Strength Analysis of Metro Kapsul Knuckle Plate and Tie Rod using Finite Element Method

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Abstract. Metro Kapsul is a public transportation system developed to solve traffic problems and save travel time in Bandung. It is categorized as a light rail train with a capacity of 70 passengers per capsule. Before it operates, analysis is needed to ensure the vehicle’s safety. Two crucial parts to be analysed are the knuckle plate and the tie rod of the train. The knuckle plate and the tie rod must meet the requirements of the standards based on PM 175 2015. Forces on each component are calculated using equations from free body diagrams. Static and fatigue analysis are then conducted using ANSYS with Finite Element Method based on load cases in UIC Code 615-4 standard. The maximum stresses that occurred are 195.64 MPa for the knuckle plate and 118.54 MPa for the tie rod. In terms of fatigue, based on the modified Goodman failure criteria, the knuckle plate and tie rod are proven to be safe.

1. Introduction
Indonesia has a population of 237.6 million and high population growth rate, which is 1.49% every year [1]. High population growth rate leads to an increased number of land transportations especially private cars. With no significant road widening, congestion is inevitable. In order to solve this problem, an efficient form of public mass transportation is needed. Therefore, Metro Kapsul is developed. Metro Kapsul is categorized as a light rail train–its weight makes the load applied on track smaller compared to trains in general. Since the tracks are located above the road, less trestlework and smaller pillars are needed which are suitable for a crowded city.

Further analysis is needed to ensure the vehicle’s safety. Two of the most crucial components in Metro Kapsul are the knuckle plate and the tie rod, which are used to transfer forces from the bogie of the train to the wheel and allow the train to travel along corners. In conclusion, the knuckle plate and the tie rod strength analysis is needed.

2. Methodology
The forces acting on each component of Metro Kapsul when traveling along corners were calculated using equations from free body diagrams and UIC Code 615-4. By using the forces calculated, the knuckle plate strength analysis was conducted using ANSYS with finite element method based on UIC Code 615-4 and PM 175 2015.
First, the movement phases of Metro Kapsul to be analysed were defined. Three movement phases were observed in this analysis, as shown in Figure 1. The first being the phase based on UIC Code 615-4 where the train was stationary, the second was when the train was about to enter a corner, and the third was when the train was cornering and there was no movement of components relative to each other. The second and third movement phases are referred to as transient and steady state cornerings, where the calculations during these two phases are based on the lateral forces distributed along the main wheels and guide wheels of the train bogie.

Figure 1. Movement phases of Metro Kapsul.

The first phase to be analysed was the transient cornering phase. Components included in Metro Kapsul bogie are shown in Figure 2. The first step in calculating the forces during this phase was to draw the free body diagrams of these components, as shown in Figures 3 to 8.

Figure 2. Metro Kapsul bogie components.

Figure 3. Guide wheel free body diagram.

Figure 4. Guide wheel frame free body diagram.

Figure 5. Left wheel and knuckle free body diagram.

Figure 6. Right wheel and knuckle free body diagram.
From these free body diagrams, the equations needed for calculating the force on each component were obtained, as shown on Equations 2.1 to 2.7.

\[ F_1 r_1 - F_2 r_2 = I_1 \alpha_{2,A} \]  
\[ a_1 = \alpha_{2,A} r_2 \]  
\[ F_2 - F_3 = m_{tr_d} a_1 \]  
\[ F_3 r_3 - F_4 r_4 = I_2 \alpha_{4,A} + T_{f_1} \]  
\[ a_2 = \alpha_{2,B} r_3 \]  
\[ F_4 - F_5 = m_{tr_b} a_3 \]  
\[ F_5 r_5 = I_3 \alpha_{4,B} + T_{f_2} \]

To obtain the forces on each component using the equations above, some variables such as angular acceleration, linear acceleration, and torsional friction needed to be calculated first. The angular accelerations were obtained from the kinematic analysis of Metro Kapsul bogie [2] with the results shown in Table 1.

| Variable | Value     |
|----------|-----------|
| \( \alpha_{2,B} \) | 0,6245 rad/s² |
| \( \alpha_{4,B} \) | 0,4711 rad/s² |

The torsional friction was calculated by determining the contact patch between the tyre and the ground first. Figure 9 shows the vehicle tyre dimension. The contact patch length is a function of vertical deformation purely on the basis of geometry and the contact patch shape is assumed to be perfectly rectangular in shape [3], as shown in Figure 10.

The vertical stiffness was calculated by using Equation 2.8 [3]. The value of the vertical stiffness and the normal force acting on each wheel were used to calculate the maximum deflection of the tyre using Equation 2.9 [3]. The contact patch length was then calculated using Equation 2.10.

\[ K_z = 0,0274 P \sqrt{WD} + 3,38 \]  
(2.8)
\[ \Delta z = N/K_z \]  

\[ l_{cp} = \sqrt{R^2 - (R - \Delta z)^2} \]  

The equation for calculating the torsional friction was obtained by integrating Equation 2.11 derived from Figure 8. Total torsional friction was calculated using Equation 2.12. This torsional friction was later used to calculate the forces on each component using equations obtained from the free body diagrams.

\[ dT_f = \mu \cdot dN \cdot h \]  

\[ T_f = \frac{\mu NH}{4} \]  

The next phase to be analysed was the steady state cornering phase. Two assumptions were used in calculating the forces during this phase, with the first one being the total lateral force during cornering received by the tyres only, and the second one being the total lateral force during cornering received by the guide wheels only. Based on these two assumptions, free body diagrams of the whole vehicle were drawn, as shown in Figures 11 and 12.

![Figure 11. Metro Kapsul free body diagram (top view).](image1)

![Figure 12. Metro Kapsul free body diagram (front view).](image2)

For the first assumption, Figure 11 was used as a reference. The total lateral force received by the tyres was in the form of friction and the maximum possible friction force was first calculated by using Equation 2.13.

\[ F_{f \text{ maks}} = \mu N \]  

For the second assumption, Figure 12 was used as a reference. The total lateral force received by the guide wheels was in the form of normal reaction force and the maximum possible force was calculated at the critical point where the vehicle was about to roll in the opposite direction of the corner. In this condition, the pivot of the vehicle was shifted to point \( O \) and the free body diagram on Figure 12 changed, becoming the one shown on Figure 13. The forces on all guide wheels were obtained by calculating the centrifugal force on the vehicle and moment of the force about point \( O \) as shown on Equations 2.14 and 2.15.

\[ F_c = \frac{m_p v^2}{R_b} \]  

\[ \sum M = F_{1 \text{ maks}} z_2 + m_p g x_1 - F_c z_1 = 0 \]
Figure 13. Metro Kapsul free body diagram (front view) the moment before rolling.

The two forces calculated were compared to obtain the percentage distribution of the two forces acting on the Metro Kapsul bogie during cornering. This percentage was validated by using the results from previous simulation conducted using SIMPACK [4], with the results shown in Table 2. The actual forces acting on the wheels and guide wheels were calculated by multiplying the percentage by the value of the centrifugal force.

| Component          | Percentage force from simulation (%) | Percentage force from theoretical calculation (%) |
|--------------------|--------------------------------------|--------------------------------------------------|
| Wheels             | 19.20                                | 18.06                                            |
| Guide wheels       | 80.80                                | 81.94                                            |
| Relative error on wheels | 5.92                                |                                                   |
| Relative error on guide wheels | 1.41                                |                                                   |

The last part of the analysis was the calculation of forces acting on the bogie by using UIC Code 615-4. There are two major loading cases in UIC Code 615-4, exceptional loads, and normal service loads [5]. The load case in exceptional loading is a combination of vertical and lateral forces as illustrated on Figure 14. The exceptional loading case was used as the basis of force calculation to ensure that the bogie received maximum possible forces.

Figure 14. Forces acting on bogie frame based on UIC Code 615-4 standard.
These forces were calculated by using the equations based on UIC Code 615-4, as shown on Equations 2.16 and 2.17.

\[
F_Z(N) = \frac{1.4 g}{2n_b}(m_v + c_1 - n_b m_b) \tag{2.16}
\]

\[
F_y(N) = 2 \left(10^4 + \frac{m_v + c_1}{3n_e n_b}\right) \tag{2.17}
\]

The results from the calculations were summarized in the tables below, where the vertical forces and lateral forces are shown in Tables 3 and 4 respectively.

### Table 3. Summary of vertical forces.

| Component          | Transient cornering phase | Steady state cornering phase | Exceptional loading on UIC Code 615-4 |
|--------------------|----------------------------|------------------------------|---------------------------------------|
| Knuckle            | 12,806.96                  | 12,806.96                    | 13,658.46                             |
| Total (on bogie)   | 51,227.82                  | 51,227.82                    | 54,633.85                             |

### Table 4. Summary of lateral forces.

| Component                  | Transient cornering phase | Steady state cornering phase | Exceptional loading on UIC Code 615-4 |
|----------------------------|----------------------------|------------------------------|---------------------------------------|
| Front left guide wheel     | 1,073.38                  | 4,952.29                     | -                                     |
| Rear left guide wheel      | 0                         | 4,952.29                     | -                                     |
| Tie rod (on guide wheel frame) | 1,795.87              | 0                            | -                                     |
| Tie rod (on left plate A)  | 1,794.60                  | 0                            | -                                     |
| Left knuckle               | 2,957.60                  | 1,091.69                     | 5,217.58                              |
| Tie rod (on left plate B)  | 1,162.99                  | 0                            | -                                     |
| Tie rod (on right plate B) | 1,161.01                  | 0                            | -                                     |
| Right knuckle              | 1,161.01                  | 1,091.69                     | 5,217.58                              |
| Total (on Metro Kapsul)    | 48,351.85                 | 48,351.85                    | 41,740.67                             |

After calculating the forces, the strength analysis on knuckle plate and tie rod was conducted using ANSYS with finite element method based on PM 175 2015 [6] starting from the knuckle. 3D model of the knuckle plate as shown in Figure 15 was made. In this model, the knuckle, the plate, and the housing were assembled for each to be analysed based on UIC Code 615-4 that regulates the method in strength analysis testing of a bogie frame.

The plates were connected to the knuckle using welding connection. Boundary conditions and forces were applied based on UIC Code 615-4. Additional lateral force acting on the plates during the second and the third phase as shown in Figure 16 and the definition of the boundary condition is shown in Table 5.

![Figure 15. 3D Model of knuckle plate.](image1.png)

![Figure 16. Knuckle plate loading arrangement.](image2.png)
Table 5. Definition of boundary conditions.

| dx  | dy  | dz  |
|-----|-----|-----|
| 0   | 0   | 0   |

Three load cases were used for strength analysis on the knuckle plate, the first being the combination of vertical and lateral forces from the exceptional loading case on UIC Code 615-4, the second one being the combination of vertical forces based on UIC Code 615-4 and lateral forces during the transient cornering phase, and the third one being the combination of vertical forces based on UIC Code 615-4 and lateral forces during the steady state cornering phase. These forces applied on knuckle plate are shown in Table 6.

Table 6. Forces applied on knuckle plate.

| No | Parameter | Load case 1 | Load case 2 | Load case 3 |
|----|-----------|-------------|-------------|-------------|
| 1  | $F_{Ax}$ | 0           | 1,794.60    | 0           |
| 2  | $F_{Bx}$ | 0           | 1,162.99    | 0           |
| 3  | $F_{Kx}$ | 5,217.58    | 2,957.60    | 1,091.69    |
| 4  | $F_{KY}$ | 13,658.46   | 12,806.96   | 12,806.96   |
| 5  | $F_g$    | 319.28      | 319.28      | 319.28      |

After strength analysis on the knuckle plate was carried out, 3D model of the tie rod as shown in Figure 17 was made. The tie rod components were connected using ball joints and they only received lateral forces during the transient cornering phase. Boundary conditions and forces were applied based on the transient cornering phase as shown in Figure 18, and the definition of the boundary condition is shown in Table 7. Forces applied on the tie rod during the transient cornering phase can be referred to Table 8.

Figure 17. 3D Model of tie rod.

Figure 18. Tie rod loading arrangement.

Table 7. Definition of boundary conditions.

| dx  | dy  | dz  |
|-----|-----|-----|
| T   | R   |     |
| 0   | Free|     |
| 0   | 0   | 0   |

Table 8. Forces applied on tie rod.

| No | Parameter | Force (N) |
|----|-----------|-----------|
| 1  | $F_x$     | 1,795.87  |
| 2  | $F_y$     | 22.74     |
| 3  | $F_g$     | 16.27     |
Weldox 700 (ASTM A514 Grade B) is the material of knuckle plate and tie rod in which its mechanical properties are shown in Table 9. There should be no permanent deformation after the removal of exceptional loads. While PM 175 2015 only allow stress less than 75% of Yield Strength of the material [3] (Weldox 700) which is 525 MPa. All the stress results of any cases must not exceed 525 MPa. The stress result will also be plotted on modified Goodman Compression and Tension Diagram.

### Table 9. Mechanical Properties of Weldox 700.

| No | Properties         | Symbol | Value | Unit |
|----|--------------------|--------|-------|------|
| 1  | Density            | $\rho$ | 7,850 | kg/m³ |
| 2  | Yield Strength     | $S_y$  | 700   | MPa  |
| 3  | Tensile Strength   | $S_{ut}$ | 780-930 | MPa  |
| 4  | Modulus of Elasticity | $E$  | 200   | GPa  |
| 5  | Poisson Ratio      | $\nu$  | 0.3   |      |

3. Result and Discussion

The results of the strength analysis using finite element method were validated for those results to be reliable. First, meshing was done on each model, and convergence test was performed to determine the correct element size to be used in the simulation. This test was done by increasing the number of elements or decreasing the element size. After that, the results were validated by comparing the normal stress on one of the axis obtained from the simulation to the normal stress on one of the axis calculated theoretically.

3.1 Knuckle Plate

The result of convergence test on the knuckle plate is shown in Table 10 and Figure 19. As the element size decreased from 15 to 6 mm, the relative error started to fall below 10%, and the relative error fell further below 1%. Based on ANSYS Conference 2008 [7] this result was adequate, therefore the element size of 6 mm was used throughout the knuckle plate strength analysis.

### Table 10. Convergence test on knuckle plate.

| Element size (mm) | Number of Elements | Equivalent stress (MPa) | Error (%) |
|-------------------|--------------------|-------------------------|-----------|
| 15                | 24,081             | 82.71                   | -         |
| 10                | 35,636             | 93.19                   | 11.25     |
| 9                 | 42,073             | 96.98                   | 3.91      |
| 8                 | 52,347             | 90.46                   | 7.21      |
| 6                 | 97,982             | 96.74                   | 6.50      |
| 5                 | 155,051            | 96.28                   | 0.48      |
| 4                 | 283,372            | 95.96                   | 0.33      |

The forces on load case 2 were validated by calculating the normal stress on point A located on the knuckle plate where a stress probe was placed using ANSYS, as shown in Figure 20. The normal stress on the z-axis calculated from the simulation was -10.83 MPa. For the theoretical calculation, the knuckle plate was cut at point A to calculate the area moment of inertia and the normal stress as shown in Figure 21. The normal stress on point A on the z-axis calculated theoretically was -10.49 MPa. The stress calculation is shown in Table 11. Since the error between the simulation result and the theoretical calculation result is below 10%, this simulation can be considered valid based on the “Concept of Model Verification and Validation” [8].
Figure 19. Convergence test on knuckle plate graph.

Figure 20. Stress probe on knuckle plate.

Figure 21. Cross section of knuckle plate.

Table 11 Theoretical calculations on knuckle plate

| No | Parameter          | Value       | Unit |
|----|--------------------|-------------|------|
| 1  | $F_{Ax}$           | -1,794.60   | N    |
| 2  | $d_z$              | 101.63      | mm   |
| 3  | $N$                | 0           | N    |
| 4  | $V$                | -1,794.60   | N    |
| 5  | $M$                | 182,376.27  | N.mm |
| 6  | $A$                | 1,200.10    | mm$^2$|
| 7  | $l_{yy}$           | 633,624.79  | mm$^4$|
| 8  | $c$                | 36.44       | mm   |
| 9  | $\sigma_x$ (theoretical) | -10.49 | MPa |
| 10 | $\sigma_x$ (simulation) | -10.83 | MPa |
| 11 | Error              | 3.26        | %    |

The equivalent stress distribution for load case 2 on knuckle plate is shown in Figure 22. The stress calculated for each load case shown in Table 12, where all of the forces calculated are below 575 MPa.
Figure 22. Equivalent stress distribution on knuckle for load case 2.

Table 12. Stress result for all load cases.

| Load case                        | $\sigma_{eq \ max}$ (MPa) | Maximum deformation (mm) | Safety factor |
|----------------------------------|-----------------------------|---------------------------|---------------|
| 1 – UIC Code 615-4               | 188.69                      | 0.27                      | 2.78          |
| 2 – Transient cornering phase    | 195.64                      | 0.24                      | 2.68          |
| 3 – Steady state cornering phase | 164.06                      | 0.22                      | 3.20          |

From three load cases, there was a critical point on the knuckle plate as a result of each loading, as shown in Figure 23. The mean stress to ultimate tensile strength ratio and the amplitude stress to endurance limit ratio on each of these critical points from each load cases were calculated using their maximum and minimum principal stresses. These values were then plotted on the modified Goodman compression and tension diagram. The mean and amplitude stresses on each of these critical points are first calculated as shown in Table 13.

Figure 23. Three critical points from (a) load case 1, (b) load case 2, and (c) load case 3.
Table 13. Mean and amplitude stress from each load case.

| Load case                          | Stress (MPa) |
|------------------------------------|--------------|
|                                    | \( \sigma_{p_{\text{max}}} \) | \( \sigma_{p_{\text{min}}} \) | \( \sigma_{m} \) | \( \sigma_{a} \) |
| 1 – UIC Code 615-4                 | 201.99       | 17.70         | 92.44         | 109.85         |
| 2 – Transient cornering phase      | 214.21       | 12.31         | 100.94        | 113.26         |
| 3 – Steady state cornering phase   | 187.36       | 13.66         | 86.84         | 100.51         |

The value of \( S_{u_{t}} \) is 780 MPa, therefore the \( S'_{e} \) is 390 MPa. By applying a correction of 0.75 on the \( S'_{e} \) based on PM 175 2015, the calculated \( S_{e} \) was 292.5 MPa. The mean stress to ultimate tensile strength ratio and the amplitude stress to endurance limit ratio on each critical point are shown in Table 14. These values were plotted on a modified Goodman compression and tension diagram, as shown in Figure 24. From this graph, it can be seen that all of these three points fall below the ABC line, which means that no crack or failure occurred on the knuckle plate for each case.

Table 14. Mean stress to ultimate tensile strength and amplitude stress to endurance limit ratio for each load case.

| Ratio                  | Load case 1 | Load case 2 | Load case 3 |
|------------------------|-------------|-------------|-------------|
| \( \sigma_{u} / S_{e} \) | 0.3150      | 0.3451      | 0.2969      |
| \( \sigma_{m} / S_{u_{t}} \) | 0.1408      | 0.1452      | 0.1288      |

Figure 24. Mean stress to ultimate tensile strength ratio and amplitude stress to endurance limit ratio on knuckle plate plotted on modified Goodman compression and tension diagram [9].

3.2 Tie Rod
The result of the convergence test on the tie rod is shown in Table 15 and Figure 25. As the element size decreased from 15 to 3 mm, the relative error started to fall below 5%, and the relative error further fell below 1%. Based on ANSYS Conference 2008 [7] this result is satisfactory, therefore the element size of 3 mm was used throughout the tie rod strength analysis.
Table 15. Convergence test on tie rod.

| Element size (mm) | Number of elements | Equivalent Stress (MPa) | Error (%) |
|-------------------|--------------------|-------------------------|-----------|
| 15                | 849                | 23.49                   | -         |
| 10                | 1,950              | 31.90                   | 26.35     |
| 8                 | 3,531              | 23.75                   | 34.32     |
| 6                 | 7,722              | 19.26                   | 23.32     |
| 3                 | 60,785             | 18.92                   | 1.77      |
| 2                 | 203,429            | 18.77                   | 0.81      |

Figure 25. Convergence test on tie rod graph.

The forces on load case 2 were validated by calculating the normal stress on point B located on the tie rod where a stress probe was placed using ANSYS, as shown in Figure 26. The normal stress on the x-axis calculated from simulation was -2.96 MPa. For the theoretical calculation, the tie rod was cut at point B to calculate the area moment of inertia and the normal stress as shown in Figure 27. The normal stress on point B on the x-axis calculated theoretically was -3.05 MPa. The stress calculation is shown in Table 16. Since the error between the simulation and the theoretical calculation result is below 10%, this simulation can be considered valid based on the “Concept of Model Verification and Validation” [8].

Figure 26. Stress probe on tie rod.

Figure 27. Cross section of tie rod.
Table 16. Theoretical calculations on tie rod.

| No | Parameter       | Value       | Unit       |
|----|-----------------|-------------|------------|
| 1  | $F_x$           | 1,795.87    | N          |
| 2  | $F_y$           | -22.74      | N          |
| 3  | $R_x$           | 0           | N          |
| 4  | $R_y$           | 75.95       | N          |
| 5  | $M_{Rz}$        | 2,250.29    | N.mm       |
| 6  | $d_{XPT}$       | 24.95       | Mm         |
| 7  | $d_{XRT}$       | 54.95       | Mm         |
| 8  | $N$             | -1,795.87   | N          |
| 9  | $V$             | -53.21      | N          |
| 10 | $M$             | 1,355.56    | N.mm       |
| 11 | $A$             | 706.86      | mm$^2$     |
| 12 | $I$             | 39,760.78   | mm$^4$     |
| 13 | $c$             | 15          | mm         |
| 14 | $\sigma_x$ (theoretical) | -3.05 | MPa |
| 15 | $\sigma_x$ (simulation) | -2.96 | MPa |
| 16 | Error           | 2.98        | %          |

The equivalent stress distribution for load case 2 on the tie rod is shown in Figure 28. The calculated equivalent stress on the tie rod and the safety factor of the tie rod were 118.54 MPa and 4.42 respectively, where the equivalent stress was below 525 MPa. There was a critical point on the tie rod as a result of each loading, as shown in Figure 29. The mean stress to ultimate tensile strength ratio and the amplitude stress to endurance limit ratio on this critical point from load case 2 were calculated using their maximum and minimum principal stresses and then plotted on the modified and tension diagram.

Figure 28 Equivalent stress distribution on tie rod for load case 2

Figure 29 Critical point on tie rod

The value of $S_{ut}$ is 780 MPa therefore the $S_e'$ is 390 MPa. By applying a correction factor of 0.75 on the $S_e'$ base on PM 175 2015, the calculated $S_e$ was 292.5 MPa. The mean stress to ultimate tensile strength ratio and the amplitude stress to endurance limit ratio on the critical point of the tie rod are shown in Table 17. These values were plotted on a modified Goodman compression and tension diagram, as shown in Figure 30. From this graph, it can be seen that all of this point falls below the ABC line, which means that no crack or failure occurred on the tie rod.

Table 17. Ratio of Mean Stress and Amplitude Stress on Tie Rod

| Tie rod (load case 2) | $\sigma_{pmax}$ (MPa) | $\sigma_{pmin}$ (MPa) | $\sigma_m$ (MPa) | $\sigma_a$ (MPa) | $S_e$ | $S_{ut}$ |
|----------------------|-----------------------|-----------------------|------------------|------------------|------|--------|
| 166.27               | 59.57                 | 53.35                 | 112.92           | 0.18             | 0.14 |
Figure 30. Mean stress to ultimate tensile strength ratio and amplitude stress to endurance limit ratio on tie rod plotted on modified Goodman compression and tension diagram [9].

4. Conclusion

Based on the analysis, the knuckle plate and the tie rod of Metro Kapsul met the requirement on PM 175 2015 since none of the stresses calculated exceeded 525 MPa. Both components passed the failure criteria of modified Goodman for each load case. In the knuckle plate, the highest stress occurred on load case 2 and the highest deformation occurred on load case 1. The safety factors of the knuckle plate calculated using load cases 2 and 1 were 2.68 and 2.78, respectively. The safety factor of the tie rod was 4.42.

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