Experimental Investigation of Brake Behavior for Modified Three-Axle Double Deck Bus in Thailand

Saiprasit Koetniyom[1] and Songwut Mongkonlerdmanee[2]
Faculty of Engineering[1]
The Sirindhorn International Thai-German Graduate School of Engineering (TGGS)[2]
King Mongkut's Institute of Technology North Bangkok,
Bangsue, Bangkok 10800, Thailand
Tel: 0-25870026, Fax: 0-25869541, [1] E-mail: saps@kmitnb.ac.th [2] E-mail: songwut41@hotmail.com

Abstract

The brake system is very important for vehicle design. It takes the kinetic energy, converts to heat in the braking unit for slow down and stop vehicle. Many types of vehicles are chosen for using in many advantages. Nowadays, the three axle double deck bus is popular in Thailand and There are many companies build it in Ratchaburi province. So the double deck bus has been used as the way of the intercity transportation. But in Thailand the process to build the modified bus starts from the used parts such as chassis, brake system and drive-train. For the brake system, the brake parts are over-designed based on technician experiences in order to achieve the failure of driving safety. However, every modified bus should be passed according to brake test regulation of the department of land transport in Thailand. The regulation specifies that the brake force on the axle should be over 50% on the axle load. Based on this condition and automotive engineering knowledge, the modified bus can be subjected to either instability or stability conditions, even though it passes this regulation.

For this reason, the objective of this research is to investigate the brake behavior of three axle double deck bus based on brake force distribution on each axle. Furthermore, the engine brake is taken into account for brake force distribution. As a result, the conditions of wheel lock and the maximum brake efficiency on each axle at the various coefficients of friction are revealed. The analytical calculation of engine brake for different gear positions is also illustrated to investigate the minor effect of vehicle energy to the engine and the speed behaviour.

Keywords: Three axle double deck bus, Brake force distribution, Wheel lock-up condition, Engine brake, The maximum brake efficiency, Speed behaviour**.**

1. Bus brake system

The brake system of three axle double deck bus is fully air brake system which requires more braking force than hydraulic brake system. The brake force is initiated from the brake pedal. The air brake pressure of approximately 8 kg/cm^2 from the pressure tank storage is sent to the brake chamber. Pressure in brake chamber $\boldsymbol{P_e}$ pushes the cam as shown in Figure 1. Then, the brake shoes are expanded against the inner surface of a rotating drum which is connected to a rotating wheel. The result of this action, the braking force $\boldsymbol{F_{BR}}$ is generated. This type of mechanism to generate brake force is the same in each brake wheel on the double deck bus.

1.1 Actual braking force

The braking force generated by the braking unit is called the actual braking force. The specifications of braking unit such as diameter of brake chamber and radius of the brake cam can be varied to determine the actual braking force on each axle. One example of the modified bus indicates the different parameters of the brake unit on each axle (See Table 1). However, the types of braking shoes are significant to generate internal transmission brake forces F_{BT} as shown in Table 2. The values of actual braking force F_{BR} are applied by Equation 1- 4.

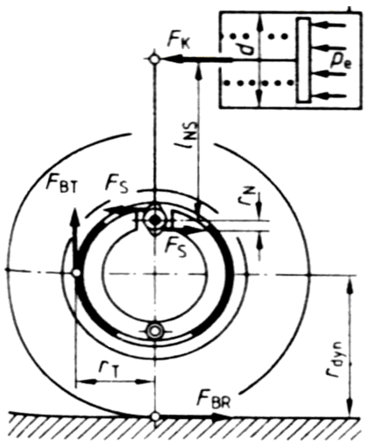

Figure 1: Drum and shoe layout

$$F_K = \frac{P_e \pi d^2}{4} \qquad (1)$$

$$F_s = \frac{F_K L_{NS}}{2 r_N} \quad (2)$$

$$F_{BT} = F_s C \quad (3)$$

$$F_{BR} = \frac{F_{BT} r_T}{R_{dyn}} \quad (4)$$

Where:
- F_K = Pressure force
- d = Diameter of brake chamber
- L_{NS} = Pressure force cantilever
- C = Internal transmission factor
- F_S = Cam force
- r_N = Distance between cam force and pivot point of cam
- F_{BT} = Internal transmission brake force
- F_{BR} = Actual brake force

Table 1: Parameters for brake chamber unit on each axle of bus [1]

Parameter	**Front axle**	**Twin axle**	**Rear axle**
Diameter of the brake chamber, *d* (m)	0.13	0.17	0.13
Pressure in the brake chamber, P_e (kg/cm^2)	8	8	8
Lever arm of the cam setter, L_{NS} (m)	0.16	0.155	0.16
Radius of the brake cam, r_N (m)	0.025	0.03	0.025
Radius of brake drum, r_T (m)	0.182	0.182	0.182
Dynamic wheel radius, R_{dyn} (m)	0.502	0.502	0.502

Table 2: Internal transmission factor C for various types of brake [2]

Type	Action	internal transmission Forward [*C*]	Internal transmission Backward [*C*]
Disc brake	single	0.8	0.8
Simplex brake	double	2	2
Duplex brake	single	3	0.9
Duo servo brake	double	4	4

1.2. Dynamic load transfer

When the accelerator pedal is released and the brake pedal is applied, extra load occurs on the front axle so called dynamic load transfer [3]. The dynamics load transfer effect induced by the longitudinal acceleration is considered in the mathematical model using the free body diagram (FBD) as shown in Figure 2.

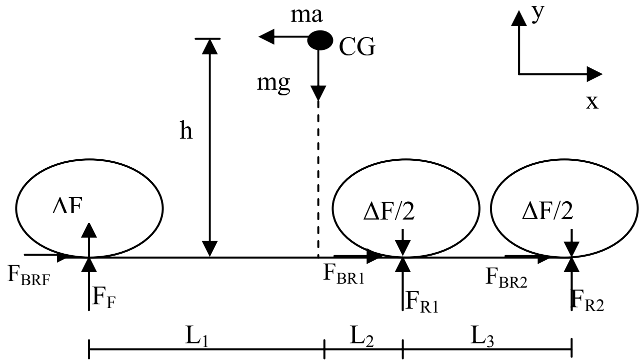

Figure 2: Free body diagram [FBD] of bus

1.3 Ideal braking force

The ideal braking force is considered to obtain the maximum brake performance while the driver applies brake pedal at various deceleration and road frictions. It is limited by dynamics load transfer, vertical loads and the road friction between tire and road surface. The parameters and values for ideal braking force on the modified double deck buses are shown in Table 3 and the equations for ideal braking force on each wheel axle are derived in the following Equations:

$$\Delta F = \frac{mah}{\left[L_1 + L_2 + \left(\frac{L_3}{2} \right) \right]} \quad (5)$$

$$F_{BRF} = (F_F + \Delta F)\mu \quad (6)$$

$$F_{BR1} = (F_{R1} - 0.5\Delta F)\mu \quad (7)$$

$$F_{BR2} = (F_{R2} - 0.5\Delta F)\mu \quad (8)$$

Where:
- ΔF = Dynamic load transfer
- F_F = Vertical load at front axle
- F_{R1} = Vertical load at twin axle
- F_{R2} = Vertical load at rear axle
- m = Equivalent vehicle mass
- a = Vehicle deceleration
- h = Height of vehicle C.G.
- $L_{1...3}$ = Lengths between axle and C.G. for the front, twin and rear wheels respectively
- F_{BRF} = Ideal brake force at front axle
- $F_{BR1..2}$ = Ideal brake force at twin and rear axles

μ = Friction coefficient

Table 3: Parameters and values of ideal braking force for modified double deck bus [1].

Parameter	Value
Distance, CG to front, twin and rear axle, L_1,L_2,L_3 (m)	4.35,1.115, 1.265
Distance, CG of high, h (m)	1.344
Vertical loads on front axle, F_F (kN)	43.949
Vertical loads on twin axle, F_{R1} (kN)	79.853
Vertical loads on rear axle, F_{R2} (kN)	42.968
Weight of bus (kN)	166.77

1.4 Friction coefficient of Road

The stopping distance of a wheel is greatly influenced by the interaction of the rotating tire tread and the road surface. The relationship between the decelerating force and the vertical load on a wheel is known as the adhesion factor. This is very similar to the coefficient of friction μ on various roads as shown in Table 4. These values are used to obtain the greatest retarding resistance in case of a correctly braked wheel force at the point of stopping.

Table 4: Typical adhesion factors for typical roads [2]

Type	Concrete and Asphalt	Tar macadam
Dry	0.76-0.85	0.58-0.62
Wet	0.48-0.52	0.38-0.42
Oily	0.35-0.40	0.25-0.30

1.5 Wheel lock condition

The locked wheel is the state that the wheel is suddenly stopped causing the slip between wheel and road. In other word, the locking condition (100% wheel slip) means that the wheel rotation is stopped while vehicle is moving. This situation occurs when there is the excess of applying brake force from the brake chamber on low friction coefficient between road and tire. Based on the brake system on modified double deck bus, the locking wheel normally occurs at rear wheel first because additional dynamic load occurs at the front wheel. Therefore, the vertical load at the front wheel is higher than that at the rear wheels. The different surfaces of road in each wheel can cause different locking conditions on wheel. To determine the wheel lock-up condition and the maximum brake efficiency, the calculated friction coefficient (μ_{brake}) from the dynamic loads and deceleration while braking can be used to compare with the friction coefficient from road (see Equation 9-11). However, there is another way to investigate wheel lock-up condition. This can be done by comparison of the actual brake force from the pressure chamber and ideal brake force from the road friction of coefficient.

$$\mu_{Brake(F)} = \frac{\max_F}{(F_F + \Delta F)} \quad (9)$$

$$\mu_{Brake(R1)} = \frac{\max_{R1}}{(F_{R1} - 0.5\Delta F)} \quad (10)$$

$$\mu_{Brake(R2)} = \frac{\max_{R2}}{(F_{R2} - 0.5\Delta F)} \quad (11)$$

Where

μbrake(F) = Friction coefficient from braking on front axle

μbrake(R1) = Friction coefficient from braking on twin axle

$\mu_{brake(R2)}$ = Friction coefficient from braking on rear axle

x_F = Brake proportion on front axle

x_{R1} = Brake proportion on twin axle

x_{R2} = Brake proportion on rear axle

a = Deceleration from braking

m = Equivalent vehicle weight

1.6 Engine brake

A moving vehicle possesses kinetic energy, whose value depends on the weight and vehicle velocity [4]. The engine provides this energy in order to accelerate the vehicle from a standstill to a given speed as shown in Equation 12. While the driver suddenly releases the acceleration pedal, the engine can provide the retardation work together with the rolling resistance work from the drive train system. Therefore, the work from the action of engine and rolling resistance is determined by the values of deceleration which affects the speed reduction as shown in Equation 13. However, this equation can represent the action from engine which has a major effect comparing to the rolling resistance.

$$KE = 0.5mv^2 \quad (12)$$

$$W_E = mas \quad (13)$$

Where

KE = Kinetic energy of vehicle

W_E = Work from engine pump and rolling resistance

a = Deceleration of vehicle

v = Vehicle velocity

s = Distance of vehicle during the engine brake

2. Standard and test for bus brake system in Thailand

According to the criteria of the department of land transport in Thailand on December 9, 2005, the criteria of the brake performance testing for service brake was described [5]. Some of them are shown as following:

(a) The suddenly response time of service brake should be less than 5 second, when it is actuated.

(b) The total braking force should be more than 50 percents of Gross Vehicle Weight Rate [GVWR] for general vehicles such as city car, van, and passenger car. But in case of bus and trailer the braking force is not less than 50 percents of weight on individual axle.

3. Methodology for investigation of double bus brake performance

To achieve this objective, the brake performances such as distance, time and deceleration from the double deck bus are considered by using the brake performance testing equipment (VC3000). Before testing, it was firstly installed at the windshield (See Figure 3a). Then, the foot brake sensor is located at the brake pedal in order to detect the actuating force referred in the black circle (see Figure 3b). In this research, there are two procedures as follows:

3.1 Engine brake test

To perform this test, the vehicle is accelerated to the maximum speed on the given gear position and then the acceleration pedal is suddenly released. It focuss on the deceleration behaviours and the speed reduction of the low gear position (3^{rd}) and the high gear position (6^{th}).

3.2 Wheel lock-up test

The dry concrete surface is only used to perform this procedure. Based on International standard regulation, the maximum speed before applied brake is set at 32 km/hr before applying brake pedal [6]. To obtain the wheel lock-up condition, there are two types of brake applications: Normal and emergency brake conditions. Thus, the results of the maximum deceleration of these conditions are investigated.

(a) (b)

Figure 3: (a) Installation of VC3000 at the windshield (b) the foot brake sensor

4. Investigation of brake force distribution on each axle

Since the brake force distribution of three axle double deck bus is rigid brake force distribution, or can not control the proportion of braking forces (brake ratios), the results of actual braking force can be calculated from the brake chamber geometry for each axle. These results are illustrated in Table 5. It reveals that the twin axle wheel can generate the actual braking force more than other axles. Its value relies on the specification of braking unit such as diameter of brake chamber, radius of the brake cam (see Table 1) and type of drum brake referred in Table 2 (see the duo servo brake). Furthermore, the total braking force on each axle according to standard in Thailand part (b) is rechecked with the actual brake force on this bus by calculation of 50% vertical load on each axle. These results are shown in Table 6. It seems that the actual brake forces from maximum pressure at the brake chamber can generate forces on each axle more than the standard requirement in Thailand. Although, actual brake force on each wheel axle is high, the wheel lock-up condition which depends on the road friction coefficient can occur. To investigate this condition, comparison between the actual brake force and the ideal brake force from the dynamics load on each axle while the driver is applying the brake pedal at various deceleration conditions is considered from Equation 6-8. Thus, the ideal force on each wheel axle can be illustrated in Table 7.

Table 5: Results of calculated actual braking force on each axle

Item	Front axle	Twin axle	Rear axle
Output force of the piston rod, F_K (kN)	10.41	17.81	10.41
Clamping force at the cam, F_s (kN)	33.33	46.02	33.33
Brake force at the brake drum, F_{BT} (kN)	133.33	184.08	133.33
Brake force at the periphery of wheel, F_{BR} (kN)	96.68	133.47	96.68
% Brake proportion on front, twin and rear axle (x_F, x_{R1}, x_{R2})	30	40	30

Table 6: Total braking force comparison on each axle between standard requirement and the modified double deck bus

Case	Total braking force (kN)		
	Front	**Twin**	**Rear**
Thailand standard regulation	22	40	22
Wheel Brake Force from Max. Pressure	96.68	133.47	96.68

From Table 6, the total braking force on each axle is calculated from maximum pressure in the brake chamber. To compare this value with the friction force with the friction force at wheel in Table 7, it is revealed that maximum pressure can generate more deceleration than the friction of road surface. Therefore, all wheels are locked because of the excess braking force. Alternatively, the coefficient of friction from the calculation in Equation 9-11 is more than the coefficient of road surface.

Table 7: The results of calculated ideal braking force on each axle

$a (m/s^2)$	ΔF (kN)	F_{BRf}(kN)	F_{BR1} (kN)	F_{BR2} (kN)
0	0	0	0	0
1	3.74	4.77	7.79	4.10
2	7.49	10.28	15.22	7.84
3	11.24	16.55	22.27	11.20
4	14.98	23.57	28.94	14.18
5	18.73	31.34	35.24	16.80
6	22.48	39.86	41.16	19.03
7	26.23	49.12	46.71	20.89
8	29.97	59.14	51.89	22.38
9	33.72	69.90	56.69	23.49
10	37.47	81.422	61.11	24.23

In Figure 4, 5 and 6, the relationship of brake force distribution in this bus brake system design is shown. For Ideal brake force, the relation can be achieved by using Equation 6-8 with various friction coefficients. Furthermore, the relation for actual brake force can be done by using proportion of brake force as shown in Table 5. From the results of study in Figure 4, it is found that the actual braking force intersects with the ideal braking force resulting in maximum brake efficiency (optimum point of design). In the same time, the area after the maximum brake efficiency is a danger due to lock wheel condition which depends on the friction of road surface. However, this optimum point can be increased at higher braking force by reducing the braking force on twin axle comparing to the front axle force. In Figure 5 and 6, the actual braking force line is above the ideal braking force line, resulting in the risk of lock wheel condition at higher braking force with low friction coefficient. To avoid this condition, the rear braking force (F_{BR2}) should be reduced in comparison with twin and front braking forces. Thus, the optimum point of design can occur at higher braking force. Therefore, the proportional braking force should be the highest at the front wheel and the lowest at the rear wheel in the three-axle double deck bus. However, design of the brake ratio on each axle must be considered based on bus geometry and the real friction coefficient of road in which the wheel lock-up condition mainly affects to the driving safety and the brake performance. Nowadays, the modern buses utilize the anti lock brake system (ABS) for solving the wheels lock-up, even though the proportional braking forces on each wheel are designed.

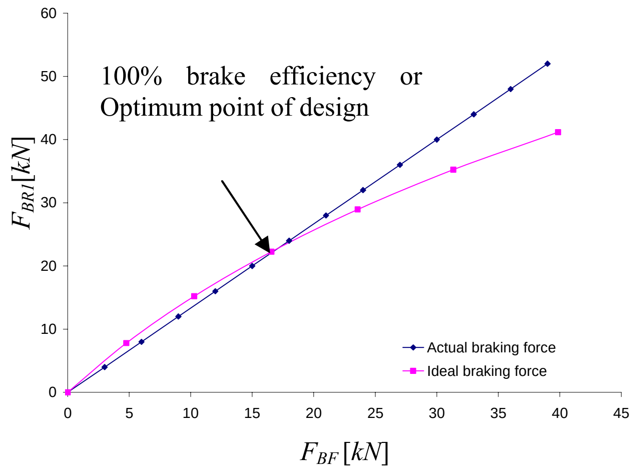

Figure 4: Relation of brake force distribution between front and twin axles

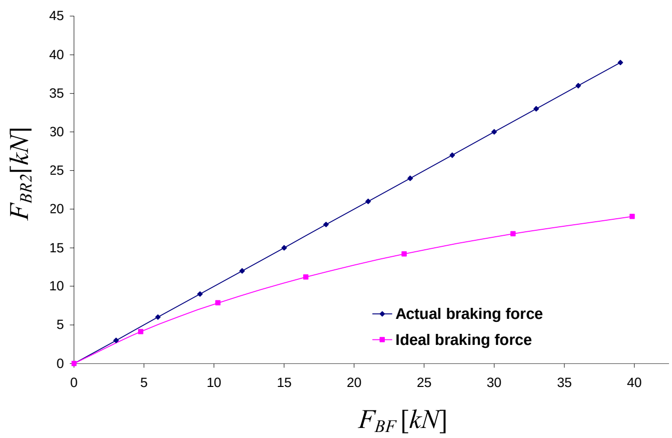

Figure 5: Relation of brake force distribution between front and rear axles

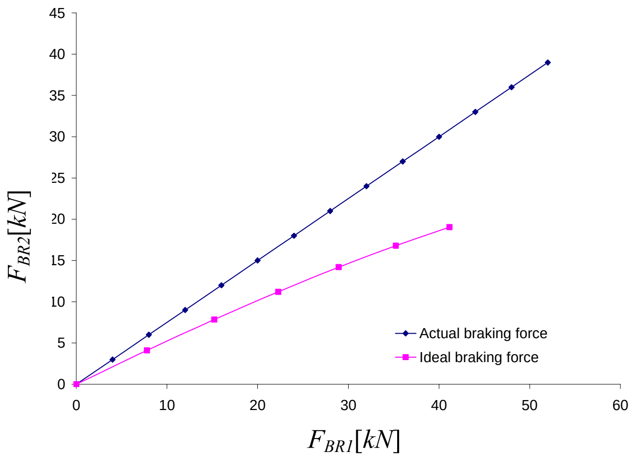

Figure 6: Relation of brake force distribution between twin and rear axles

5. Investigation of engine brake

For the engine brake test, vehicle is driven to the given speed and then is kept in (3^{rd}) low and (6^{st}) high gear positions for each test without pressing acceleration pedal. Furthermore, the speed behaviour and the work from traction of engine are determined at various decelerations on the different gear position which is described in Figure 7. The experimental results are used to calculate the kinetic energy and work from traction of engine (See Equation 12-13) and shown in Table 8.

Table 8: Engine brake test

Case		Low gear position		High gear position	
Max. speed (km/hr)		23.2		14.7	
Max. deceleration (m/s^2)	Bus travel distance (m)	2.37	5.56	1.24	2.9
Work from traction of engine (kJ) for mass factor: 1 and 1.5 respectively		224	337	61.4	92.1
Kinetic energy at Max. speed (kJ)		352.70		146.0	
Energy absorbed by Engine for mass factor 1 and 1.5 respectively		63%	95%	42%	63%

Table 9: Experimental results of wheel lock-up test at the dry concrete surface

1 = Emergency braking force

2 = Normal braking force

Case	Max. deceleration (m/s^2)	Friction from braking (μ_{brake})			Wheel (lock/not lock)		
		front	twin	rear	front	twin	rear
1	5.65	0.44	0.53	0.77	not	not	not
2	2.47	0.23	0.22	0.39	not	not	not

From Figure 7, the vehicle speed in the low gear position decreases faster than that in the high gear position because of high inertia moment of rotating part. Furthermore, it is found that both vehicle response in low and high gear positions providing two different velocity gradients due to engine speed effect. In addition, the work from vehicle kinetic energy can be highly absorbed by the traction of engine through transmission system in low gear position with high mass factor as shown in Table 8. They are compared with the kinetic energy at maximum speed as shown in table 8. However, the work from traction of engine is not enough to sudden stop.

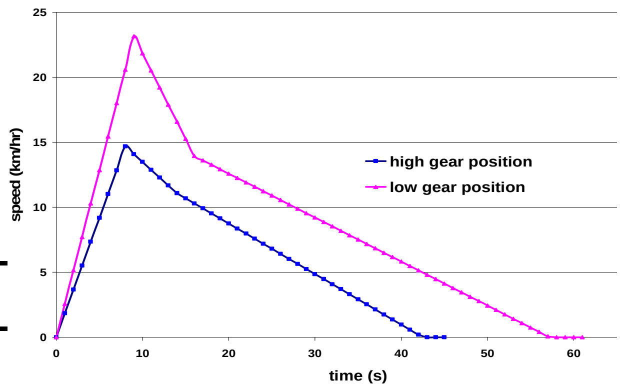

Figure 7: Speed behaviour during engine brake tests

6. Investigation of wheel lock-up and the maximum brake efficiency on each axle

From the wheel lock-up test, the bus was tested at dry concrete surface only and has a friction coefficient approximately 0.8 (see Table 4). In experimental results, the maximum deceleration of normal and emergency cases and the friction from braking (μ_{brake}) on each axle according to Equations 9-11 are shown in Table 9. It reveals that there are no wheel-lock conditions on each wheel based on comparison with the friction coefficient of dry concrete surface.

Based on Equation 9-11, the brake friction coefficient can be plotted against the deceleration as shown in Figure 8. It shows that the rear axle is on the verge of lock in which it is similar to the experimental results (coefficient of Friction: 0.77 and deceleration: 5.65 m/s^2) in emergency case. On the front axle, the requirement of friction coefficient is determined by the brake system and dynamic load. In addition, lmost drivers slightly apply a brake for driving safety and comfort. Therefore, the brake system design for this double deck bus is required to the maximum brake efficiency at a lower deceleration. In case of twin axle, the vertical loads on the twin axle are the highest. Thus, it is very dangerous if the twin axle are locked. This condition will cause out-of-control condition (unstable). Therefore, the brake system for this bus is designed to have the wheel lock-up condition at the higher deceleration. On the rear axle, although it has no maximum brake efficiency but it can support the brake system according to the regulation. Therefore, the wheel lock-up condition is limited from the friction between tire and road surface at that time and geometry of brake system determined by brake bus design.

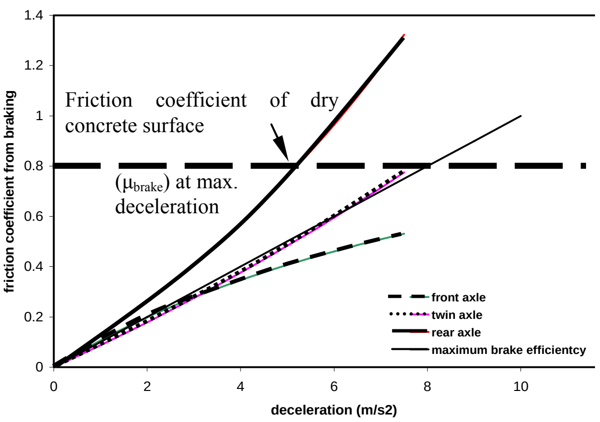

Figure 8: Friction coefficient from braking at various decelerations

7. Conclusion

The brake force distribution for the modified three axle double decker bus is constant or can not control braking force from braking unit. This force is called actual braking force. At the same time, when the driver applies brake pedal, generating dynamics load transfer in the front axle. This requires amount of braking force in the front axle more than that in other axles. This force is called ideal braking force.

The wheel locking condition can be defined by comparison between the required friction coefficient while braking and friction coefficient of road surface. Alternatively, the quantity of required braking force on individual axle is compared with the friction generated braking force from road surface. From the experimental results, it reveals that all wheels are not lock-up because friction coefficient while braking is less than friction coefficient of road surface. For the engine brake test, vehicle velocity is slower at the low gear position than high gear position. It can be decreased or slowed down the speed of the bus. But, it is unable to stop vehicle because of the traction from engine is less than the values of kinetic energy.

The maximum brake efficiency occurs in low deceleration front axle because the front axle is determined by the dynamic load transfer for driving safety and comfort. At the same time, the twin axle requires maximum brake efficiency at igh deceleration, determined by wheel lock-up condition. Finally, the calculated results obtained from the dynamic load transfer and road friction coefficient can be used to predict the brake behavior of the double deck bus based on comparison with the experimental results.

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