Lateral stability simulation and analysis for wheel loaders based on the steady-state margin angle

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Abstract: Wheel loaders are prone to lateral tilting accidents due to their complex working environment, variable structure frame and the large offset of the centroid position. The accidents may seriously threaten the driver’s safety. This paper proposes a steady-state margin (SSM) angle as a new instability threat indicator to analyse lateral stability of wheel loaders. Firstly, the structural characteristics and the rollover process of a wheel loader are presented, and the SSM angle is defined. Secondly, on the assumption that the tyres are rigid, the calculation process of the SSM angle is described in detail, then the SSM angle of the XG953 wheel loader is computed. A virtual prototyping simulation was performed under the environment of ADAMS. The maximum relative error between the theoretical calculation and simulated result was 4.55%, within the permissible speed range. When changing the tyre parameters to normal tyre parameters, the relative error became larger. Finally, an indirect method is proposed to deal with the influence of tyres on the SSM angle and reduce the relative error.

Keywords: wheel loaders; lateral stability; steady-state margin angle; ADAMS; simulation.


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1 Introduction

1.1 Motivation

The wheel loader is a kind of off-road heavy vehicle which is widely used in engineering construction because of its high efficiency, good mobility and easy operation (Zhang et al., 2009). With regard to its complex working environment, variable structure frame and the large offset of the centroid position, the wheel load lacks stability. Compared with on-road vehicles (Ren et al., 2013), wheel loaders are more prone to lateral tilting accidents, which is a serious threat to the driver’s safety (Bao et al., 2011). Some effective protective measures must be taken to prevent lateral tilting accidents.

It has been proven that the extent of the driver’s injury is related to the extent of the roof deformation (Orlowski et al., 2004). A roll-over protective structure (ROPS), which is a passive safety technology, was introduced to protect the driver’s safety by regulating a strength requirement and an energy absorption requirement for the operator compartment (Clark et al., 2006). In the lateral tilting accident, ROPS can minimise the driver’s injuries due to the operator compartment deformation. However, it has been proven that drivers are still often seriously hurt or die in a lateral tilting accident even though ROPS meets the requirements of the standard.

To fundamentally prevent lateral tilting accidents, taking passive safety measures is not enough. The more effective solutions should be a roller early warning or active control and making the change from ‘passive safety’ to ‘active safety’ (Islam and He, 2013; Wideberg and Dahlberg, 2013; Chadli et al., 2008). The rollover stability of wheel loaders consists of lateral stability and longitudinal stability, and the lateral instability happens frequently during the wheel loader operation (Bao et al., 2011). Foreseeing the lateral tilting accident and providing early warning information can alert the driver or the control system to actively take action to avoid the occurrence of lateral tilting. This paper aims to develop an instability threat indicator which can effectively indicate the driving stability of wheel loaders.

1.2 Related work

In recent years, a large number of theoretical studies of the stability of on-road vehicles have been conducted (Du et al., 2013; Zhang et al., 2013), such as lateral stability analysis, the early warning for rollover accidents and the active control for lateral tilting. In 1987, the rollover protection energy reserves (RPER) was developed to predict the vehicle tripped rollover (Nalecz and Bindemann, 1987), which is defined as the difference between the rotational kinetic energy and the rollover required energy. Rakheja and Piche (1990) proposed the lateral acceleration as an instability threat indicator; which predicts the state of the vehicle by comparing the lateral acceleration with a threshold value. It performs poorly for wheel loaders, since the threshold of the lateral acceleration changes with its operating state. A lateral-load transfer ratio (LTR) was proposed in the same year as an instability threat indicator (Preston-Thomas and Woodroofe, 1990).

The instability indicators described above are based on on-road vehicles, and they cannot be directly applied to wheel loaders that have a variable mass and a variable centroid position. Therefore, a stable angle is proposed as a rollover indicator based on the steering angle of the vehicle and normal forces on four wheels to characterise the instability of an articulated vehicle effectively (Zhu et al., 2014). But it is difficult in practice to measure the normal forces on four wheels.

This paper proposes a steady-state margin (SSM) angle as an instability threat indicator for wheel loaders. The SSM angle is defined as the angle between the rollover plane and the resultant force of gravity and the lateral force to predict the lateral instability of a wheel loader. The SSM angle is computed from attitude and kinetic parameters, which can be measured relatively easily.

1.3 Notation

The notations used in the following sections are firstly presented as follows:

\[ \theta \] the rotating angle of the working device
\[ \alpha \] the steering angle
\[ l_1, l_2 \] the distance from the articulated point to the front and rear axle
\[ \gamma \] the swinging angle of the rear axle
\[ M_i \] the mass of each part

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2 Analyse the lateral stability of wheel loaders

2.1 The structure of a wheel loader

A structure diagram of an articulated wheel loader is illustrated in Figure 1. The front and rear frame are connected by a vertical revolute joint. This kind of articulated steering structure has advantages of a smaller turning radius and a more robust design (Rehnberg, 2008), but this structure also makes lateral stability inadequate (Rehnberg et al., 2011). When the front frame of a fast moving wheel loader needs to turn sharply to avoid the obstacle, the rear frame still has the tendency to move forward, which further increases the steering angle, resulting in increased lateral force and poor lateral stability. This phenomenon is defined as the ‘rearward amplification ratio’ (RWA) (Bao et al., 2011).

Figure 1 Structure diagram of a wheel loader

For most wheel loaders, the rear axle is mounted on a pivoting joint. This suspension structure ensures that the four wheels contact the ground at the same time in complex situations. However, this structure also limits the driving speed. These special structural features of wheel loaders make their rollover process different from on-road vehicles.

2.2 The lateral rollover process

As shown in Figure 2, a four-wheeled wheel loader can be regarded as a vehicle body supported by three points: A, B and E. A and B are the points where the two front wheels contact the ground respectively, and E is the intersection of the swinging axes on the rear axle and the normal plane through the axis of the rear axle. Generally, E is further away from the floor than A and B.

Figure 2 The lateral roll-over process (see online version for colours)

When the lateral force is large enough, the vehicle body rotates around the axis BE (or the axis AE). Until the normal force on A (or B) wheel becomes zero, this condition can be treated as first order instability. If the vehicle body continues rotating around the axis BE (or the axis AE), the vehicle body contacts with the rear bridge then the vehicle body rotates around the axis BC (or the axis AD) with the rear bridge together, resulting in the A and D wheels (or the B and C wheels) off the ground, known as second order instability. The wheel loader will be in danger of roll-over when the first order instability happens. Therefore, the first order instability can be used as the theoretical model to predict the lateral rollover accident.

3 The calculation of SSM angle

3.1 Definition

To analyse the force that influences the stability of wheel loaders, both internal and external forces are taken into account, including gravity, inertial forces, and aerodynamic drag. The resultant force, \( F \), is expressed at the centroid position (Steven and Iagnemma, 2008).

During the rollover process, the angle between the line of action of \( F \) and the roll-over plane \( \mathbf{GBE} \) (or \( \mathbf{GAE} \)) is decreasing (Figure 3). When the line of action of \( F \) lies outside the rollover plane, the roll-over happens (Iagnemma et al., 2003). The angle between the line of action of \( F \) and the normal vector of the rollover plane is defined as \( \varphi \). For the purposes of this study, \( \Phi = \varphi - 90^\circ \) is defined as the SSM angle. The wheel loader is stable when \( \Phi \) is greater than zero, and becomes unstable when \( \Phi \) is less than zero.
3.2 Calculation process

To simplify the calculation process, two assumptions have been made:

1. The tyres of wheel loaders are rigid, and the stiffness and damping are big enough.
2. The friction coefficient of pavement is big enough.

As an engineering vehicle with variable structure, the centroid position of the wheel loader will change when steering, bucket lifting and vehicle body swinging around the rear axle.

The wheel loader should be divided into four parts, namely bucket, front frame, rear frame and rear axle. Four coordinate systems $O_{0}(x_{0}, y_{0}, z_{0})$, $O_{1}(x_{1}, y_{1}, z_{1})$, $O_{2}(x_{2}, y_{2}, z_{2})$ and $O_{3}(x_{3}, y_{3}, z_{3})$ are fixed on the four parts, as illustrated in Figure 4. The points $O_{0}$ and $O_{1}$ are the intersection of the axis of the steering hinge and the swinging axis on the rear axle. The point $O_{2}$ is the intersection of the rotation axis of the working device and the longitudinal symmetry plane of the vehicle body. The point $O_{3}$ coincides with point $E$.

In order to compute the SSM angle, the roll-over plane $\odot GBE$ and represented resultant force, $F$, must be obtained. The point $E$ is a fixed point in $O_{0}(x_{0}, y_{0}, z_{0})$, which position is represented as $\overset{o}{p}_{E}$. And the point $B$ is a fixed point in $O_{1}(x_{1}, y_{1}, z_{1})$, which position is represented as $\overset{1}{p}_{B}$. The centroid position $\overset{3}{p}_{G_{i}}$ needs to be computed to obtain the roll-over plane $\odot GBE$.

\[
\begin{align*}
\overset{2}{p}_{G_{i}} &= (0, y_{1}, z_{1})^T \quad (1) \\
\overset{1}{p}_{O} &= (0, dy, dz)^T \quad (2) \\
\begin{bmatrix}
1 & 0 & 0 \\
0 & \cos \theta & -\sin \theta \\
0 & \sin \theta & \cos \theta \\
\end{bmatrix} \quad (3) \\
\overset{1}{p}_{G_{i}} &= (0, y_{2}, z_{2})^T \quad (4) \\
\overset{0}{p}_{O} &= (0, 0, 0)^T \quad (5) \\
\begin{bmatrix}
\cos \alpha & -\sin \alpha & 0 \\
\sin \alpha & \cos \alpha & 0 \\
0 & 0 & 1 \\
\end{bmatrix} \quad (6) \\
\overset{3}{p}_{G_{i}} &= (0, y_{3}, z_{3})^T \quad (7) \\
\overset{0}{p}_{O} &= (0, -l_{2}, 0)^T \quad (8) \\
\begin{bmatrix}
\cos \gamma & 0 & \sin \gamma \\
0 & 1 & 0 \\
-\sin \gamma & 0 & \cos \gamma \\
\end{bmatrix} \quad (9) \\
\overset{0}{p}_{G_{i}} &= (0, y_{4}, z_{4})^T \quad (10) \\
\begin{align*}
\overset{0}{p}_{G_{i}} &= \overset{0}{p}_{O} + \overset{1}{R} \overset{1}{p}_{G_{i}} \quad (11) \\
\overset{0}{p}_{G_{i}} &= \overset{0}{p}_{O} + \overset{1}{R} \left( \overset{1}{p}_{20} + \overset{1}{R} \overset{2}{p}_{G_{i}} \right) \quad (12) \\
\overset{0}{p}_{G_{i}} &= \overset{0}{p}_{O} + \overset{3}{R} \overset{3}{p}_{G_{i}} \quad (13) \\
\overset{0}{p}_{G} &= \sum_{i=1}^{4} M_{i} \overset{0}{p}_{G_{i}} \quad (14)
\end{align*}
\end{align*}
\]
Lateral stability simulation and analysis for wheel loaders based on the steady-state margin angle

\[ i_p = \left( \frac{d l_2 - h}{d} \right) \times \bar{R} \]

\[ \bar{R} = \hat{p}_o \times (0, -l_2, 0) \]

\[ n = \frac{P_G \times P_E}{P_G + P_E} \]

Then the steering centre, \( O \), can be used to obtain the steering radius of the centroid and compute the centrifugal inertia force.

\[ R_i = \frac{l_1 + l_2 \cdot \cos(\alpha)}{\sin(\alpha)} \]

\[ \bar{R}_i = M_a \cdot \omega^2 \cdot \bar{R} \]

The resultant of gravity and the centrifugal inertia force is obtained as follows.

\[ \vec{G} = (M_a \cdot g \cdot \sin(\gamma), 0, M_a \cdot g \cdot \cos(\gamma)) \]

\[ \vec{F} = \vec{G} + \vec{F}_i \]

Finally, the SSM angle can be obtained by equation (25).

\[ \phi = \arcsin \left( \frac{\vec{F} \times \hat{n}}{\vec{F} \cdot \hat{n}} \right) - 90^\circ \]

The calculation process is illustrated in Figure 5.

4 Numerical analysis

The XG953 wheel loader (Xiaogong Machinery Inc., China) was used as a model to compute the SSM angle. Table 1 shows some characteristic parameters required for the calculation.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>y1</td>
<td>2,734 mm</td>
<td>z1</td>
<td>−1,681 mm</td>
</tr>
<tr>
<td>M1</td>
<td>1,740 kg</td>
<td>y2</td>
<td>741 mm</td>
</tr>
<tr>
<td>z2</td>
<td>−851.8 mm</td>
<td>M2</td>
<td>6,412 kg</td>
</tr>
<tr>
<td>y4</td>
<td>−1,952 mm</td>
<td>z4</td>
<td>255 mm</td>
</tr>
<tr>
<td>M4</td>
<td>6,903 kg</td>
<td>y3</td>
<td>−1,953 mm</td>
</tr>
<tr>
<td>z3</td>
<td>−241 mm</td>
<td>M3</td>
<td>2,518 kg</td>
</tr>
<tr>
<td>l1</td>
<td>830 mm</td>
<td>l2</td>
<td>1,958 mm</td>
</tr>
<tr>
<td>h</td>
<td>−119.5 mm</td>
<td>d</td>
<td>2,700 mm</td>
</tr>
<tr>
<td>dy</td>
<td>79 mm</td>
<td>dz</td>
<td>1,148 mm</td>
</tr>
</tbody>
</table>

4.1 Lateral stability analysis of static vehicle

To analyse the lateral stability characteristics of wheel loaders during static state, the velocity of the vehicle was set to zero. In this condition the load mass, elongation of the boom cylinder and slope angle of the floor were explored to see whether they affected the SSM angle.

The curves, illustrated in Figure 6, show the effects of the load mass and the elongation of the boom cylinder on the SSM angle in the case of a slope angle of 30°. Through calculation, it can be determined that when the elongation of the boom cylinder is 329 mm, the SSM curve is parallel to the x-axis, which means that the load mass has no effect on the SSM angle. When the load mass is constant the increase of the elongation of the boom cylinder will reduce the stability of the wheel loader.
With a load mass of 500 kg and the elongation of the boom cylinder of 200 mm, the effects of the slope angle on the SSM angle can be seen in Figure 7. With the fixed load mass and elongation of the boom cylinder, the increase of the slope angle will reduce the SSM angle. When the SSM angle becomes zero, the first order instability appears; this slope angle is defined as the maximum parking slope angle that changes according to different load mass and the elongation of the boom cylinder. As can be seen from Figure 8, the maximum parking slope angle decreases when load mass and elongation of the boom cylinder become larger.

4.2 Lateral stability analysis of driving vehicle

To analyse the lateral stability characteristics of wheel loaders during operation, the load mass, elongation of the boom cylinder and slope angle of the floor were set to fixed values. In this condition, the velocity and the steering angle of the wheel loader were explored to see how these affect the SSM angle.

As shown in Figure 9, the steering angle has almost no effect on the SSM angle when the velocity is zero. However, when the velocity increases, the steering angle will have an increasing influence on the SSM angle. Similarly, the velocity will have an increasing influence on the SSM angle as the steering angle increases.

The intersection curve of the SSM angle surface and the x-y plane in Figure 9 is illustrated in Figure 10. This intersection curve shows the relationship between the velocity and the steering angle when the SSM angle is zero. Therefore, this curve can be used as the boundary between the lateral stable state and the lateral unstable state. It can also be concluded that a higher velocity requires a smaller steering angle to keep a wheel loader laterally stable, and larger steering angle requires a lower velocity to keep a wheel loader laterally stable.
5 The virtual prototype simulation

5.1 Building a virtual prototype model

To verify the SSM angle-based lateral stability indication, a simulation process was conducted with the use of a virtual prototype built by 3D CAD software: Solidworks and ADAMS. A 1:1 solid prototype model of XG-953 was built (Figure 11). To correspond with the assumptions made before the calculation, the road friction coefficient was set to 4 in the road file (Seneviratne et al., 2009), and the stiffness and damping of the tyre were set to a sufficiently high level in the loaded tyre file.

5.2 Simulation conditions

As mentioned in Section 2.2, when the normal force on the tyre becomes zero, the tyre is off the ground and the wheel loader is unstable. So, the normal force on the tyre can be used to judge the state of wheel loaders. A fishhook curve is commonly used as steering input (Steven and Iagnemma, 2008). But in this case, the normal force on the tyre is fluctuating, which makes it difficult to judge the state of wheel loaders precisely, so a constant is used as steering input in this study. The motion path of the wheel loader will be a circle, as illustrated in Figure 12, on a radius which is related to the steering angle. When the wheel loader is moving along a circular path, the normal force on the tyre is relatively stable.

The velocity under the conditions of each given steering angle was changed until the normal force on the inside front tyre became zero. As illustrated in Figure 13, the red curve is the normal force on the inside front tyre, and the blue curve is the normal force on the inside rear tyre. It can be seen that the red curve is close to zero, which means the inside front tyre is almost off the ground; this is the critical state of rollover.

<table>
<thead>
<tr>
<th>Steering angles (°)</th>
<th>Simulation results (m/s)</th>
<th>Calculation results (m/s)</th>
<th>Relative errors</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>21.68</td>
<td>17.82</td>
<td>17.8%</td>
</tr>
<tr>
<td>10</td>
<td>13.14</td>
<td>12.62</td>
<td>3.96%</td>
</tr>
<tr>
<td>15</td>
<td>10.66</td>
<td>10.29</td>
<td>3.47%</td>
</tr>
<tr>
<td>20</td>
<td>9.17</td>
<td>8.88</td>
<td>3.16%</td>
</tr>
<tr>
<td>25</td>
<td>8.03</td>
<td>7.88</td>
<td>1.87%</td>
</tr>
<tr>
<td>30</td>
<td>7.33</td>
<td>7.12</td>
<td>2.86%</td>
</tr>
<tr>
<td>35</td>
<td>6.82</td>
<td>6.51</td>
<td>4.55%</td>
</tr>
</tbody>
</table>

5.3 The test data analysis

The maximum steering angle of XG953 is 35° known from the manual data. The experiments were conducted at every 5° to find the velocity where the normal force on the inside front tyre became zero. In each experiment, the velocity could be recorded which brought the normal force on the
inside front tyre close to zero, as shown in Table 2. The velocity and steering angle data were plotted and compared with the curve in Figure 10 (Figure 14).

The two curves are very close, and the relative error is less than 5% when the velocity is lower than 11 m/s. The minimum relative error between the calculated result and simulated result is 1.87% when the steering angle is 25°, and the velocity is around 8 km/h.

But when the velocity is higher than 11 m/s the two curves separate gradually, which indicates that the deviation between the experimental results and the calculated results increases slowly. According to gyroscope rigidity, a rotating tyre can keep its axis direction constant because of its inertia and the higher the velocity of the tyre, the greater is its inertia. However, in the previous calculation process we did not consider the influence of this inertia, which caused the deviation between the experimental results and the calculated results; the higher the velocity of the wheel loader, the greater this deviation. For example, the maximum relative error between the calculated result and simulated result was 17.8% when the steering angle was 5° and the simulated velocity was higher than 21 m/s.

Wheel loaders mostly work at low speed, for example the maximum velocity of XG953 is 10.56 m/s. Therefore, the calculated results are accurate enough within the permissible speed range.

5.4 Steady-state angle calculation considering tyre effect

The above calculation and simulation experiments were carried out on the assumption that the tyres are rigid. But the fact is, tyres are the only elastic damping elements on wheel loaders; therefore the properties of tyres have a great influence on the stability of wheel loaders. It is difficult to build a mathematical model to accurately describe how the tyres influence the stability of wheel loaders, so an indirect method was carried out to deal with this problem. Before this, it was necessary to find which kinetic parameter is most affected by the tyres, through simulation experiments.

5.4.1 Simulation experiments with normal tyre parameters

Detailed tyre parameters of the XG953 are shown in Table 3. The parameters in the tyre file were set to the values in this table, and then the same simulation experiments were carried out.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tyres vertical stiffness</td>
<td>727,000</td>
<td>N/m</td>
</tr>
<tr>
<td>Tyres vertical damping</td>
<td>2,920</td>
<td>Ns/m</td>
</tr>
<tr>
<td>Cornering stiffness</td>
<td>150,000</td>
<td>N/rad</td>
</tr>
<tr>
<td>Longitudinal slip stiffness</td>
<td>150,000</td>
<td>N/rad</td>
</tr>
<tr>
<td>Camber stiffness</td>
<td>58,200</td>
<td>N/rad</td>
</tr>
<tr>
<td>Unloaded radius</td>
<td>0.8</td>
<td>m</td>
</tr>
<tr>
<td>Width</td>
<td>0.597</td>
<td>m</td>
</tr>
</tbody>
</table>

Comparing the kinetic parameters in these two experiments, the difference between the steering radiuses was found to be greatest when the normal force on the inside front tyre becomes zero. The steering radiuses in these two experiments are shown in Table 4. The change rate of the steering radius was as high as 127% when the steering angle was 15°.

<table>
<thead>
<tr>
<th>Steering angles (°)</th>
<th>Experiments with rigid tyre</th>
<th>Experiments with normal tyre</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>10,693</td>
<td>24,273</td>
</tr>
<tr>
<td>20</td>
<td>8,046</td>
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<td>25</td>
<td>6,465</td>
<td>9,754</td>
</tr>
<tr>
<td>30</td>
<td>5,416</td>
<td>7,493</td>
</tr>
<tr>
<td>35</td>
<td>4,672</td>
<td>6,079</td>
</tr>
</tbody>
</table>

5.4.2 Indirect calculation with normal tyre parameters

Only the steering radius was taken into account. Linear fitting was completed with the data in Table 4, and the fitting result had a correlation coefficient of 0.975, which is shown in Figure 15. The fitting results were then used instead of calculated results as the steering radius to compute the SSM angle. When the SSM angle was zero, a velocity was computed in the case of each steering input. The results are recorded in Table 5; the maximum relative error between the calculated results and the simulated results was 6.65%.
Therefore, the changes of tyre characteristics cause changes in the steering radius, and the changes of steering radius affect the lateral stability of wheel loaders by influencing the centrifugal force. So, in the practical application of the SSM angle, an acceleration sensor can be added to measure the centrifugal force to deal with the complex influence of tyre properties.

6 Conclusions

In this paper, an instability threat indicator, the SSM angle, is proposed for articulated wheel loaders. When this angle is greater than zero, the wheel loader is stable; when it is less than zero, the wheel loader is unstable. The detailed calculation process is described and virtual prototyping simulations were performed. When the velocity is less than 11 m/s, the calculation data are matching with the experimental results. It is clear that the changes of tyre properties affect the lateral stability of wheel loaders by influencing centrifugal force. Therefore, when the centroid positions of the four parts of wheel loaders are known, the SSM angle can be computed by the use of the data from the sensors which can measure the load mass, the elongation of the boom cylinder, slope angle, velocity, steering angle, swinging angle and the lateral force.

Our future work will focus on real-time monitoring of the lateral stability of wheel loaders by using on-board gyroscopes.

Table 5 The relative errors between the simulated velocity and calculated velocity with different steering angles when considering tyre model

<table>
<thead>
<tr>
<th>Steering angles (°)</th>
<th>Simulation results (m/s)</th>
<th>Calculation results(m/s)</th>
<th>Relative errors</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>15.09</td>
<td>15.23</td>
<td>0.93%</td>
</tr>
<tr>
<td>20</td>
<td>11.62</td>
<td>11.49</td>
<td>1.12%</td>
</tr>
<tr>
<td>25</td>
<td>9.72</td>
<td>9.40</td>
<td>3.29%</td>
</tr>
<tr>
<td>30</td>
<td>8.59</td>
<td>8.13</td>
<td>5.36%</td>
</tr>
<tr>
<td>35</td>
<td>7.82</td>
<td>7.30</td>
<td>6.65%</td>
</tr>
</tbody>
</table>

References


