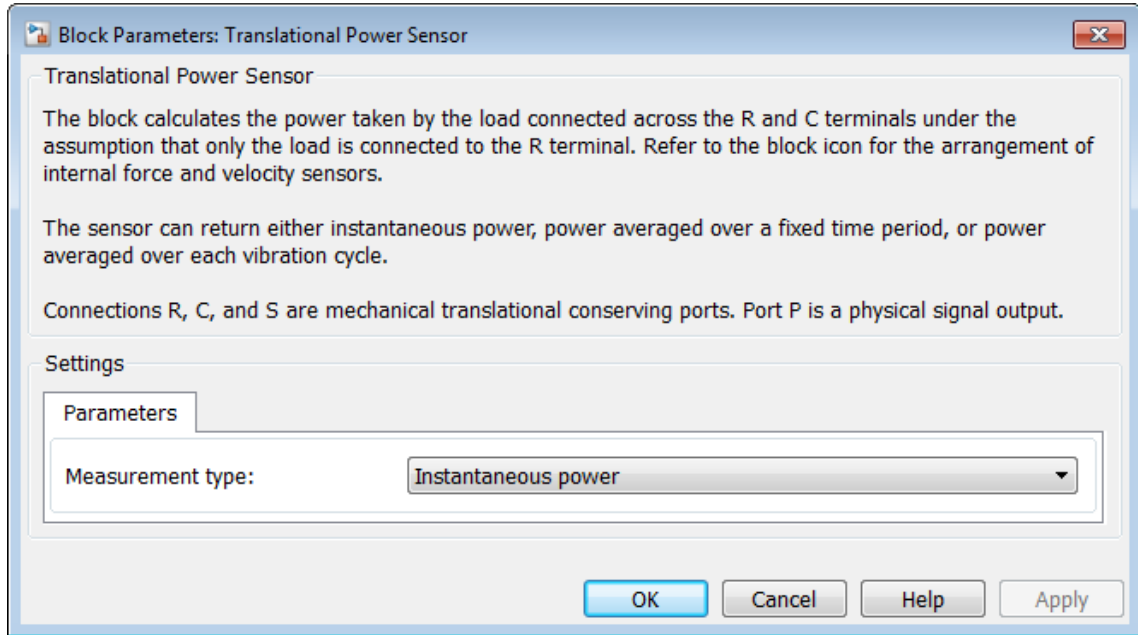
Measurement Type

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.

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
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

Purpose   Clutch transmitting torque in one direction only

Library   Clutches
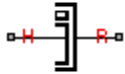
Description   The Unidirectional Clutch block models an ideal one-way clutch based on a coupled ring-hub. It transmits torque when the relative velocity of hub and ring becomes positive or negative, depending on how you set the transmission direction. For model details, see "Unidirectional Clutch Model" on page 1-363.

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.

Unidirectional Clutch

Block Parameters: Unidirectional Clutch

Unidirectional Clutch

Represents a clutch that transmits torque when the relative velocity between the hub (H) and the ring (R) becomes either positive or negative, depending on the direction set by the Transmission direction parameter.

Connections H and R are mechanical rotational conserving ports associated with the hub and ring, respectively.

Settings

Parameters

Transmission direction:	From hub to ring for positive rotation
Initial state:	Unlocked

OK   Cancel   Help   Apply

Transmission direction

From the drop-down list, select the relative rotation direction (sign of) 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.
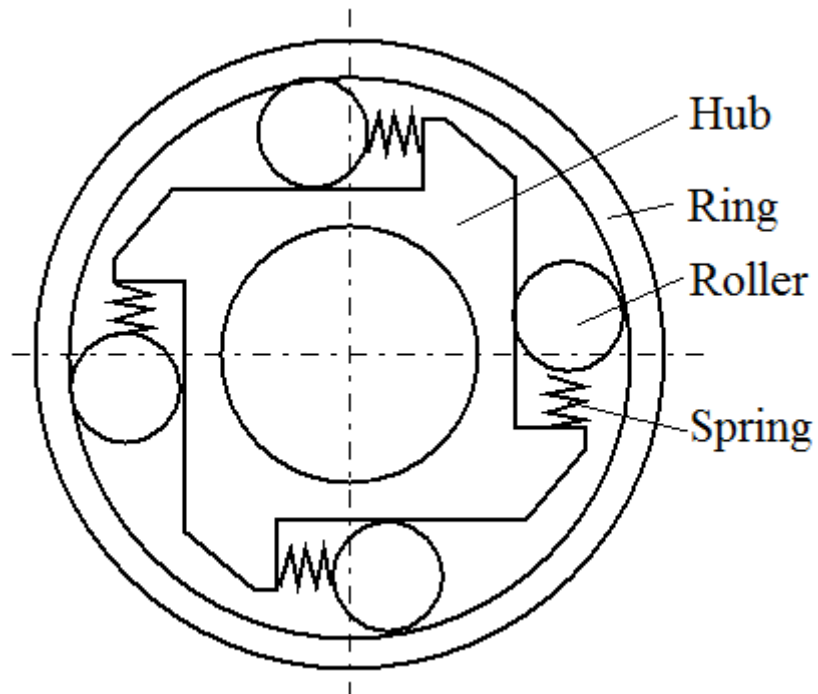
Clutch Variable

Symbol	Definition	Significance
ω	Relative angular velocity (slip)	$\omega_H - \omega_R$

Unidirectional Clutch

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 ω 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 ω 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; ω 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 a clutch schematic, see "Clutch Geometry, Variable, and State Summary" on page 1-363.

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
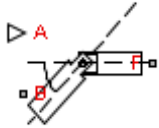
Related Examples   • "Gear Coupling Control with Clutches"

Universal Joint

**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.
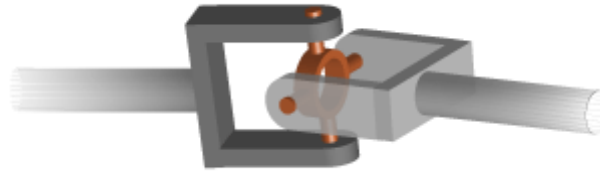
The ratio of the shaft angular velocities depends on two parameters: the intersection angle between the two shafts and the rotation angle of the base shaft. A physical signal input provides the intersection angle while a dialog box parameter provides the initial base shaft angle. These two angles fix 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:

- ω_F is the angular velocity of the follower shaft about its length axis.

- ω_B is the angular velocity of the base shaft about its length axis.

- θ_B is the rotation angle of the base shaft about its length axis.

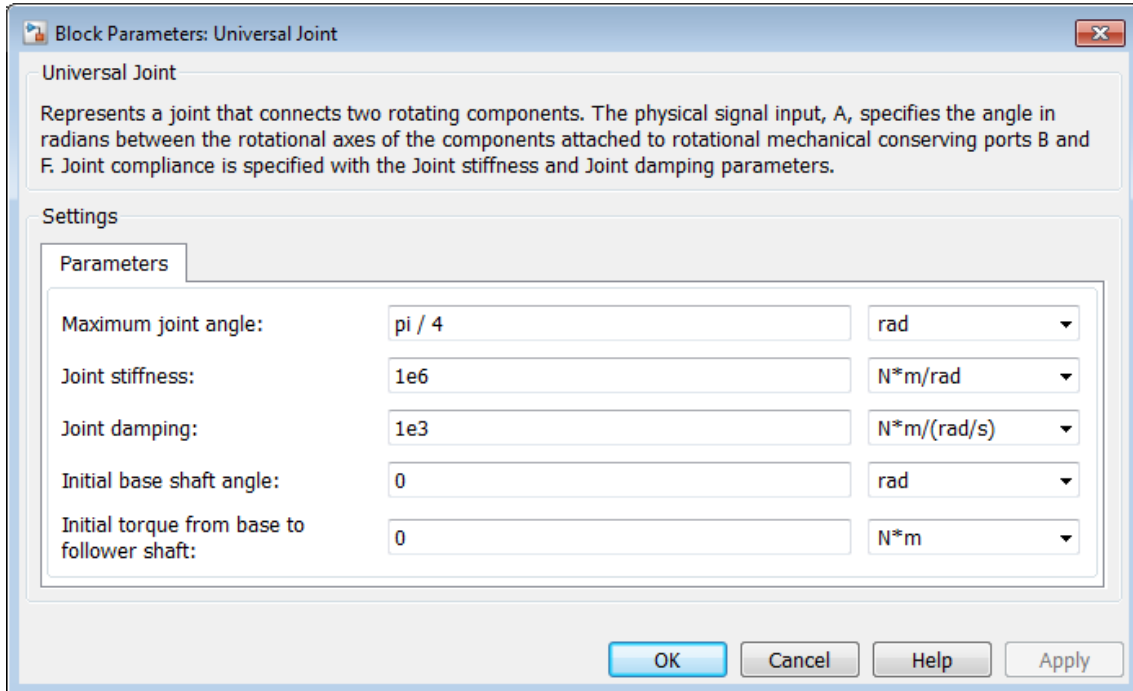- A is the intersection angle between base and follower shafts about the base shaft pin.

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° about their length axes (dashed line segments).
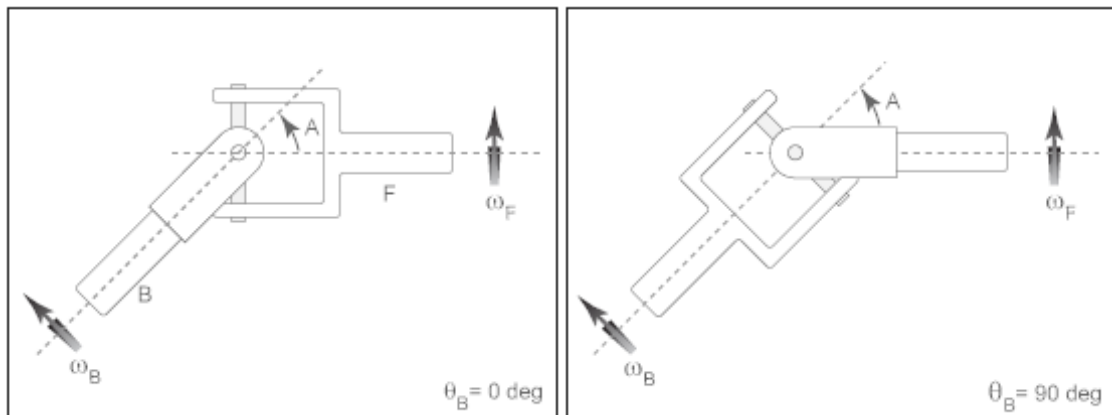
In the figure, the intersection angle is the angle between the two shafts about the base shaft's pin. The absolute value of this angle must fall in the range 0 A<Maximum intersection angle. The base shaft angle is the angle of the base shaft about its length axis. The base shaft angle is also the time-integral of the base shaft angular velocity, ω_B.

Universal Joint

Dialog Box and Parameters

Maximum joint angle

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. From the drop-down list, select a physical unit. The default value is pi/4 rad.

Joint stiffness

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 N*m/rad.

Joint damping

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 N*m/(rad/s).

Initial base shaft angle

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 rad.

Initial torque from base to follower shaft

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 N*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

Variable Ratio Transmission

Purpose   Dynamic gearbox with variable and controllable gear ratio, transmission compliance, and friction losses

Library   Couplings & Drives
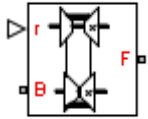
Description The Variable Ratio Transmission block represents a gearbox that dynamically transfers motion and torque between the two connected driveshaft axes, base and follower.

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, ω_F and ω_B have the same sign. If they rotate in opposite directions, ω_F and ω_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 1-373.

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
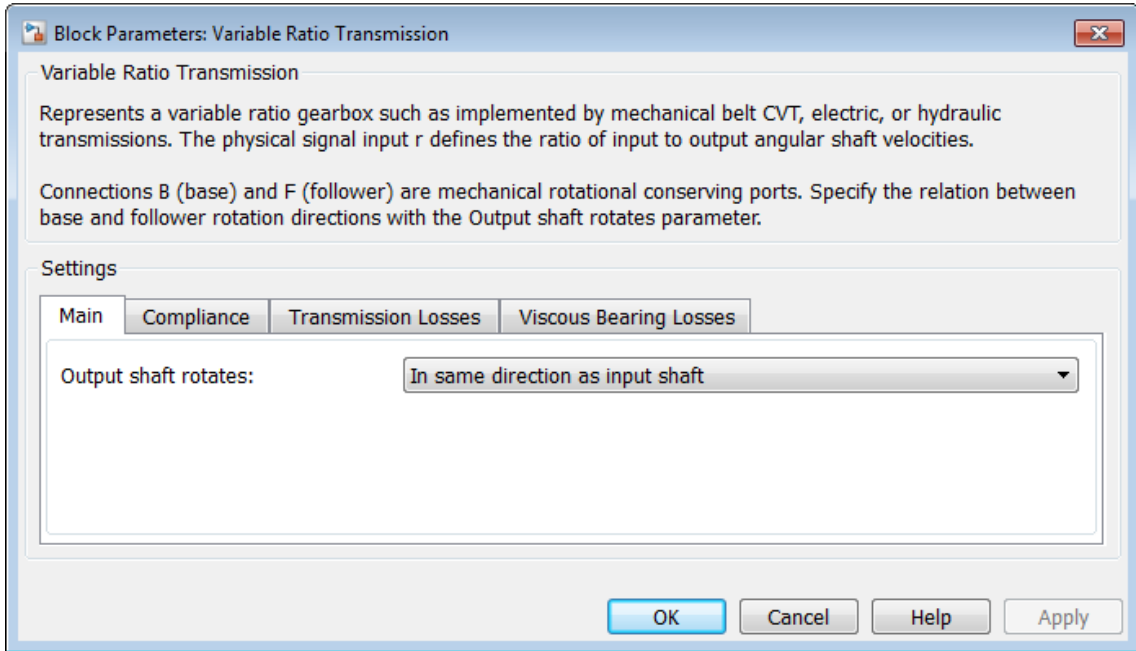
Main

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`.

Variable Ratio Transmission

Compliance

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

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 η satisfying $0 < \eta \leq 1$. If you select this option, the panel changes from its default.

Constant Efficiency

Efficiency

> Effective torque transfer efficiency η between the base and follower driveshafts.

Follower angular velocity threshold

> Absolute angular velocity threshold ω_{th} above which full efficiency loss is applied, for follower velocity ω_F.

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

Viscous Bearing Losses

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 φ 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 φ that dynamically responds through base compliance parameters k_p and k_v:

Variable Ratio Transmission

$$d\varphi/dt = \pm g_{FB}(t) \cdot \omega_F - \omega_B \ ,$$

$$\tau_B = -k_p \varphi - k_v d\varphi/dt \ ,$$

$$\pm g_{FB}(t) \, \tau_B + \tau_F - \tau_{loss} = 0 \ .$$

$\tau_{loss} = 0$ 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, $\tau_{loss} \neq 0$. 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 η.

- Viscous coupling of driveshafts with bearings, parametrized by viscous friction coefficients μ.

$$\tau_{loss} = \tau_{Coul} \cdot \tanh(4\omega_{out}/\omega_{th}) + \mu_B \omega_B + \mu_F \omega_F \ , \ \tau_{Coul} = |\tau_F| \cdot (1 - \eta) \ .$$

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 ω_{out}
Forward	$\omega_B \tau_B > \omega_F \tau_F$	Follower, ω_F
Reverse	$\omega_B \tau_B < \omega_F \tau_F$	Base, ω_B

Meshing Efficiency

The friction loss represented by efficiency η is fully applied only if the absolute value of the follower angular velocity ω_F is greater than a velocity threshold ω_{th}.

If this absolute velocity is less than ω_{th}, the actual efficiency is automatically regularized to one at zero velocity.

Example   The sdl_variable_gear example model provides a basic example of a simple gear.

See Also   Simple Gear | Torque Converter

Related Examples   • "Gear Coupling Control with Clutches"

Variable Rotational Damper

Purpose   Rotational damper with variable damping coefficient

Library   Couplings & Drives/Springs & Dampers
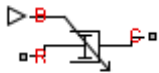
Description   The block represents a rotational viscous damper with a variable damping coefficient. A physical signal input port provides the magnitude of the damping coefficient. The magnitude of the damping torque is equal to the product of the physical signal input and the relative angular velocity 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:

$$T = \begin{cases} B \cdot \omega & B \geq B_{\min} \\ B_{\min} \cdot \omega & B < B_{\min} \end{cases},$$

The parameters are:

- T — Torque transmitted through damper
- B — Viscous damping coefficient
- ω — Relative angular velocity measured between the two rotational conserving ports according to $\omega = \omega_R - \omega_C$
- ω_R — Angular velocity of rotational conserving port R
- ω_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.

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
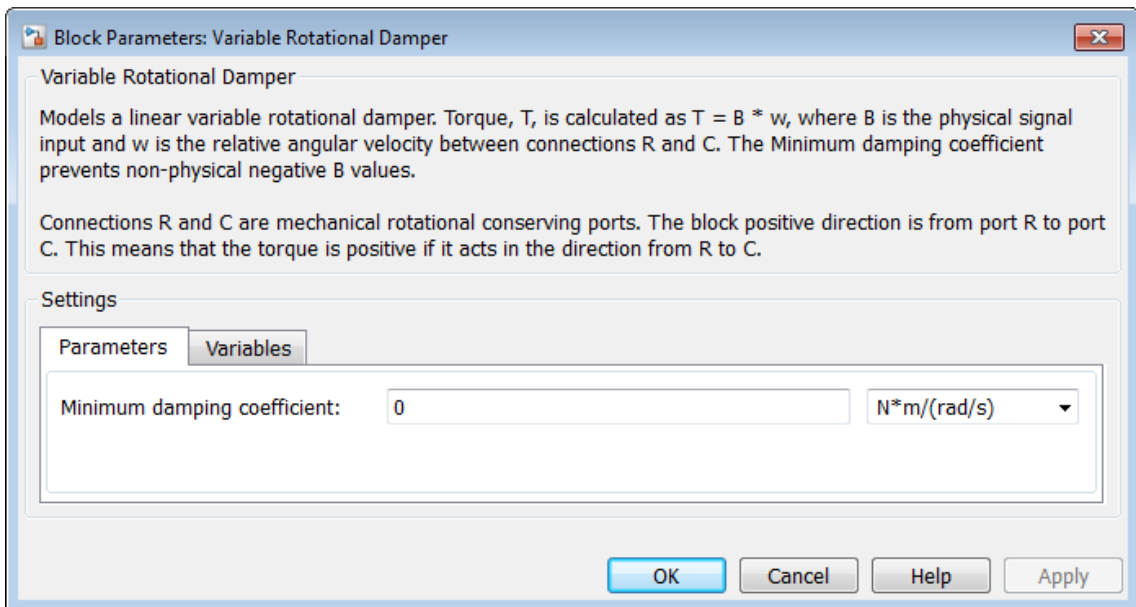
Parameters

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

Variables

Specify the desired initial values of the block dynamical variables. These variables guide model assembly at simulation time zero. When two or more variables conflict with each other, the priority setting determines which of the variables to satisfy. For more information, see "Set Priority and Initial Target for Block Variables".

Variables that you can specify include:

- t — Spring torque
- w — Relative rotational velocity

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

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:

$$T = \begin{cases} K \cdot \theta & K \geq K_{\min} \\ K_{\min} \cdot \theta & K < K_{\min} \end{cases},$$

The parameters are:

- T — Torque transmitted through the spring

- K — Spring rate (stiffness)

- K_{min} — Minimum allowable spring rate (stiffness)

- θ — Relative angular displacement measured between the two rotational conserving ports according to $\theta = \theta_{init} + \theta_R - \theta_C$

- θ_{init} — Initial angular deformation between the two rotational conserving ports

- θ_R — Absolute angular displacement of rotational conserving port R

- θ_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
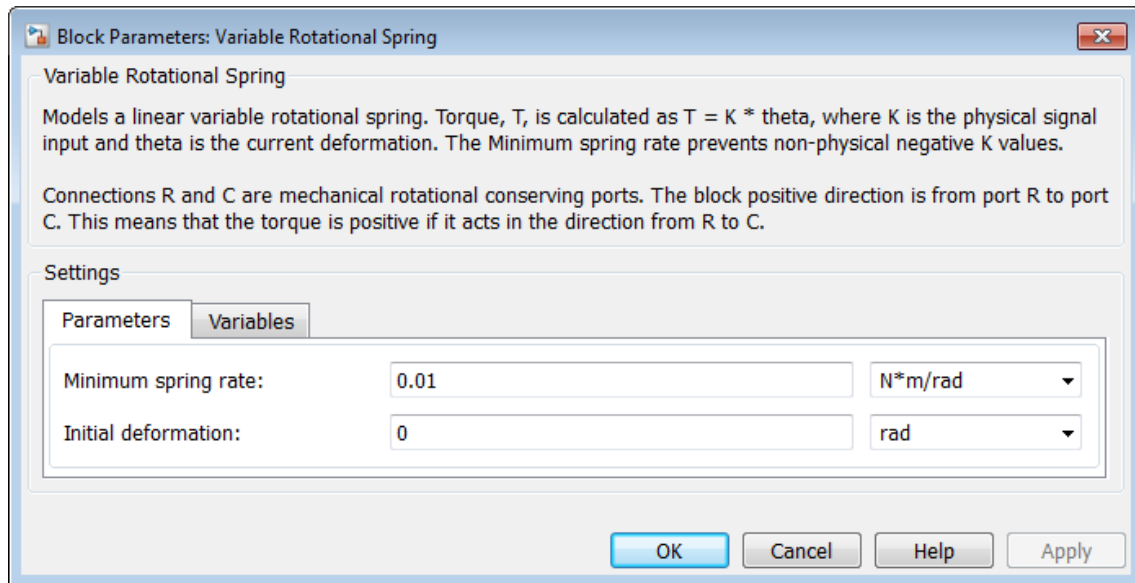
Parameters

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`).

Initial Deformation

Initial angular displacement between rotational conserving ports R and C. The initial value is 0. The initial unit is rad.

Variables

Specify the desired initial values of the block dynamical variables. These variables guide model assembly at simulation time zero. When two or more variables conflict with each other, the priority setting determines which of the variables to satisfy. For more information, see "Set Priority and Initial Target for Block Variables".

Variables that you can specify include:

- t — Spring torque
- w — Deformation rate
- theta — Rotational deformation

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
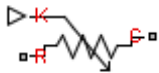
The block represents a translational viscous damper with variable damping coefficient. A physical signal input port provides the magnitude of the translational damping coefficient. The magnitude of the damping force is equal to the product of the physical signal input and the relative linear velocity between the two translational conserving ports. A minimum damping coefficient prevents non-physical negative damping values.

The translational damping force satisfies the following expression:

$$F = \begin{cases} B.v, & B \geq B_{\min} \\ B_{\min} \cdot v, & B < B_{\min} \end{cases},$$

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.

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
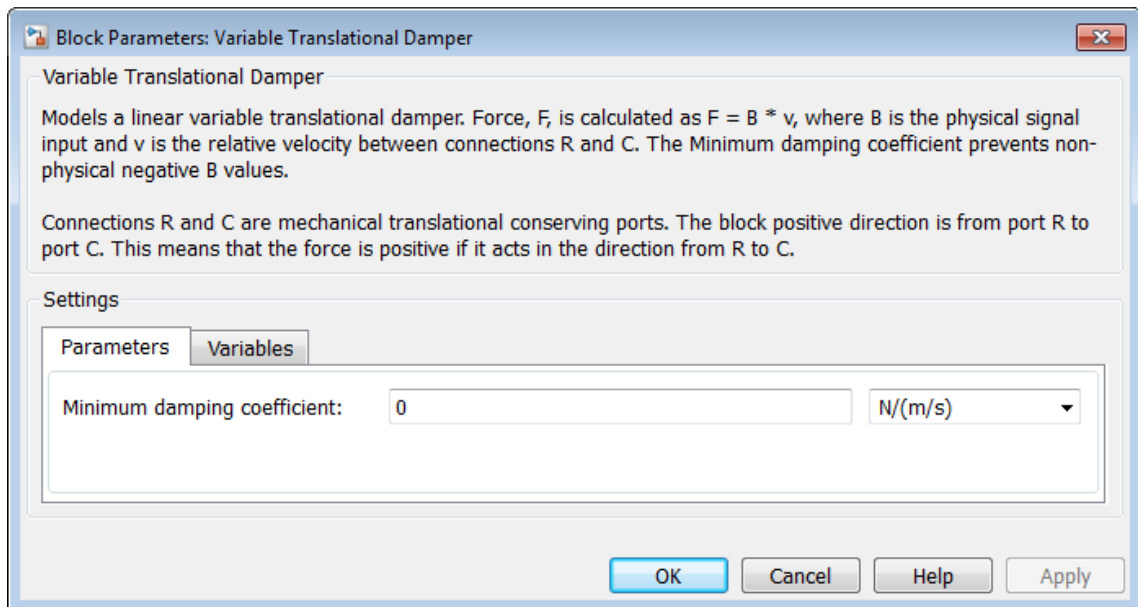
Parameters

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

Variables

Specify the desired initial values of the block dynamical variables. These variables guide model assembly at simulation time zero. When two or more variables conflict with each other, the priority setting determines which of the variables to satisfy. For more information, see "Set Priority and Initial Target for Block Variables".

Variables that you can specify include:

- f — Damper force
- v — Relative translational velocity

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
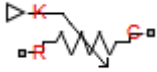
Purpose   Translational spring with variable spring stiffness

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:

$$F = \begin{cases} K \cdot x, & K \geq K_{\min} \\ K_{\min} \cdot x & K < K_{\min} \end{cases},$$

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 $x = x_{init} + x_R - x_C$

- x_{init} — initial spring deformation

- x_R — Absolute displacement of translational conserving port R

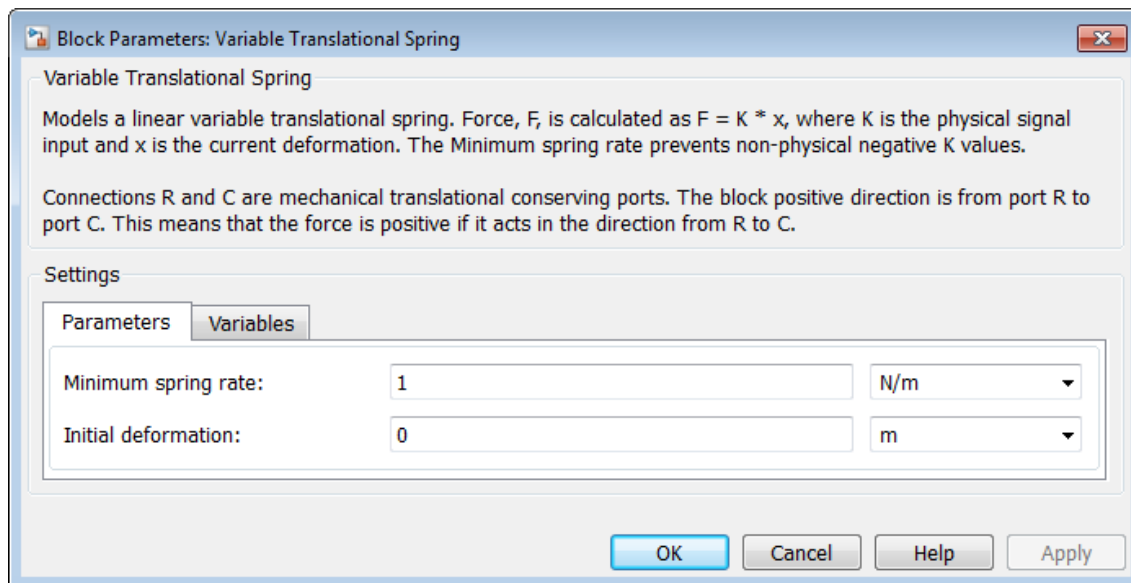- x_C — Absolute displacement of translational conserving port C

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

Block Parameters: Variable Translational Spring

Variable Translational Spring

Models a linear variable translational spring. Force, F, is calculated as F = K * x, where K is the physical signal input and x is the current deformation. The Minimum spring rate prevents non-physical negative K values.

Connections R and C are mechanical translational conserving ports. The block positive direction is from port R to port C. This means that the force is positive if it acts in the direction from R to C.

Settings

| Parameters | Variables |

| Minimum spring rate: | 1 | N/m |
| Initial deformation: | 0 | m |

OK   Cancel   Help   Apply

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.

Variables

Specify the desired initial values of the block dynamical variables. These variables guide model assembly at simulation time zero. When two or more variables conflict with each other, the priority setting determines which of the variables to satisfy. For more information, see "Set Priority and Initial Target for Block Variables".

Variables that you can specify include:

- f — Spring force
- v — Deformation rate
- x — Translational deformation

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

Vehicle Body

Purpose   Two-axle vehicle with longitudinal dynamics and motion and adjustable
mass, geometry, and drag properties

Library   Tires & Vehicles

Description
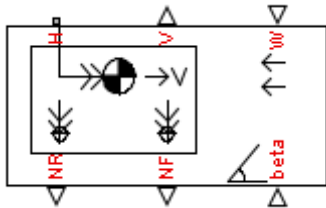
The Vehicle Body block models a two-axle vehicle, with an equal number
of equally sized wheels on each axle, moving forward or backward along
its longitudinal axis.

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 1-392.

Ports

You specify the headwind speed V_W (in meters/second) and the road
inclination angle β (in radians) through physical signal inputs at ports
W and beta, respectively.

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.

Vehicle Body

Dialog Box and Parameters

Block Parameters: Vehicle Body

Vehicle Body

Represents a two-axle vehicle body in longitudinal motion. The block accounts for body mass, aerodynamic drag, road incline, and weight distribution between axles due to acceleration and road profile. The vehicle does not pitch or move vertically relative to the ground.

Connection H is the mechanical translational conserving port associated with the horizontal motion of the vehicle body. The resulting traction motion developed by tires should be connected to this port. Connections V, NF, and NR are physical signal output ports for vehicle velocity and front and rear normal wheel forces, respectively. Wheel forces are considered positive if acting downwards. Connections W and beta are physical signal input ports corresponding to headwind speed and road inclination angle, respectively.

Settings

Parameters

Mass:	1200	kg
Number of wheels per axle:	2	
Horizontal distance from CG to front axle:	1.4	m
Horizontal distance from CG to rear axle:	1.6	m
CG height above ground:	0.5	m
Frontal area:	3	m^2
Drag coefficient:	0.4	
Initial velocity:	0	m/s

OK   Cancel   Help   Apply

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 computer the aerodynamic drag force on the vehicle. The default is `3`.

From the drop-down list, choose units. The default is meters-squared (`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

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).

Vehicle Body Model

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 β, 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.
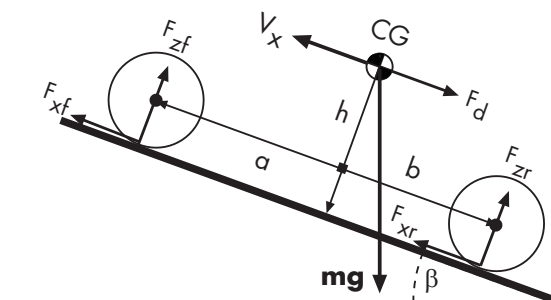
Vehicle Dynamics and Motion

Vehicle Model Variables

Symbol	Description and Unit
g	Gravitational acceleration = 9.81 m/s^2
β	Incline angle
m	Vehicle mass
h	Height of vehicle CG above the ground

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
ρ	Mass density of air = 1.2 kg/m^3
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.

Vehicle Body

$$mV_x = F_x - F_d - mg \cdot \sin\beta,$$
$$F_x = n(F_{xf} + F_{xr}),$$
$$F_d = \frac{1}{2}C_d\rho A(V_x - V_W)^2 \cdot \mathrm{sgn}(V_x - V_W)$$

Zero normal acceleration and zero pitch torque determine the normal force on each front and rear wheel:

$$F_{zf} = \frac{-h(F_d + mg\sin\beta + m\dot{V}_x) + b \cdot mg\cos\beta}{n(a+b)},$$

$$F_{zr} = \frac{+h(F_d + mg\sin\beta + m\dot{V}_x) + a \cdot mg\cos\beta}{n(a+b)}$$

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)

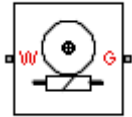**Related Examples**   • "Complete Vehicle Model"

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, ω_W and ω_G have the same sign. If the worm thread is left-handed, ω_W and ω_G have opposite signs.

Ports

W and G are rotational conserving ports. The ports represent the worm and the gear, respectively.

Worm Gear

Dialog Box and Parameters
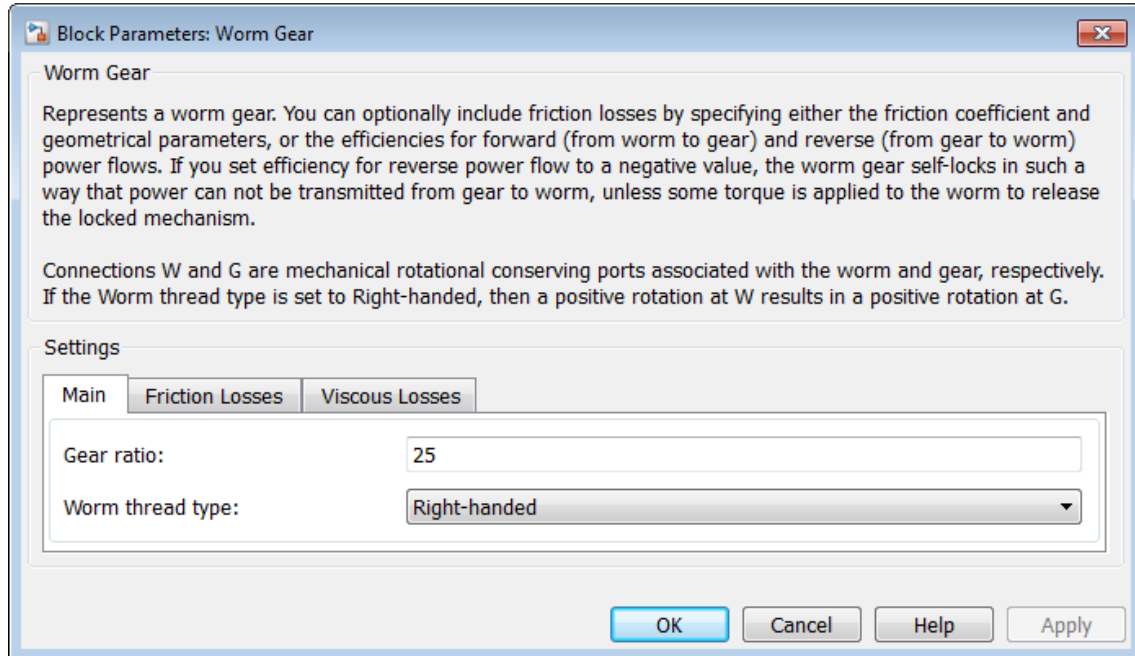
Main

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

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

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 a 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

Worm-gear efficiency

Efficiency η_{WG} of the power transfer from worm to gear.

Gear-worm efficiency

Efficiency η_{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

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
ω_W	Worm angular velocity
ω_G	Gear angular velocity
a	Normal pressure angle
λ	Worm lead angle
L	Worm lead
d	Worm pitch diameter
τ_G	Gear torque
τ_W	Torque on the worm
τ_{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.
τ_{fr}	Steady-state value of the friction torque at $\omega_G \rightarrow \infty$.
k	Friction coefficient
η_{WG}	Torque transfer efficiency from worm to gear
η_{GW}	Torque transfer efficiency from gear to worm
ω_{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 μ

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.

Power Flow Condition	Power Loss Condition	Output Driveshaft	Friction Torque τ_{fr}
Forward	$\omega_W\tau_W > \omega_G\tau_G$	Gear, ω_G	$R_{WG}|\tau_W| \cdot (1 - \eta_{WG})$
Reverse	$\omega_W\tau_W \leq \omega_G\tau_G$	Worm, ω_W	$|\tau_G| \cdot (1 - \eta_{GW})$

Geometric Surface Contact Friction

In the contact friction case, η_{WG} and η_{GW} are determined by:

- The worm-gear threading geometry, specified by lead angle λ and normal pressure angle a.

- The surface contact friction coefficient k.

$\eta_{WG} = (\cos\alpha - k \tan\lambda)/(\cos\alpha + k/\tan\lambda)$,

$\eta_{GW} = (\cos\alpha - k/\tan\lambda)/(\cos\alpha + k \tan\lambda)$.

Constant Efficiencies

In the constant friction case, you specify η_{WG} and η_{GW}, independently of geometric details.

Self-Locking and Negative Efficiency

η_{GW} has two distinct regimes, depending on lead angle λ, separated by the *self-locking point* at which $\eta_{GW} = 0$ and $\cos\alpha = k/\tan\lambda$.

- 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 η_{GW}, the larger the torque must be to release the mechanism.

η_{WG} is conventionally positive.

Meshing Efficiency

The efficiencies η 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 μ_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 $-\mu_W\omega_W$. ω_W is the angular velocity of the worm with respect to its mounting.

Worm Gear

The viscous friction coefficient μ_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 $-\mu_G\omega_G$. ω_G is the angular velocity of the gear with respect to its mounting.

Limitations
- Gear inertia is negligible. It does not impact gear dynamics.
- Gears are rigid. They do not deform.
- Coulomb friction slows down simulation. See "Adjust Model Fidelity".

See Also   Leadscrew | Sun-Planet Worm Gear

Related Examples
- "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_mdl 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

3

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

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...

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-generationRotational 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".
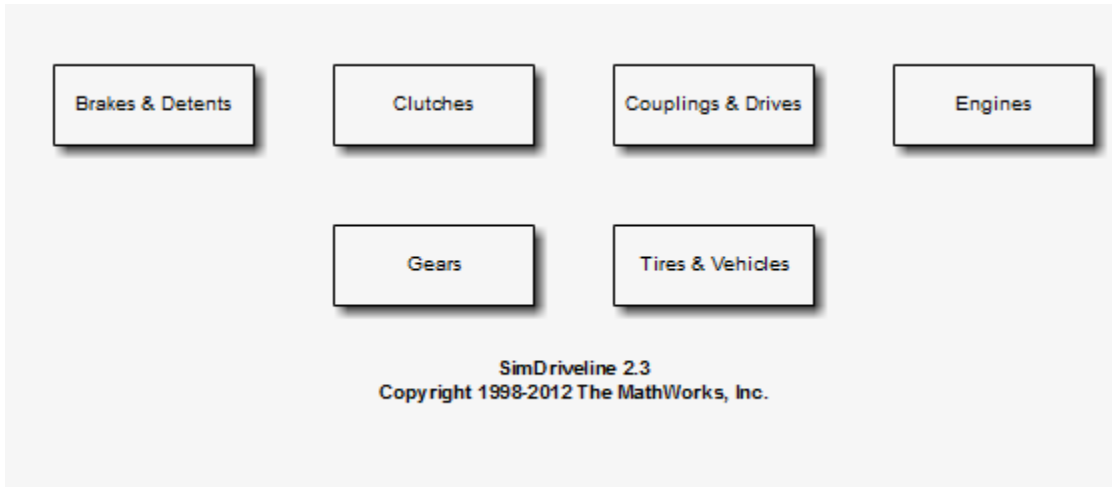

First Generation Block Library

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 `sdl_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 `sdl_classic`
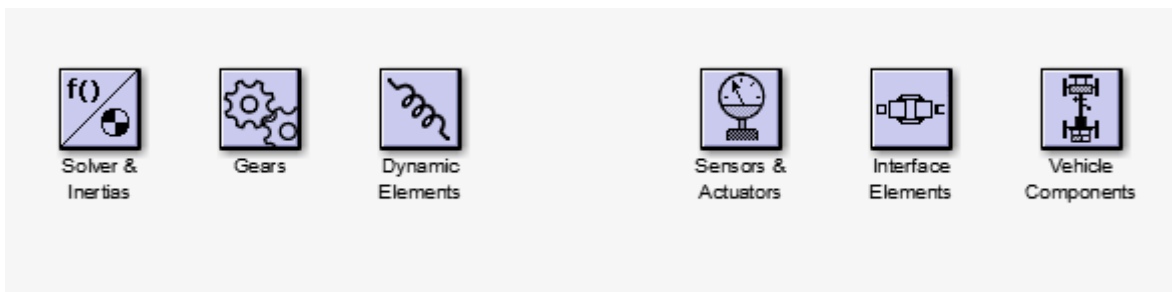
Transitional Library

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 Block	Second-Generation Replacement
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 Block	Simscape Foundation Replacement
Driveline Environment	Solver Configuration
Housing	Mechanical Rotational Reference

First-Generation Block	Simscape Foundation Replacement
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 Block	Simulink Replacement
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 **Initial Velocity** 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 μ, 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

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 `sdl_update` function converts first-generation models and user-defined libraries to second-generation equivalents. To convert a model, at the command line, enter:

```
newmdlname = sdl_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 `sdl_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.

Library	Block
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

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 `sdl_update`.

Convert Example Model

At the command line, enter:

```
v2mdlname1 = sdl_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 = sdl_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 = sdl_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...
"Abbreviations" on page A-2
"Angular Motion" on page A-2
"Gear Ratios" on page A-2

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
θ	Angle
ω	Angular velocity
τ	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 ω and τ have opposite signs from the input ω and τ.

B

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