Journal of Scientific Research and Development 3 (2): 87-91, 2016
Available online at www.jsrad.org
ISSN 1115-7569

Prediction model of rollover threshold of a high deck bus in quasi static constant radius cornering

A.R Abdul Manap *, M. Baharom, M. Maharun

Faculty of Mechanical Department University Technology Petronas Seri Iskandar Perak Malaysia

Abstract: This paper discusses on the use of a prediction model of rollover risk threshold for a high deck bus. The model was developed using multi body system Adams/Car simulation software. The results of lateral accelerations and yaw rates of the simulated model were compared and validated against the physical measurements pertinent to the real vehicle. Multiple simulation of quasi static constant radius cornering under different types of loading distribution, bank angle and lateral force were performed. A logistic regression model was used to predict the probability of rollover risks. The result indicates that the rollover threshold of the high deck bus started at the onset of 0.3g and increased significantly at 0.4g. This study also demonstrates that the bank angle and loading distribution have major impact on the rollover threshold value of the high deck bus simulation model.

Key words: Multi body system; Quasi static constant radius cornering; Rollover threshold

1. Introduction

High deck buses have become popular for use in Malaysia by bus operators due to higher capacity of passenger load per trip. In terms of physical dimension specification, the height of a single deck bus varies between 3 to 3.5 m while a double deck bus and high deck bus can go up to 4.5m and 3.9m respectively. The other significant difference between these categories (single and multiple decks) is the seat capacity. Usually double deck and high deck buses are designed to accommodate up to 50 to 60 seat passengers while single deck is limited to 30 to 40 seat passengers.

Due to recent accidents involving high deck buses, the issue on the suitability of use of such vehicles on Malaysian roads has been raised. The hazards imposed by high deck buses are mainly due to the greater risk of rollover. This type of bus has a lower stability static factor ratio (i.e. SSF 0.6-0.75) and a higher load distribution compared to other types of heavy vehicles (Prochowski et al., 2012). In addition, high deck buses which are commonly used for transporting tourists to mountainous attractions, have a greater exposure to a higher number of horizontal and vertical road curves, thus increasing the risk of rollover accidents(Aqbal et al., 2012; Chu, 2014).

Repercussions of such incidents has raised serious concerns among the public and other stakeholders giving rise to the demand for improved bus safety especially in the use of high deck buses that ply mountainous roads in Malaysia. The Malaysian Government has set up an Independent Advisory Board that serves the Minister of Transport Malaysia to evaluate and recommend the best intervention to prevent such incidents from happening (MIROS, 2013) The Board has stipulated the following recommendations:

1. To restrict public service vehicles specifically buses and commercial vehicles from using hilly roads, and
2. To disallow double deck and high deck buses from plying hilly roads until further improvement on bus and road designs are implemented.

However, imposing a blanket restriction which involves banning high deck buses on Malaysian roads poses a great challenge for the government to enforce. As such, there is practically no existing regulations that pertain to safety guidelines on the use of high deck buses, much less about adapting any in order to assess the buses' dynamic performance on Malaysian road environment (Aqbal et al., 2012).The current existing UNECE R66 regulation relates only to structure crashworthiness of vehicle categories of large single-deck buses and double-deck buses(UNECE, 2005).

The goal of this study is to develop and to test a model in predicting the risk of rollover of a high deck bus under various operating speeds and road conditions. A simulation analysis using Multi body Software ADAMS car was proposed for this study. Specific quasi static constant radius analyses were performed under different lateral accelerations, radius of cornering, different loadings and bank angles.

2. Material and methods

The bus chosen from this study was purposively selected from a local high-deck bus operator. Its dimensions are 12.19m in length, 4.1m in height and

* Corresponding Author.

2.5m in track width. Its seat configuration comprises upper and lower deck compartments. The seat capacity at the upper deck is 35 seats while at the lower deck, it is 13 seats. The maximum front axle load and rear axle load designated for this vehicle are 7500kg and 17500kg respectively. Details are presented below (Table 1):

Table 1: Vehicle Properties

Vehicle Properties	Value	Unit
Weight (Unladen)	17050	kg
COG x†	4350.06	mm
COG y	0.19	mm
COG z	1400.00	mm
Inertia Ixx	8.880E+010	kg-mm²
Inertia Iyy	9.760E+011	kg-mm²
Inertia Izz	9.280E+011	kg-mm²

A model of high deck has been developed using the multi-body simulation software package MSC.ADAMS/Car (Abdul et al., 2015). The complete model consists of front axle suspension, drive axle and tag axle for rear suspension and also a rigid body chassis (Fig. 1).

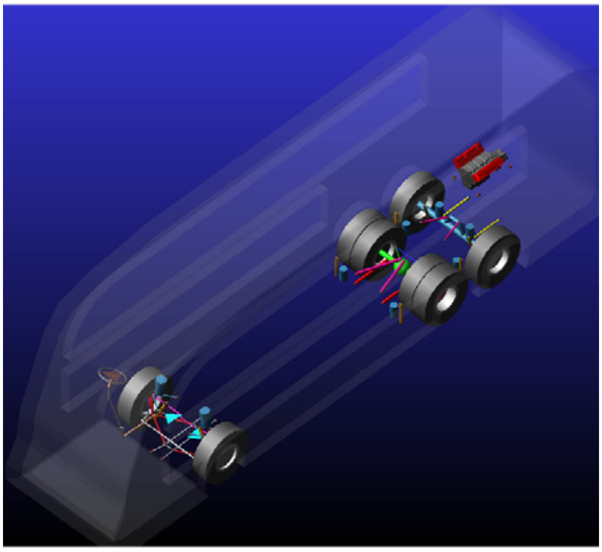

Fig. 1: Vehicle model

An experiment for the real world dynamic performance of the bus during cornering was conducted. A local highway stretching from Kuala Lumpur to Klang, Malaysia, was selected as a physical track for this study (Fig. 2). A DL2 data logger with GPS was firstly mounted on the bus chassis. Information such as longitudinal velocity, lateral acceleration and 3D X, Y and Z positions of the bus were collected during maneuvers. Information on pitch, yaw and roll angles of the bus were captured using Sensor Wave Application that utilized an embedded gyroscope in a smart phone.

For validation purposes, the data collected from the experimental study was simulated using a full vehicle analysis in ADAMS Car Standard Interface. The data obtained from the data logger was then imported as an XML file in order to produce a track map of the vehicle. The data was then analyzed and a particular road curve that exhibited relatively a constant radius curve during cornering was selected from the track.

Road Builder Function inside ADAMS Car was used to construct a virtual 3D road that specifies the X, Y and Z coordinates for the point along the path. The friction value and roadway width were set at 0.85 and 3.5m respectively.

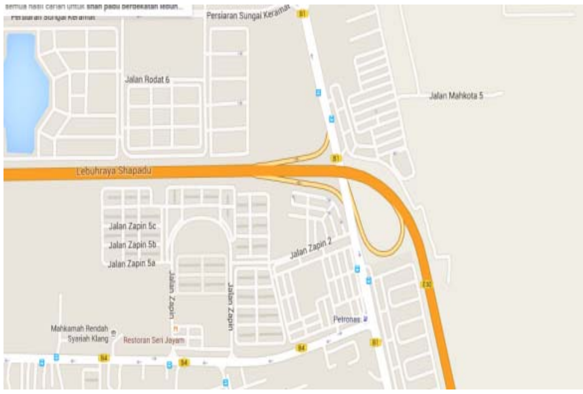

Fig. 2: Road selection

For a simplified analytical equation of rollover analysis during quasi static constant radius cornering, the load transfer model during the rollover event was governed by the following equation(Gillespie, 1992):

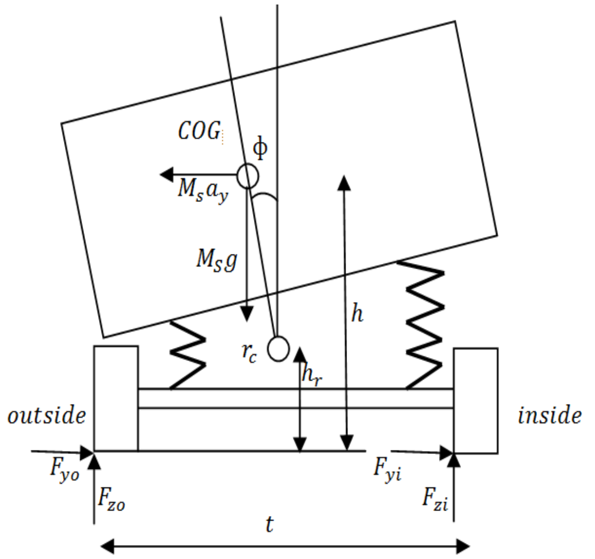

Fig. 3: Rollover Free Body Diagram

$$\sum M_o = 0 = M_s a_y h - M_s g[t/2 - \phi(h - h_r) \quad (1)$$

Where M_o is the moment at a point outside the tire wheel in contact with the ground. Roll over is assumed to occur when the normal force between the contact force (reaction force) on inside tire and ground is zero. M_s is the sprung mass of the vehicle, while $a_y$is the lateral force acting on the center gravity of mass, with h and h_r corresponding to the height of center of gravity and height of roll center. Lastly, t is the track width of tire of the vehicle (Fig. 3).

† Reference marker was selected at intersection of center front axle (origo) to the ground

Substituting $\phi = R_{\phi} a_y$ where R_{ϕ} is defined as roll rate of vehicle, and solving for lateral acceleration from equation (1) yields:

$$\frac{a_y}{g} = \frac{t}{2h}\left[\frac{1}{1-R_{\phi}(1-h/h_r)}\right] \quad (2)$$

From equations (1) and (2), it can be observed that the quasi static rollover during cornering depends on the magnitude of lateral acceleration during cornering, the ratio of track width, and the height of center of gravity of the roll center.

In order to automate multiple simulations, the use of macro in ADAMS/Car was utilized. In creating macro (cmd) command file for quasi static constant radius cornering for this analysis, the initial step is to assign load distribution of passengers. Modification of mass properties (i.e. mass and inertia tensor) for the bus model was performed by creating a macro that utilized ADAM View command *part_modify rigid_body_mass®.*

Another automated macro was created for the quasi static constant radius cornering analysis. The load distribution, lateral acceleration and road embankment were selected as the input while the normal tire force was is selected as the output of the simulation (Table 2). The values from Table 2 were as selected based on existing guidelines on road designs (JKR 1986).

Table 2: Input value for simulation

Input	Minimum Value	Maximum Value
Bank angle(degrees)	0	5
Lateral Acceleration(g)	0	0.5
Radius of Curve(m)	40	70
Load Distribution	Unladen (empty); Upper (35 passengers); Full(48 passengers)	

In this analysis, the rollover initiation was defined when the normal tire force acting on the contact patch of tire and ground became zero. The incident of roll was treated as a dichotomous variable with the outcomes of roll and no roll. In modeling the odd of dichotomous dependent variable that is categorical, logistic regression model is commonly used to measure and predict the relationship between the independent (predictor) and dependent response variables (Bollapragada et al., 2014) For this analysis the simulation model of rollover is defined with the following equation:

$$P(R) = \frac{1}{1+e^{-q}} \quad (3)$$

The parameter q is modeled as a linear function of independent variables x_(1,) x_(2,) and x_3 defined as follows:

$$q = \beta_0 + \beta_{x1} + \beta_{x2} + \beta_{x3} \quad (4)$$

Where q is a logit function, β_(0)is an interceptor, x_(1,) x_(2,) x_(3,) are model predictors and β is the coefficient associated with each predictor. Equation (4) can be rewritten e as the follows:

$$F(R) = \frac{1}{1+e^{-(\beta_0 + \beta_{x1}+\beta_{x2}+\beta_{x3})}} \quad (5)$$

The modeling was performed using SPSS software with a significant test value of $p < 0.05$.

4. Result

Prior to simulation analysis, a validation process was performed by comparing simulation results and field operation test data. The results of lateral accelerations and yaw rates of the simulated model were compared against the physical measurement. The simulation results of the model in comparison to the physical measurement are depicted in Figs. 4 and 5.

Figs show that the model correlated well both quantitatively and qualitatively with the experimental data during the first and middle part of simulation but it somehow experienced a small peak in acceleration toward the end of simulation time. Possible explanations were road disturbance coupled with deviation due to bank angle or increase in speed after leaving the curve (Gertsch and Eichelhard, 2003).

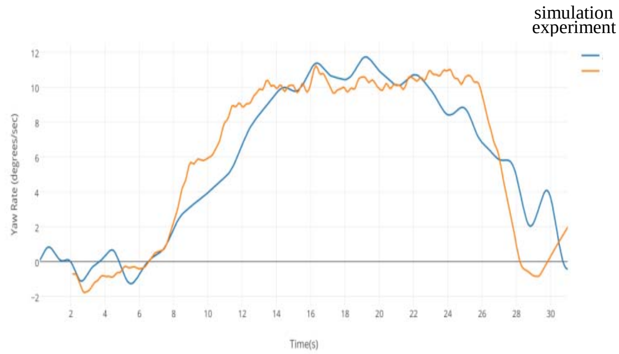

Fig. 4: Yaw Rate vs. time

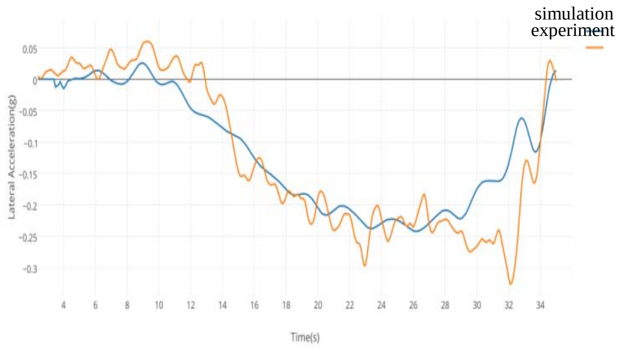

Fig. 5: Lateral Acceleration vs. time

From the simulation process, a total of 1080 quasi-static constant radius cornering analyses were performed. Out of 1080 analyses, 146 (14%) indicate an incident of rollover.

Fig. 6 shows the incident of rollover against the increase of lateral acceleration of the bus chassis. It can been seen that the occurrence of rollover is detected when the lateral force began at 0.3g and increased sharply at 0.4g and 0.5g.

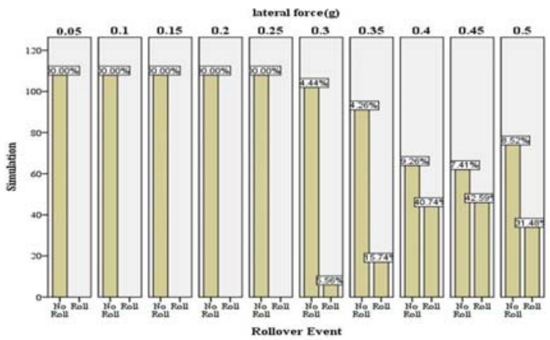

Fig. 6: Lateral force vs. rollover event

Fig. 7 shows the incident of rollover against the loading distribution of vehicle. The incident of rollover is slightly higher in an unladen vehicle (18.9%) as compared to other subcategories.

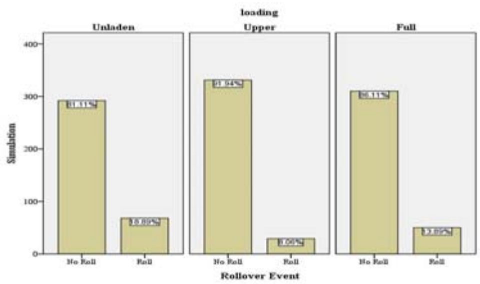

Fig. 7: Loading vs. rollover event

The rollover event against the bank angle of road was depicted in Fig. 8. The Fig. indicates that in general when the bank angle was increased, the incident of rollover was decreased. The highest incident of rollover was when bank angle was at zero degree. The Fig.s also indicates that the rollover incident was significantly reduced only when the bank angle was at 5° (6.67%).

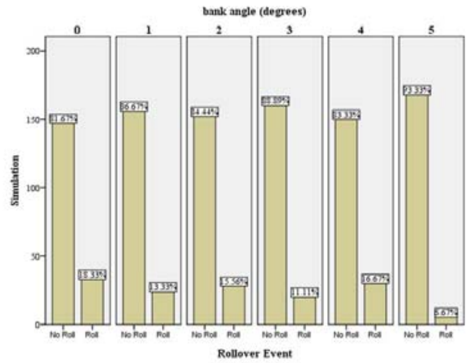

Fig. 8: Bank angle vs. rollover event

Table 3 shows the parameter estimate of type of loading, lateral force, bank angle and radius on the outcome of rollover risk. The strongest predictor of rollover is lateral force during cornering (βlateral_force =13.64). The model also indicates that the increase of mass and bank angle decreased the risk of rollover (βfull_loading=-0.776, βbank_angle=-0.177)

Table 3: Prediction risk model of rollover event

Predictor	Estimate	S.E.	Wald Z	df	*P*
Loading (unladen)			22.67	2	0.000
Loading (upper)	0.514	0.241	4.55	1	0.033
Loading (full)	-0.776	0.277	7.81	1	0.005
lateral force	13.614	1.195	129.83	1	0.000
bank angle	-0.177	0.062	8.17	1	0.004
Radius	0.017	0.012	1.87	1	0.171
Interceptor	-7.227	0.902	64.27	1	0.000

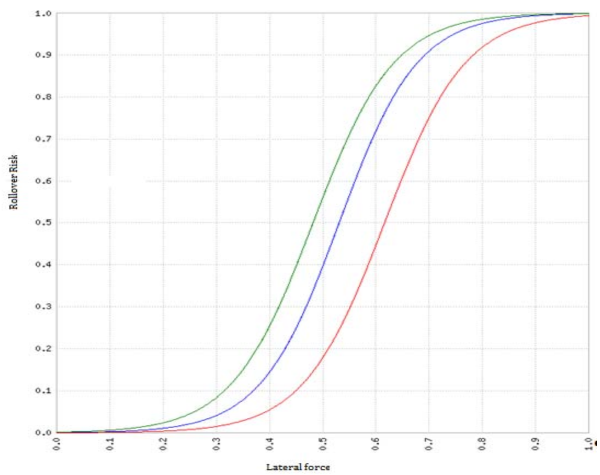

Fig. 9: Rollover risk curve against different lateral force and bank angles

$$F(x) = 1/1 + e^{-(-7.227+13.641x-0.177(0))}$$
$$F(x) = 1/1 + e^{-(-7.227+13.641x-0.177(5))}$$
$$F(x) = 1/1 + e^{-(-7.227+13.641x-0.177(-5))}$$

Fig. 9 depicts the predicted logit function when a variety of bank angles (ranging from -5° to 5°) were used for all types of loading. Bank angle shows a significant improvement in rollover threshold limit in the model. For instance, at 0.4g lateral acceleration, the risk of rollover was reduced by 10% with an increase bank angle of 5°.

5. Discussion

Since the findings of this study on one high deck bus can be highly limited, the findings were further compared to other studies of rollover thresholds on other types of commercial vehicles to obtain further insights. This study reveals that the rollover threshold of the high deck bus started at 0.3g with a 50% probability risk of rollover at approximately 0.52g. A previous study using a tractor-trailer simulation model indicates that rollover threshold was estimated approximately to be between 0.3g to 0.53g for all heights of center of gravity configuration (Gertsch and Eichelhard, 2003). Also, another study on road tanker estimated that the rollover threshold for all types of road tanker configurations at flat bank angle was approximately between 0.36g to 0.51g (Ibrahim et al., 2003). Therefore, in terms of rollover threshold range, it could be concluded that high deck buses exhibit a similar range of rollover threshold of heavy tanker trucks(Ervin, 1983).

This study also shows that the reduction of weight has the tendency to increase the risk of rollover. A study using a mathematical model involving multivariable steady state of double deck bus stability indicates a similar finding (Malviya and Mishra, 2014). The paper argues that as the vertical load of tire increases linearly with the increase of the mass of vehicle, it will therefore lead to greater stability. It is to be noted that this study is based on the assumption that rollover occurs when one of the tires is equal to zero. This conservative analysis of rollover may not reflect the actual event of rollover in which the vehicle may still be maneuvered even in the event of tire lifting. The interpretation of the results may also be only confined to quasi static analysis only. In future, more studies on dynamic stability of high deck buses are proposed.

A long term solution for a new route that is roadworthy for high deck buses may not be possible yet due to high cost of road construction. Hence, a short term solution of road improvement, such as increasing road bank angle could potentially reduce rollover risk.

6. Conclusion

This paper presents the use of a multi body system technique in investigating rollover threshold of a high deck bus. The study demonstrated that the rollover threshold of a high deck bus was at 0.3-0.4g. This study also demonstrated that the bank angle and loading distribution had a major influencing effect on the rollover threshold value.

Acknowledgment

The authors would like to be obliged to Universiti Teknologi Petronas for providing laboratory facilities and financial assistance.

References

G. Eason, B. Noble, and I. N. Sneddon (1955). On certain integrals of Lipschitz-Hankel type involving products of Bessel functions. Phil. Trans. Roy. Soc. London, vol. A247, pp. 529–551, April 1955. *(references)*

Abdul Rahmat, A. M., B. Masri, et al. (2015). "Expected Performance of a Multibody Model of a High Deck Bus in Open-Loop Steering Events " 3rd Bi-annual Post Graduate Conference, UTP.

Aqbal, H., R. Mohd Khairudin, et al. (2012). Stability of High-Deck Buses in a Rollover and Contact-Impact with Traffic Barriers. MRR 01/2012 Kuala Lumpur: Malaysian Institute of Road Safety Research.

Bollapragada, V., T. Kim, et al. (2014). "Influence of Driving Attributes on the Risk of Rollover and the Touchdown Conditions of a Sedan in Case of Corrective Maneuvers." IRCOBI Conference Proceedings.

Chu, H.-C. (2014). "Assessing factors causing severe injuries in crashes of high-deck buses in long-distance driving on freeways." Accident Analysis & Prevention **62**(0): 130-136.

Ervin, R. D. (1983). The influence of size and weight variables on the roll stability of heavy duty trucks, SAE Technical Paper.

Gertsch, J. and O. Eichelhard (2003). Simulation of dynamic rollover threshold for heavy trucks, SAE Technical Paper.

Gillespie, T. D. (1992). Fundamentals of vehicle dynamics, SAE Technical Paper.

\JKR, P. W. D. (1986). "Technical Instruction (Road) 8/86 A guide on geometric design of roads Kuala Lumpur."

Malviya, V. and R. Mishra (2014). "Development of an analytical multi-variable steady-state vehicle stability model for heavy road vehicles." Applied Mathematical Modelling.

MIROS (2013). The Independent Advisory Panel to The Minister of Transport Malaysia Report on Genting Highlands Bus Crash, Ministry of Transport.

Prochowski, L., K. Zielonka, et al. (2012). "Analysis of the process of double-deck bus rollover at the avoidance of an obstacle having suddenly sprung up." Journal of KONES **19**: 371-380.

UNECE (2005). TRANS/WP.29/1045. Concerning the common definitions of vehicle categories,masses and dimensions (s.r. 1).