

Aerodynamic stability of road vehicles in dynamic motion

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ABSTRACT: A method for dynamic coupling simulation of flow and vehicle motion was developed based on large eddy simulation technique with moving boundary methods. The method was applied to investigate the aerodynamic stability of vehicle under a transient driving situation. A coefficient to quantify the aerodynamic damping was defined. For the sedan-type, simple body models investigated, the underbody provides the highest proportion of aerodynamic damping. However, it is the trunk deck contribution that causes the different damping magnitudes in the models with distinct A- and C-pillar geometrical configurations.

KEYWORDS: Aerodynamics, transient, LES, vehicle, stability, damping, pitching.

1 INTRODUCTION

In principle, automotive aerodynamics comprises the drag, lift, and side force coefficients, in conjunction with the rolling, yawing, and pitching moment coefficients. In real-world situation, the aerodynamic forces and moments which act on a vehicle are of transient nature. However, development of vehicle aerodynamics to date has mainly been focused on the steady-state components, particularly the drag coefficient, C_d . This coefficient can only be used to evaluate performances related to fuel efficiency and top speed, and provides no indication in regard to the vehicle's performance in terms of stability.

To consider the stability factors under the effect of transient aerodynamics, several assessment methods have been proposed in the literature. These methods rely on either drive test (e.g. Howell and Le Good, 1999; Okada et al, 2009) or wind tunnel measurement (e.g. Aschwandten et al, 2006). The former can only be performed after a development mule is produced, while the latter requires a complex test rig to manipulate the vehicle motion for a dynamic assessment. In addition, due to limited numbers of probe that can be attached to the test vehicle without altering the surrounding flow, drive test and wind tunnel measurement provide very limited flow information about the test. The lack of flow information could impede detailed flow analysis which is needed for identifying the underlying mechanism.

To overcome these limitations, thus the main objective of the present study is to develop a numerical method for the assessment of vehicle aerodynamic stability performance under a transient driving situation. The method allows manipulation of vehicle body motion during flow simulation, and quantification of vehicle stability performance on the basis of the aerodynamic damping generated by the vehicle, which is depending on the vehicle's body shape configuration.

2 SIMPLE BODY MODELS

Okada et al (2009) reports that the main differences between the upper body geometry of vehicles with different pitching stability characteristics lie in the A- and C-pillar shapes; the lower stability vehicle has a more angular A-pillar and rounder C-pillar configurations. Hence, to emphasize these differences, the present study creates two simple body models with opposite A- and

C-pillar geometrical configurations. In particular, the model that represents the lower-stability sedan adopts an angular A-pillar and rounded C-pillar configurations. Whilst, the model represents the higher-stability sedan adopts the opposite configurations. For convenience of discussion, the model represents the sedan of lower stability is designated “model A”, and the one represents the higher stability sedan is “model B”, respectively, hereafter (see Figure 1(a)). In general, both models are the 1:20-scale, simple bluff-bodies with same height h , width w , and length l measurements of 65, 80, and 210 mm, respectively. The models have A- and C-pillars with the same slant angles of 30° and 25° , respectively, which are based on the configurations of real vehicle.

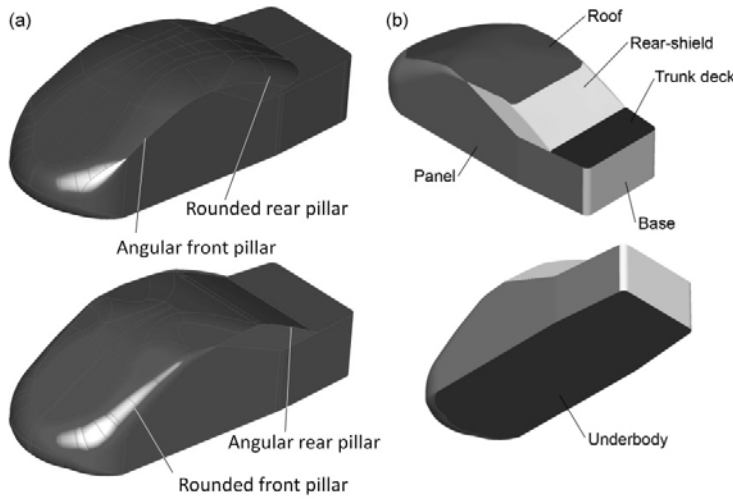


Figure 1. Simple body models: (a) model A (Top) and model B (Bottom); (b) Designations of body part.

3 NUMERICAL METHODS

3.1 Governing equations

The LES solves the following spatially filtered continuity and Navier-Stokes equations:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial}{\partial x_j} \bar{u}_i \bar{u}_j = -\frac{\partial \bar{P}}{\partial x_i} + 2 \frac{\partial}{\partial x_j} (\nu + \nu_{SGS}) \bar{S}_{ij} \quad (2)$$

$$\bar{P} = p / \rho + (\overline{u_i u_j} - \bar{u}_i \bar{u}_j) / 3 \quad (3)$$

where u_i , p , ρ , ν and S_{ij} are the i -th velocity component, pressure, density, kinematic viscosity and strain rate tensor. The over-bar denotes a spatially filtered quantity. Meanwhile, the standard Smagorinsky model is adopted to model the subgrid-scale (SGS) eddy viscosity ν_{SGS} of Eq. (2). The simulation software is an in-house CFD code, namely, the “FrontFlow/red-Aero”. Originally, the researchers develop the code for the “Frontier Simulation Software for Industrial Science” project. Then, Tsubokura et al. (2009a) optimizes it for vehicle-aerodynamics simulation. Tsubokura et al. (2009b) has validated the code on the basis that the results obtained with the code compared favorably with the wind-tunnel measurements on the pressure distribution along

the centerline of ASMO model and the flow field around a full-scale production car (including a complicated engine room and under-body geometry).

3.2 Discretization

The governing equations were discretized by using the vertex-centered unstructured finite-volume method. We adopted the second-order central differencing scheme for spatial derivatives, and exploited the blending of 5% first-order upwind scheme for the convection term for numerical stability reason. Meanwhile, pressure-velocity coupling was preserved by the SMAC (simplified marker and cell) algorithm.

For time advancement, the LES adopts the Euler implicit method. This is because an implicit scheme can accommodate larger time difference than an explicit one without causing numerical instability, especially in the case of a vehicle simulation in which the velocity and mesh size vary strongly. With larger permissible time difference ($\Delta t = 1 \times 10^{-5}$ s), the scheme needs lesser time steps (hence, shorter simulation time) to obtain a reliable time- and phase-averaging statistic. Such feature is important in dynamic LES cases, because they normally need over hundred thousand of time steps to obtain an adequate phase-averaging statistic. In the present study, the computations took about 50,000 simulation steps (over five pitching cycle) to reach a stable periodic condition and the subsequent 150,000 steps to covers 15 cycles of pitching oscillation for obtaining an adequate phase-averaging statistic.

3.3 Computational domain and boundary conditions

The computational domain resembles a rectangular wind-tunnel test section. Its cross section covers $1.52l$ on both sides of the model and height of $2.23l$. This set-up produces a small blockage ratio of 1.53%, which is well within the typically accepted range of 5% in automotive aerodynamic testing (Hucho and Sovran, 1993). The model was situated near the domain floor at a ground clearance of $0.071l$. The inlet boundary was located $3.14l$ upstream, while the outlet boundary was $6.86l$ downstream.

At the inlet boundary, the air flow approaches at a constant velocity of 16.9 m/s, corresponding to Re of 2.3×10^5 (based on vehicle length l). Meanwhile, a zero-gradient condition is imposed at the outlet boundary. The ceiling and side boundaries of the domain were treated with free-slip wall-boundary condition. The ground surface is divided into two zones. The upstream zone (which covers $3l$ from the inlet boundary) is defined as a free-slip wall condition to avoid boundary-layer formation. This setting simulates the wind-tunnel experimental condition, thus ensure the consistency of flow condition between the LES and wind tunnel test so that direct comparison between their results is allowed during validation. The remaining ground and vehicle surface are treated with the logarithmic-law ($y^+ > 11.63$) or linear law functions ($y^+ < 11.63$) depending on the obtained y^+ values. We have found that the very fine spatial resolution adopted produces the $y^+ < 4$ around the vehicle surface, thus the estimation is by the linear law function, which corresponds to the no-slip wall condition.

错误! 未找到引用源。 FORMULATION OF AERODYNAMIC DAMPING COEFFICIENT

4.1 Periodic-pitching-oscillation condition

By employing the ALE technique, we imposed a periodic pitching oscillation on the models during flow simulation to probe their dynamic response. The axis of rotation is located at the lower part of the front section of model at $0.821l$ from the rear end, corresponding to the front-wheel axle of a real vehicle. This setting is in accordance with the road-test results of Okada et al.

(2009), in which the notchbacks experienced more significant ride-height fluctuation at the back than the front. Hence, the models were rotated in a manner that simulates the rear-ride height fluctuation of the real vehicles. The pitch angle θ of the models is defined as:

$$\theta = \theta_0 + \theta_1 \sin \varphi(t), \quad \varphi(t) = 2\pi f_p t \quad (4)$$

By setting θ_0 and θ_1 equal to 2° , the vehicle models were forced to oscillate at amplitude of 2° . Although this value is larger than the range a vehicle would encounter under normal driving conditions, it has the advantage of producing more distinct aerodynamic damping effect in vehicles of different stability characteristics. Thus make it easier to interpret the underlying physical mechanism. Frequency f_p was 10 Hz, which corresponds to a Strouhal number (St) of 0.13, normalized by l and U_{inlet} . This value was chosen in consideration of the St of 0.15 obtained by road test by Okada et al. (2009). Figure 2 shows the sign convention for aerodynamic pitching moment M and angle θ .

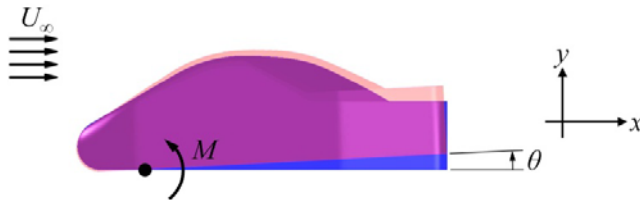


Figure 2. Sign convention for M and θ .

4.2 Periodic-pitching-oscillation condition

The estimated phase-averaged pitching moment $\langle M \rangle_p$ can be decomposed into steady and unsteady components. The equation for phase-averaged pitching moment $\langle M \rangle_p$ in terms of pitch angle θ is given as the following expansion:

$$\langle M \rangle_p = C_0 + C_1 \theta + C_2 \dot{\theta} + C_3 \ddot{\theta} \quad (5)$$

where, respectively, the single dot and double dots in the third and fourth terms indicate the first and second derivatives with respect to time t . Both C_0 and C_1 are static components; the former denotes the pitching moment M at zero pitch, while the latter describes the quasi-static behavior by taking into account the pitch-angle variation in a static manner. C_2 is associated with aerodynamic damping, and C_3 is an added moment of inertia that is proportional to angular acceleration.

Substituting Eq. (4) into (5) and rearranging gives:

$$\langle M \rangle_p = (C_0 + C_1 \theta_0) + \left[C_1 - (2\pi f_p)^2 C_3 \right] \theta_1 \sin \varphi(t) + 2\pi f_p \theta_1 C_2 \cos \varphi(t) \quad (6)$$

The above equation can then be rewritten by using new parameters, namely, M_{stat} , M_{dis} and M_{ang} as

$$\langle M \rangle_p = M_{\text{stat}} + M_{\text{sin}} \sin \phi(t) + M_{\text{cos}} \cos \phi(t) \quad (7)$$

where, M_{stat} is a constant, which set the baseline for the $\langle M \rangle_p$. M_{dis} is the amplitude of the term which in-phase with the imposed pitching displacement, and M_{ang} is the amplitude of the term in-phase with the angular velocity.

4.3 Definition of aerodynamic damping coefficient

During one pitching cycle, time t varies from 0 to $2\pi/\omega$. Hence, the work done by the aerodynamic pitching moment M on the vehicle model during one pitching cycle is:

$$W = \frac{1}{\omega} \int_0^{2\pi} M \frac{d\theta}{dt} d(\omega t) \quad (8)$$

Substituting Eq. (7) and (10) into eq. (11), the work done during one pitching cycle becomes

$$\begin{aligned} W = & M_{\text{stat}} \theta_1 \int_0^{2\pi} \cos \omega t d(\omega t) \\ & + M_{\text{dis}} \theta_1 \int_0^{2\pi} \sin \omega t \cos \omega t d(\omega t) \\ & + M_{\text{ang}} \theta_1 \int_0^{2\pi} \cos \omega t \cos \omega t d(\omega t) \end{aligned} \quad (9)$$

The first and second integrals yield the value zero, and that the third one is π . Hence, the net work per pitching cycle is:

$$W = M_{\text{ang}} \theta_1 \pi \quad (10)$$

The result of the integration reveals that the net work done on the vehicle by aerodynamic pitching moment M over a pitching cycle is depends on the component in-phase with the angular velocity M_{ang} . In Eq. (10), θ_1 and π are given. Hence, the parameter M_{ang} reflects the dynamic response of the vehicle. This parameter can be presented in a non-dimensional form. If normalized in a similar manner to the pitching-moment coefficient, it becomes:

$$C_{\text{AD}} = \frac{M_{\text{ang}}}{\frac{1}{2} \rho U_{\infty}^2 A l_w} \quad (11)$$

where ρ , U , A , and l_w are fluid density, mainstream velocity, vehicle frontal area, and wheel-base. It depends on the sign of C_{AD} , a negative value implies a tendency for aerodynamics to damp the pitching oscillation, whereas a positive value enhances the vehicle motion (i.e. negative damping). The coefficient thus enables quantitative evaluation of vehicle stability; therefore, it is termed “aerodynamic-damping coefficient.”

5 RESULTS AND DISCUSSIONS

5.1 Aerodynamic damping coefficient of simple body models

Figure 3 shows the $\langle M \rangle_p$ as a function of phase angle φ for models A and B. The coefficients in Eq. (7) are obtained by fitting the equation to the $\langle M \rangle_p$ data set by nonlinear least squares regression. Solid lines in Figure 3 are the fitted functions for the two models. Table 1 summarizes the corresponding C_{AD} for comparison. As shown in the table, the aerodynamic damping coefficient C_{AD} for the two models are negative, implying a tendency to resist the pitching motion. Between them, however, model B has a higher aerodynamic-damping coefficient C_{AD} , by about 37%. This finding is consistent with the fact that model B was created on the basis of the pillar-shape configurations of real vehicle with higher stability.

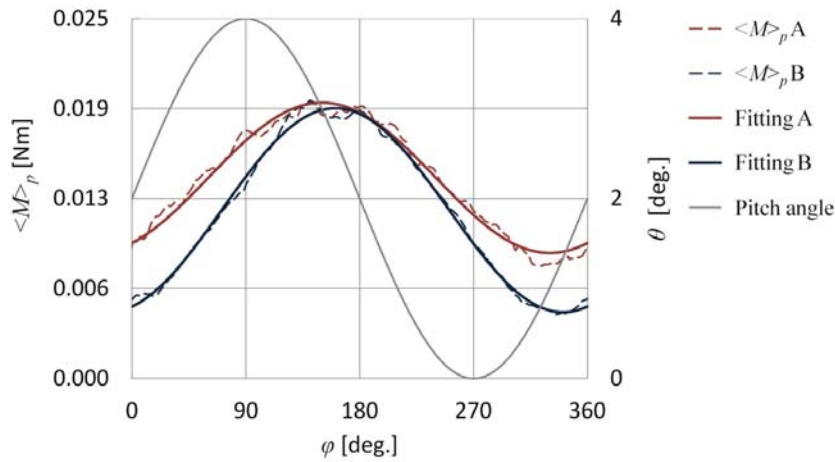


Figure 3. Phase-averaged M and fitted functions.

Table 1. Aerodynamic damping coefficient C_{AD}

Body part resignations	Model A	%	Model B	%	diff	%
Underbody	-0.0309	78.6	-0.0347	60.6	-0.00376	21.0
Trunk deck	-0.00205	5.2	-0.0100	17.5	-0.00795	44.4
Rear-shield	-0.000350	0.9	-0.00709	12.4	-0.00674	37.7
Roof	-0.00658	16.7	-0.00607	10.6	0.000513	-2.9
Base	0.000162	-0.4	0.000128	-0.2	-0.0000342	0.2
Panel	0.000376	-1.0	0.000453	-0.8	0.0000769	-0.4
Overall	-0.0394	100.0	-0.0573	100.0	-0.0179	100.0

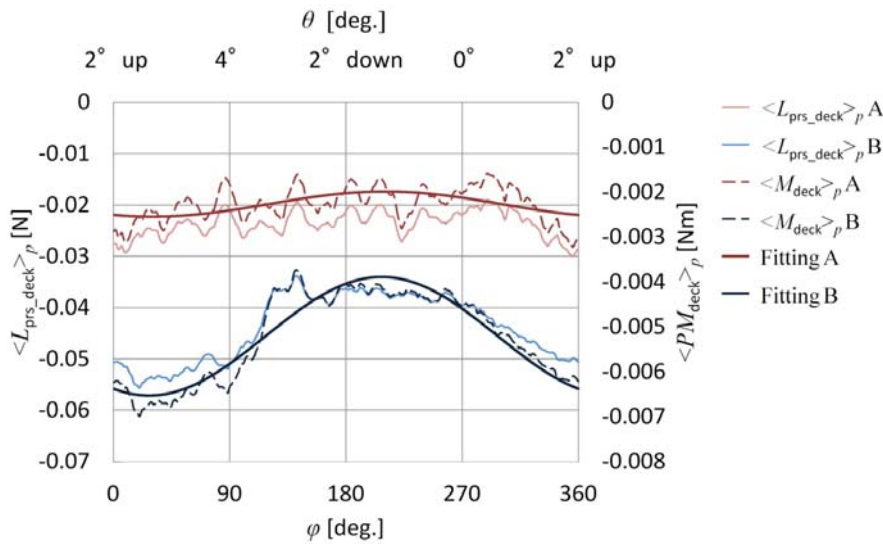


Figure 4. Phase-averaged M and fitted functions.

In general, as shown in Table 1, the main contribution in model A, which was created based on the characteristic aerodynamic features of lower stability sedan, is from the underbody and roof. Whilst, the main contribution in model B is from the underbody, upper rear section (i.e. trunk deck and rear shield) and roof. The reason for the models to have a different damping magnitude is because of the relatively high damping contribution from the upper rear section of model B,

particularly the trunk deck. Figure 4 shows that the curves of trunk deck fitted function of the two models have very similar phase shift. However, the relatively low trunk deck contribution in model A is caused by the smaller fluctuation amplitude.

5.2 Aerodynamic damping mechanism

5.2.1 Main aerodynamic damping contribution

The underbody has the highest damping contribution due to two reasons: First, the dynamic effect, i.e. vehicle motion, has caused the phase of phase-averaged underbody pitching moment $\langle M_{\text{underbody}} \rangle_p$ curve to shift (by about 128° and 134° in model A and model B, respectively) at the pitching oscillation frequency (see Figure 5), thus produces a negative M_{ang} ; Second, its relatively large surface area and moment arm produce a significantly larger M_{ang} magnitude than other body parts.

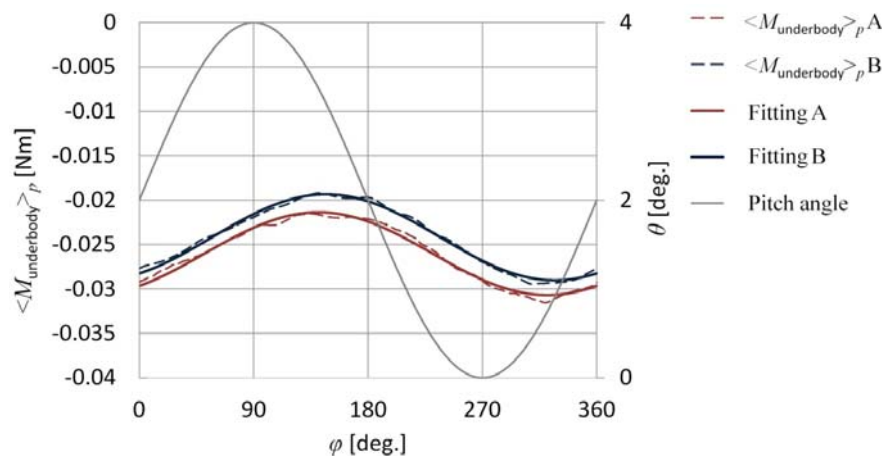


Figure 5. $\langle M_{\text{underbody}} \rangle_p$ and fitted functions of underbody: (a) Model A; (b) Model B.

The behavior of $\langle M_{\text{underbody}} \rangle_p$ curve can be explained by first considering the quasi-steady flow conditions with the model fixed at a range of pitch angle θ . As the underfloor clearance downstream of the pitch axis increases with θ , the flow decelerates and causes the static pressure to rise. The increase of static pressure with increasing θ causes a corresponding increase in underbody pitching moment. Hence, in the quasi-steady conditions, the maximum and minimum peaks would lie at 4° and 0° pitch, respectively. However, due to the pitching motion of models, there is an additional dynamic effect which causes the curve to overshoot. Figure 6 shows the properties of airflow in the underfloor clearance of model B at four pitching stages (similar trend is obtained in model A, thus only the results of model B is used for the discussion). As the underbody moves downwards from 4° pitch, the decreasing underfloor clearance is accompanied by flow acceleration, which may be evident by the relatively high streamwise velocity component during the 2° tail-down pitching cycle. However, the static pressure of underbody increases despite the acceleration of streamwise velocity component. In addition, for the same underfloor clearance at 2° tail-up and tail-down pitching cycles, the former exhibits a relatively strong cross flow velocity. Hence, it may be deduced that the increment of static pressure during a tail-down pitching cycle is caused by the conversion of cross flow kinetic energy into the dynamic pressure at the underbody. Meanwhile, the further decrement in static pressure during the tail-up pitching cycle is associated with "suction" effect (i.e. surrounding fluid is being "pull" away) the leeward side of a bluff body immersed in a flowing fluid would normally experience, as the underbody surface is now at the leeward side.

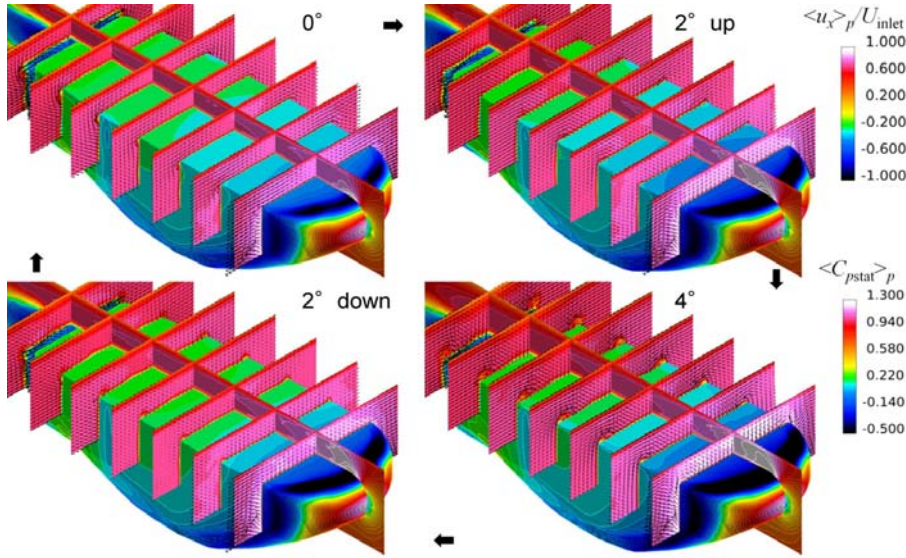


Figure 6. Phase-averaged velocity distribution at underfloor clearance and static pressure of underbody; model B.

5.2.2 Comparison between two aerodynamic configurations

Figure 4 shows that the phase-averaged trunk deck pressure lift $\langle L_{prs_deck} \rangle_p$ and phase-averaged trunk deck pitching moment $\langle M_{deck} \rangle_p$ curves of the two models are matching well, implies that the $\langle M_{deck} \rangle_p$ is mainly caused by the trunk deck surface static pressure. Ideally, the pressure force should be in-phased with the angular velocity of pitching to produce a maximum damping. That is, the $\langle L_{prs_deck} \rangle_p$ peaks at $\varphi = 180^\circ$, and reaches the minimum at $\varphi = 0$ or 360° . The $\langle L_{prs_deck} \rangle_p$ curves for the two models nearly meet this criterion, with only a slight phase shift. Hence, the C_{AD} obtained from the two models are having the same sign. However, the relatively large fluctuation range in model B has resulted in a higher damping magnitude.

Next, we discuss the reasons that cause the trends observed in Figure 4. As shown, model B has a relatively low attainable $\langle L_{prs_deck} \rangle_p$, which is caused by its concentrated C-pillar vortices (marked "B" in Figure 7). The concentrated vortices induced a narrow, low-static-pressure region at the sides of its trunk deck (marked "A" in Figure 7). In contrast, the C-pillar vortices in model A were weaker and less concentrated (marked "D" in Figure 7). As a result, the vortices induced a wider low static pressure region (marked "C" in Figure 7), which results in the higher attainable $\langle L_{prs_deck} \rangle_p$ in model A.

At $\varphi = 0$ or 360° , $\langle L_{prs_deck} \rangle_p$ in the two models were at the lower range, which was due to the increase of static pressure at the sides of trunk deck (i.e. the low pressure region narrows down). This tendency is caused by the decrease in the slant angle of C-pillar during tail-up pitching cycle. Hence, the models generate the weaker C-pillar vortices which diminish the drop in static pressure.

At $\varphi = 90^\circ$, the C-pillar vortices in model A were elevated by the trunk deck surface, thus its distance from A-pillar vortices (marked "E" in Figure 7) decreases (see Figure 7). These two pillar vortices, which rotate in directions opposite one another, interact with each other and generate a strong cross flow which passing through them and rolled upwards at the centerline, forming an upwash inducing, circulatory structure (marked "F" in Figure 7). Due to the strong cross flow and the upwash inducing circulatory structure, the static pressure in the central region drops and causes the $\langle L_{prs_deck} \rangle_p$ in model A to rise.

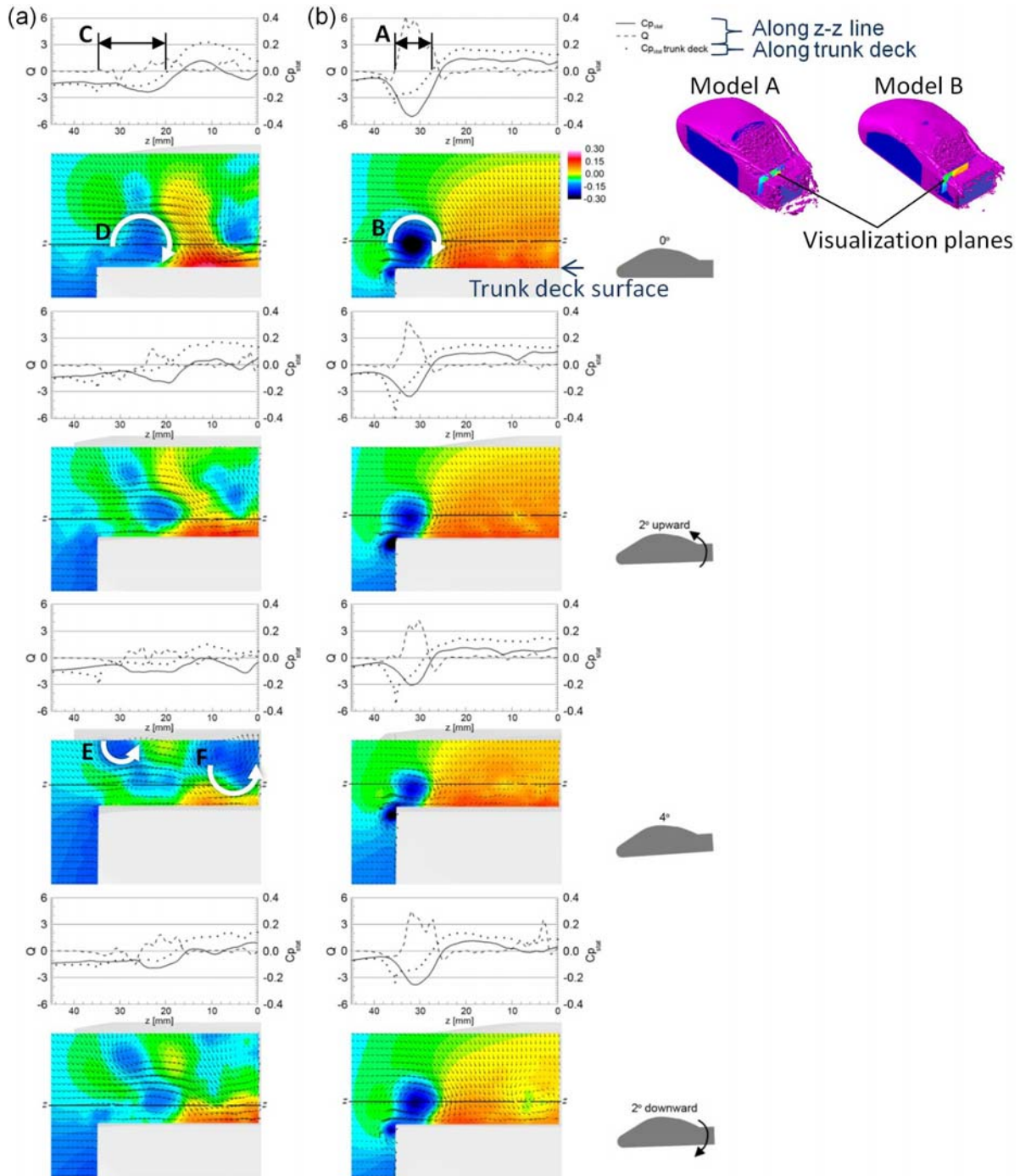


Figure 7. Distribution of cross flow velocity and static pressure above trunk deck at four different pitching stages:
(a) Model A; Model B.

At $\varphi = 180$ and 270° , $\langle L_{\text{prs_deck}} \rangle_p$ in the two models are at the higher range. The higher $\langle L_{\text{prs_deck}} \rangle_p$ in these instances was mainly caused by the decrease of static pressure at the sides of trunk deck. The slant angle of C-pillar increases with decreasing pitch angle. Hence, during the tail-down pitching cycle, the models generate the stronger C-pillar vortices which result in larger pressure drop on the trunk deck surface.

Although the models share the same tendency in how the C-pillar vortices affect the surface pressure at the side of trunk deck, but the distinct flow topology in the central region has caused the $\langle L_{\text{prs_deck}} \rangle_p$ curves to behave differently at $\phi = 180^\circ$. In model A, despite the larger pressure drop at the side, the $\langle L_{\text{prs_deck}} \rangle_p$ failed to reach a much higher value because of the increased of static pressure in the central region. This increment is caused by the attenuation of cross flow velocity and circulatory structure with decreasing pitch angle. As has been discussed earlier, with decreasing pitch angle, the gap between the A- and C-pillar vortices becomes larger, and thus their interaction which promotes the cross flow, becomes weaker. In model B, however, the drop in static pressure in the central region is due to the formation of the circulatory structure, and the dynamic effect (i.e. low static pressure at the leeward side) has caused the $\langle L_{\text{prs_deck}} \rangle_p$ to further increase. This produces the relatively large M_{ang} , and hence, a higher damping coefficient.

6 CONCLUSIONS

The present study has shown the potential use of LES as a tool to assess the aerodynamic stability of vehicle which takes into account the effect of transient aerodynamics. The proposed aerodynamic damping coefficient enables direct comparison of aerodynamic stability performance between two vehicle. For the simple body models investigated, the underbody provides the highest proportion of aerodynamic damping. However, it is the trunk deck contribution that causes the different damping magnitudes in the models with distinct A- and C-pillar geometrical configurations.

7 ACKNOWLEDGEMENTS

This work was supported by the 2007 Industrial Technology Research Grant program from the New Energy and Industrial Technology Development Organization (NEDO) of Japan. Development of the base software FFR was supported by the FSIS and “Revolutionary Simulation Software (RSS21)” projects sponsored by MEXT, Japan. The first author’s Ph.D. program is sponsored by Ministry of Higher Education and Universiti Teknikal Malaysia Melaka, Malaysia.

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