Friction models for grease lubricated ball-race contacts

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Abstract. In this paper the authors theoretically investigated the influence of the hydrodynamic rolling friction forces in greased ball-race contacts at low loads. Two IVR-EHL transitions equations for hydrodynamic rolling friction forces have been applied to the ball-race contacts in a modified thrust ball bearing by using three types of greases having base oil viscosities between 40 mm²/s and 1000 mm²/s. Based on the total friction torque analytically estimated and experimentally determined has been evidenced the influence of the hydrodynamic rolling friction force on the total friction torque as the most important parameter for both proposed transition equations. The theoretical values of the total friction torque have been experimentally validated. The experiments was realized with a modified tree ball thrust bearing, without cage in a range of speed between 60 rpm and 400 rpm and for a normal load of 1.422 N for each ball.

1. Introduction
The friction in a lubricated ball bearing is a cumulative effect of the ball-race contacts, ball-cage contacts, cage ring contact, drag friction and sealing friction. All these friction sources are influenced by lubricant properties (usually viscosity of oil or base oil viscosity for grease), rotational speed, operating temperature, applied load, rolling bearing type and geometry. The total friction torque, as a cumulative friction sources in a rolling bearing, can be estimated by SKF methodology [1]. So, for a non sealing rolling bearing and for small quantity of lubricant, the total friction torque \( M \) includes two components: rolling \( (M_r) \) and sliding friction \( (M_s) \) moments. In both two component is present the viscosity of oil or of the base oil of the grease at the operating temperature. Cousseau et al. [2] experimentally evaluated the friction torque in 51107 thrust ball bearing using five sort of greases having mineral and synthetic base oil with viscosity between 33 and 130 mm²/s. The results of the total friction torque for each grease have been compared with the total friction torque calculated by SKF methodology and various values for sliding friction coefficient \( \mu_s \) depending on the grease type have been obtained. Cousseau et al. [2] concluded that in mixed lubrication regime each grease will have another weight factor \( \varphi_{bl} \) [1] for the sliding friction coefficient, depending on grease rheology. Cousseau et al. [3] continued the experimental research and determined the friction torque in a 51107 thrust ball bearing using seven different sorts of greases. They determined the viscosity of the bleed oil for all seven greases and compared with the viscosity of base oil. Important differences have been obtained between base oil viscosity and bleed viscosity, especially for grease with polypropylene thickener. Also, the authors calculated the two components of the total friction torque from SKF methodology by using the viscosity of bleed oil and compared with the experimental values of the total friction torque for each grease. Vengudusamy et al. [4] experimentally determined that film thickness and friction coefficient in a test rig including rolling and sliding steel ball and glass disc
contact. For all nine types of tested greases the authors observed that for very low speed the film thickness is high at very low rolling speed speeds (0.0001 to 0.01 m/s), then drops with increasing speed to 0.1 m/s until it starts increasing again corresponding to base oil curve at high speeds. The transition from thickener to base oil-dominant region is dependent on thickener type. The friction coefficient obtained by the authors [3] for all nine greases varies between 0.04 to 0.08 depending on the rolling speed and thickener type. De Laurentis et al. [5] experimentally determined the relationship between bearing grease composition and rolling-sliding friction in lubricated contacts in a ball-on-disc tribometer. Eight commercially available greases having mineral and synthetic base oil with viscosity at 40°C between 18 and 420 mm²/s have been tested between 0.01 to 1 m/s. Friction coefficient values between 0.01 to 0.08 have been obtained and the higher values correspond to the lithium soap greases. Also, the authors concluded that base oil viscosity, nature of base oil and the interaction between grease thickener and base oil influence the friction coefficient.

By using a modified thrust ball bearing the authors performed experimental tests in order to establish the rolling friction in lubricated ball-race contacts. Olaru et al. [6] proposed a methodology based on the spin-down applied to a modified thrust ball bearing considering only three balls and no cage. With this methodology the authors determined the rolling friction coefficient in low load dry ball-race contacts until the values of 0.0002-0.0004.

Bălan et al. [7] determined the friction torque in a 51205 modified thrust ball bearing having only 3 balls and no cage. The tests were realized by spin-down method in lubrication conditions with three mineral oils having the viscosities between 0.02 to 0.08 Pa·s. The authors evaluated the influence of the lubricant viscosity on the friction torque using the Biboulet and Houpert transition model [8] for hydrodynamic rolling forces developed in ball-race contacts and experimentally validated one of the four relations derivate from [8].

Therefore, Ianuş et al. [9] experimentally determined the friction torque in a modified thrust ball bearing with three balls both operating in dry and lubricated conditions with two types of greases. The experimental values of the total friction torque generated by the three balls have been compared with the friction torque values obtained with SKF methodology [1] adapted for a trust ball bearing with 3 balls. In the range of rotational speed between 60 and 300 rpm and with a normal load of 1.42 N on each ball the friction torque calculated with SKF methodology resulted in (3-4) times smaller than the experimental values. In addition, the authors adapted the Biboulet and Houpert model for hydrodynamic rolling forces considering the base oil viscosities of the two greases, and validated the experimental results.

Recently, Ianuş et al. [10] developed a complex methodology to evaluate the total power loss in grease lubricated ball-race contacts considering both power loss caused by sliding speed on contact ellipses and power loss as result of rolling friction. Total power loss calculated for all six contact ellipses in a modified thrust ball bearing with three balls has been validated by the total power loss resulting as product between measured friction torque and angular speed.

In this paper the authors propose to found the best theoretical model for hydrodynamic rolling friction force derived from the BibouletandHoupert methodology acceptable for three different types of grease. Two IVR-EHL transitions equations for hydrodynamic rolling friction forces are proposed to be applied to determine the friction torque generated by a ball-race contact in a thrust ball bearing. To validate these equations the authors realized a lot of experiment on a modified thrust ball bearing having only three balls and lubricated with the different greases.

2. Analytical model

2.1. Forces and moments acting on a ball

In figure 1 are illustrated a thrust ball bearing operating in lubrication conditions having the rotating upper race and lower race fixed and the forces and moments in the ball-races contacts. Was ignored in our analysis the ball-cage forces and moments. Following forces and moments have been considered: the hydrodynamic rolling forces $FR_{l,u}$, the pressure forces $FP_{l,u}$, the traction forces $FS_{l,u}$, the inertial
force acting on the ball $F_{ib}$, the rolling resistant moments due to elastic hysteresis $MER_{l,u}$ and the pivoting friction moment $MP_{l,u}$. The index $l$ and $u$ refer to lower and upper race–ball contact respectively.

![Thrust Ball Bearing Diagram](image)

**Figure 1.** (a) A normal thrust ball bearing; (b) the forces and moments acting on a ball in rolling direction.

All forces and moments acting on a ball can be developed with various equations which not need to solve the sliding speed distribution on contact ellipses excepting the traction forces $FS_{l,u}$. In these circumstances the best solution is to determine the traction forces from the equilibrium of the forces and moments acting on the ball in rolling direction. Must be note that when the race radius is infinite in the rolling direction (as in a thrust ball bearing), the pressure forces are the following expression: $FP = 2 \cdot FR$ [11]. With this specification, the traction forces on contact ellipses for lower and upper races are determined by following equations [11,12]:

$$FS_l = \frac{MER_l + MER_u}{d_b} + \frac{F_{ib}}{2} + FR_u$$  \hspace{1cm} (1)

$$FS_u = \frac{MER_l + MER_u}{d_b} - \frac{F_{ib}}{2} + FR_l$$  \hspace{1cm} (2)

Considering the algebraic sum of all the three forces acting in the upper ball-race contact (as in figure 1) it can be obtained the total tangential force acting on rotating upper race, $F_{t_u}$, according to the following relation:

$$F_{t_u} = \frac{MER_l + MER_u}{d_b} + (FR_u + FR_l) - \frac{F_{ib}}{2}$$  \hspace{1cm} (3)

By similar procedure, it can be obtained the total tangential force in ball-lower race contact $F_{t_l}$ when the lower race is fixed:

$$F_{t_l} = \frac{MER_l + MER_u}{d_b} + (FR_u + FR_l) + \frac{F_{ib}}{2}$$  \hspace{1cm} (4)

In equations (1) – (4) $d_b$ is the ball diameter.

The friction torque generated by a ball on the upper race is determined by following relation:

$$T_{z_u} = \frac{(MER_l + MER_u) \cdot d_m}{2 \cdot d_b} + \frac{(FR_u + FR_l) \cdot d_m}{2} + MP_u - \frac{F_{ib} \cdot d_m}{4}$$  \hspace{1cm} (5)

where $d_m$ is the mean diameter of the thrust ball bearing.
Equation (5) can be considered as a sum of four friction torque components generated by a single ball on the upper race:

\[ T_{z_u} = T_{z_{MER}} + T_{z_{FR}} + T_{z_{MP}} - T_{z_l} \]

where \( T_{z_{MER}} \), \( T_{z_{FR}} \), \( T_{z_{MP}} \) and \( T_{z_l} \) are the components generated by elastic hysterisis, hydrodynamic rolling friction, pivoting friction and ball’s inertia, respectively.

2.2. Rolling resistant moment due to hysteresis

\( \text{MER}_{l,u} \) are the rolling resistant moments due to elastic hysteresis losses in ball-race contact, and can be determined by equations [11]:

\[
\text{MER}_{l,u} = 7.4810^{-7} \left( \frac{d_k}{2} \right)^{0.33} \cdot Q^{1.33} \cdot \left[ 1 - 3.519 \cdot 10^{-3} (k_{l,u} - 1)^{0.8063} \right]
\]

(7)

In equation (7) \( Q \) is normal load and \( k_{l,u} \) are the radius ratio for lower and upper ball-race contacts, respectively and are determined by the relations [10]:

\[
k_l = \frac{2 \cdot f_l}{2 \cdot f_l - 1} \quad k_u = \frac{2 \cdot f_u}{2 \cdot f_u - 1}
\]

(8)

where \( f_l \) and \( f_u \) are the conformities of the two races [10].

2.3. Hydrodynamic rolling friction forces

For the hydrodynamic rolling friction force FR miscellaneous relationships are suggested in literature [11,12]. Depending of the IVR (Iso Viscous Rigid) or EHL (Elasto Hydrodynamic Lubrication) regime in ball-races contacts can be use different hydrodynamic rolling friction relations [7,11,12]. Biboulet and Houpert [8] proposed a transition equation for IVR and EHL regime given by following equation:

\[
FR = \left[ \frac{1}{1 + M / 6.6} \right] \cdot FR_{IVR} + \left[ \frac{M / 6.6}{1 + M / 6.6} \right] \cdot FR_{EHL}
\]

(9)

where \( FR_{IVR} \) and \( FR_{EHL} \) are the hydrodynamic rolling forces for IVR and EHL lubrication regime, respectively and \( M \) is a transition parameter from IVR to EHL regime.

The parameter \( M \) is definite by Biboulet and Houpert [8] with following equation:

\[
M = 0.5549 \cdot k^{0.6029} \cdot W \cdot U^{-0.75}
\]

(10)

where \( W \) and \( U \) are the load and speed dimensionless parameters.

For the hydrodynamic rolling friction forces we propose to consider both the Houpert’s equations [11,12] and the BibouletandHoupert’s equations [8], as follows:

- Houpert’s equations for \( FR_{IVR} \) and \( FR_{EHL} \) [11,12]:

\[
FR_{IVR,H} = 1.213 \cdot E^* \cdot R^2 \cdot k^{0.358} \cdot U^{0.636} \cdot W^{0.364}
\]

(11)

\[
FR_{EHL,H} = 2.765 \cdot E^* \cdot R^2 \cdot k^{0.35} \cdot U^{0.656} \cdot W^{0.466} \cdot G^{0.022}
\]

(12)

- Biboulet& Houpert’s equations for \( FR_{IVR} \) and \( FR_{EHL} \) [8]:

\[
FR_{IVR,B} = 2.9766 \cdot E^* \cdot R^2 \cdot k^{0.3316} \cdot W^{1/3} \cdot U^{2/3}
\]

(13)

\[
FR_{EHL,B} = 7.5826 \cdot E^* \cdot R^2 \cdot k^{0.4055} \cdot W^{1/3} \cdot U^{3/4}
\]

(14)

where \( E^* \) is reduced Young modulus of the ball and race, \( R_e \) is the equivalent ball-race radius in rolling direction and \( G \) is the dimensionless material parameter.

By including equations (11) - (14) in equation (9) it can be obtained two different equations for the transition hydrodynamic rolling force as follows:

- Houpert’s transition rolling force \( FR_{Trans,H} \) definite by equation:

\[
FR_{Trans,H} = \left[ \frac{1}{1 + M / 6.6} \right] \cdot FR_{IVR,H} + \left[ \frac{M / 6.6}{1 + M / 6.6} \right] \cdot FR_{EHL,H}
\]

(15)

- Biboulet transition rolling force \( FR_{Trans,B} \) definite by equation:

\[
FR_{Trans,B} = \left[ \frac{1}{1 + M / 6.6} \right] \cdot FR_{IVR,B} + \left[ \frac{M / 6.6}{1 + M / 6.6} \right] \cdot FR_{EHL,B}
\]

(16)
2.4. Pivoting friction moment

The pivoting friction moments $MP$ normal to the centre of the contact ellipse, are determined with the following relations [11]:

$$MP = \frac{3}{8} \mu_s \cdot Q \cdot a$$

(17)

where $\mu_s$ is the sliding friction coefficient on ball-race contact ellipse and $a$ is semi major contact ellipse axis. The sliding friction coefficient is depending of the film thickness and of the ball and race’s roughness. The semi major contact ellipse axis is depending of the ball-race geometry and normal load and can be determined according to Houpert relations [11].

2.5. Pivoting friction moment

The inertial force $F_{ib}$ is given by equation:

$$F_{ib} = -\frac{m_b \cdot d_y}{2} \cdot \frac{d\omega_c}{dt}$$

(18)

The inertial force is included in calculus when the orbital speed of the ball $\omega_c$ is in decelerating process.

Finally, the two models to determine the total friction torque generated by a single ball on upper race are derived from equations (6), (15), (16):

$$T_{z_{u-H}} = T_{z_{u-M}} + T_{z_{u-Trans-H}} + T_{z_{u-M}} - T_{z_l}$$

(19)

$$T_{z_{u-B}} = T_{z_{u-M}} + T_{z_{u-Trans-B}} + T_{z_{u-M}} - T_{z_l}$$

(20)

3. Simulation program

Both two friction torque equations (19) and (20) has been applied for a modified 51205 thrust ball bearing having only three balls, low loaded and lubricated with three different greases A, B and C. The most important characteristics of the greases are indicated in the Table 1.

The geometrical parameters of the modified thrust ball bearing are: the balls diameter $d_b = 7.938$ mm ($5/16\,\text{"}$), the race conformities (ratio between curvature radius of race and ball diameter) are $f_r = f_u = 0.53$, the mean diameter $d_m = 36$ mm, the average roughness for both races $R_{q_r} = 0.06 \, \mu m$ and the average roughness of the balls was $R_{q_b} = 0.03 \, \mu m$. Simulation was realized for an axial load $F_a=4.26$ N and distributed for three balls the normal load for each ball $Q = 1.42$N. The rotational speed of upper race varied between 100 rpm and 400 rpm.

| Type of grease | Grease A | Grease B | Grease C |
|---------------|----------|----------|----------|
| NLGI grade    | MOL Liton 00 | MOL Alubia AK 2G | Silicone grease SSX |
| Base oil viscosity | 40 mm$^2$/s at 40°C | 150 mm$^2$/s at 40°C | 1000 mm$^2$/s at 25°C |
| Thickener      | Lithium and calcium | Aluminium complex | Amorphous fumed silica. |
| Dropping point | 180 °C | 240 °C | 250°C |
| Type of base oil | synthetic base oil (PAO) | Mineral oil | PDMS Silicon Oil |

In equations (11) - (14) following relations for equivalent radius in rolling direction $R_y$ and for average entrainment speed in ball-race contacts $v_{lu}$ have been considered:

$$R_y = R_{yu} = \frac{d_y}{2}$$

(21)

$$v_i = v_{lu} = \frac{\pi \cdot d_m}{120} \cdot n_u$$

(22)

where $n_u$ is rotational speed of upper race in rpm.
In equation (17) the friction coefficient in pivoting motion $\mu_s$ is evaluated according to the lubricant parameter $\Lambda$ using relations [7]:

$$
\mu_s = \mu_{EHL} \cdot 0.82 \cdot \Lambda^{0.28} + \left(1 - 0.82 \cdot \Lambda^{0.28}\right) \mu_a \\
\mu_s = \mu_{EHL} \\
$$

if $\Lambda < 3$

if $\Lambda > 3$

(23)

where $\mu_{EHL}$ is the friction coefficient in full film conditions and was considered having an average value of 0.05 and $\mu_a$ is the friction coefficient in the asperities contact considered with a value of 0.11.

The lubricant parameter $\Lambda$ is defined by equation:

$$
\Lambda = \frac{h_o}{\sqrt{Rq_r^2 + Rq_b^2}}
$$

where $h_o$ is the central film thickness in ball-race contacts and is calculated according to Hamrock-Dowson equation considering the base oil viscosities [10]. For grease A the friction coefficient $\mu_s$ varied between 0.07 to 0.05 and for grease B and C the friction coefficient $\mu_s$ was equal to 0.05 in the range of the rotational speed between 100 and 400 rpm. Also, the semi major axis of the contact ellipse $a_{lu} = 0.124$ mm.

The components of the total friction torques given by equations (19) and (20) have been calculated for the grease A considering temperature of 25°C, normal load $Q = 1.42$N and a variation of the rotational speed of the upper race between 100 and 400 rpm. The inertial component has been ignored in this simulation. The results have been presented in figure 2.

First conclusion is that the most important component for the friction torque (approx 98%) is given by hydrodynamic rolling friction component. The second conclusion consists in existence of an important difference between the two proposed transition rolling friction forces: $FR_{Trans_H}$ and $FR_{Trans_B}$. So, the friction torque generated by $FR_{Trans_B}$ is higher with approximatively 70% from the friction torque generated by $FR_{Trans_H}$. Similar differences have been obtained for grease C and D.

![Figure 2](image-url)  
**Figure 2.** Variation of the friction torque components as function of rotational speed for Grease A.

Because the hydrodynamic rolling force are the most important parameter in evaluation of the friction torque, the IVR and EHL partition were determined for all three greases, according to following relations [13]:

$$
IVR_{Partition} = \left[\frac{1}{(1 + M / 6.6)}\right] \cdot 100 \quad (\%) \quad (25)
$$

$$
EHL_{Partition} = \left[\frac{M / 6.6}{(1 + M / 6.6)}\right] \cdot 100 \quad (\%) \quad (26)
$$

where $M$ is Biboulet and Houpert parameter given by equation (10).
In figure 3 are presented the variation of the two partitions for the three greases between 100 and 400 rpm rotational speed of the upper race.

![Figure 3. Variation of the IVR and EHL partition as function of upper race rotational speed for greases A, B and C.](image)

As a conclusion, for all three greases the IVR partition is dominant in the rotational speed considered. It can be explained by the very low normal load acting in the ball-race contact combined with high base oil viscosities for the greases (especially for grease B and C).

The next question consists in what is the adequate model for each grease validated by experiments.

4. Experimental validation
The major difference between the two models for total friction torque generated by a ball in very low load conditions illustrated in figure 2 for grease A need to be verified by experiments for all three greases.

The modified 51205 thrust ball bearing has only 3 balls equidistant positioned between the lower and upper race and the cage was eliminated. A solitary loading disc with the running track is mounted on upper race. The lower race is mounted alone with a rotating table as in figure 4. The weight of the disc and upper race is 0.435 kg that means an axial load \( F_a = G = 4.267 \) N. This axial load divided to the three balls lead to a normal load \( Q \) of 1.422 N on each ball-race contact. The experiments are based on spin-down method in which was monitored the motion of the upper race by a camera in the deceleration process from a given rotational speed to zero.

![Figure 4. The modified 51205 thrust ball bearing with three balls.](image)
This method consists on the start of the rotating table and the lower race with a constant rotational speed \( n_l \). During a time, as a result of friction between the balls and the races the rotational speed of the upper race and the attached disc, \( n_u \), will increases until a stable rotational speed equal to the rotational speed of the lower race. At this moment, the rotational table with lower race is suddenly stopped, while upper race and disc starts a deceleration process. So the kinetic energy of the ensemble including upper race and disc will be consumed by the friction in the six ball-race contacts. The deceleration process takes time to stop the upper race and disc. The total number of the disc’s rotations and the deceleration time monitored by the camera were the basis for the integration of the following dynamic equation:

\[
J \frac{d\omega_u}{dt} + T_z(\omega_u) = 0 \tag{27}
\]

where \( J \) is the moment of inertia of the ensemble formed by the upper race and disc and \( T_z(\omega_u) \) is the total friction torque developed by friction generated in all six ball-race contacts as a function of angular speed of the upper disc \( \omega_u \).

Based on the demonstration that the total friction torque in our conditions is dominant by the hydrodynamic rolling force, the total friction torque \( T_z(\omega_u) \) was expressed by following relation:

\[
T_z(\omega_u) = K^* \cdot \omega_u^n \tag{28}
\]

where \( K^* \) and exponent \( n \) have been obtained for every experiment according to the monitored rotations and time in deceleration process. Details on the integration of the equation (27) are presented in [7, 9, 13].

The total friction torque experimentally determined for each grease have been indicated in figure 5 as function of the product between base oil viscosity at the operating temperature (23-25)°C and rotational speed of the upper race.

In the same graphics are presented the total friction torques calculated for three balls according to equation (19) for grease A and B and according to equation (20) for grease C. The inertial component from the total friction torque for three balls have values between \( 5 \cdot 10^{-7} \) and \( 5 \cdot 10^{-6} \) N·m.

![Figure 5](image)

**Figure 5.** The variation of the experimental and theoretical total friction torque as function of the product between base oil viscosity and upper race rotational speed for all three greases.

As a first conclusion it can be observed that both theoretical models for the total friction torque described by equations (19) and (20) can approximate very well the experimental values of the friction torque generated only by the ball-race contacts in a modified thrust ball bearing.
In the same time important differences between the total friction torques obtained with the three greases have been observed:

- The theoretical model for the total friction torque based on the Biboulet’s transition rolling force $FR_{Trans,B}$ is the best theoretical model for grease A. Also this theoretical model is a good approximation for the grease B.
- The theoretical model for the total friction torque based on the Houpert’s transition rolling force $FR_{Trans,H}$ is the best theoretical model for grease C, although the base oil viscosity of this grease is the biggest.

  In our opinion this difference between the friction torque generated by grease C and the two greases A and B can be caused by the differences between thickener soap properties.

  Bălan et al. [7] using four mineral oils with viscosities between 0.02 to 0.35 Pas and the same modified 51205 thrust ball bearing having 3 balls, no cage and the similar normal load obtained that the best theoretical model validated by experiments was based on Houpert’s transition hydrodynamic rolling force.

  Comparing our results obtained using greases A and B with the results obtained by Bălan et al.[7] using mineral oils it can be concluded that presence of the thickener soap leads to an increase of the hydrodynamic rolling force and, as a consequence the Biboulet’s transition rolling force is better to be use in the total friction torque model.

5. Conclusions

Two analytical models based on the Biboulet’s and Houpert’s transition hydrodynamic rolling force in greased ball – race contacts have been used to evaluate the friction in rolling ball-race contacts.

The two models have been included in the equation for total friction torque generated in a modified thrust ball bearing having only 3 balls without cage to be evidenced only the influence of the rolling friction in greased ball-race contacts.

Three different greases having base oil viscosity between 40 to 1000 mm$^2$/s have been considered both for total friction torque simulation and experimental determinations.

It has been demonstrated that for very low axial load the dominant effect in the ball-race contact is given by the hydrodynamic rolling forces. The total friction torque including the two analytically models were experimentally verified using the spin-down method by monitoring the motion of the upper race with a camera in the deceleration process from a given rotational speed to zero.

A 51205 modified thrust ball bearing having 3 balls with 7.938 mm diameter was used in the experiments. The normal load of 1.422 N on every ball is obtained by adding a disc on the upper race, that means a maximum contact pressure between the balls and the races of 0.264 GPa.

The simulation and the tests were realized for a rotational speed of upper race between 100 and 400 rpm that means values for rolling speed in ball race contacts between 0.1 and 0.4 m/s.

It was observed that the theoretical model for the total friction torque based on the Biboulet’s transition rolling force can be used to estimate with good accuracy the friction torque for grease A and B and the theoretical model based on the Houpert’s transition rolling force can be used to estimate with good accuracy the friction torque for grease C, although the base oil viscosity of this grease is the biggest.

6. References

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