International Research Journal of Engineering and Technology (IRJET) **e-ISSN: 2395-0056**
Volume: 07 Issue: 06 | June 2020 **www.irjet.net** **p-ISSN: 2395-0072**

Effect of Operating Conditions on Vehicle Ride Comfort and Road Surface Friendliness with Air Suspension System

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Abstract - *The effects of the operating conditions of on vehicle ride comfort and road surface friendliness are presented in this study. In order to analyze them, a quarter-vehicle dynamic model is established under random road excitation. The weighted r.m.s acceleration responses of the vertical vehicle body (a_{wb}) and dynamic load coefficient (DLC) are chosen as the objective functions which use Matlab/Simulink software to simulate the vehicle dynamics model and calculate the objective functions. The road surfaces and vehicle speeds are selected for analysis and evaluation. The results show that* the *impact of the operating conditions of on vehicle ride comfort and road surface friendliness is very obvious. Especially, both vehicle ride comfort and road surface friendliness are going to be worse if the road condition deteriorates. The a_{wb} and DLC values increase quickly, making the negative effects on vehicle ride comfort and road surface friendliness.*

Key Words: Vehicle, air suspension system, operating condition, Ride comfort, *road friendliness.*

1. INTRODUCTION

The operating conditions of the vehicles have a significant effect on vehicle ride comfort as well as road surface friendliness. To improve the operating conditions of the vehicles, the vehicle's suspension systems have always been interested by researchers and designers, the effects of suspension design parameters of a semi-trailer truck on vehicle ride comfort as well as road surface friendliness were analyzed by using a half-dynamic vehicle model with 12 degrees of freedom[1]. Both stiffness and damping parameters of vehicle suspension were investigated the effects on vehicle ride comfort based on a three-dimensional vibration model of bus with 10 degrees of freedom[2]. The nonlinear geometric characteristics of the suspension systems of a 5-axle heavy truck was analyzed by using a 3 D nonlinear dynamic model[3]. Air suspension is increasingly used on heavy vehicles due to its capabilities of providing better ride quality and reducing the negative impacts of road surfaces. The different kinds of air spring dynamic models such as Nishimura, VAMPIRE, SIMPAC, and GENSIS are proposed and applied to analyze the characteristics of its dynamics[4]. The performances of the traditional and new air suspension systems were analyzed by using a 3D dynamic model with 14 degrees of freedom to compare the performance of the air suspension systems for reducing the negative impacts on the road surface[5]. Both the dynamic air spring suspension and the passive suspension were compared in terms of r.m.s of body acceleration using a 2-DOF quarter mathematical model[6]. The performance of the hydro-pneumatic suspension system was investigated by using a 3D dynamic model with 15 degrees of freedom to compare with the rubber and air spring suspension systems so that the negative impacts on the road surface could be reduced [7]. The influence of heavy vehicle dynamic parameters on ride comfort with random road excitation was examined by using a 3D dynamic model with 13 degrees of freedom[8].

In order to improve vehicle ride comfort and road surface friendliness, the methodology on the concept of optimum design of a road-friendly suspension to attenuate the tire load exerted by vehicles on pavement was presented in the research[9]. The control and optimization methods for air suspension systems have been discussed in some of the following studies: The genetic LQG and PID control were used to control an air suspension system[10]; the semi-active fuzzy control for controlling the air suspension system of heavy trucks[11] and the semi-active isolation systems for the effectiveness of the three-position control suspension of a heavy truck[12]. The influence of semi-trailer truck operating conditions such as road surfaces, vehicle speeds, vehicle loads on the dynamic tire loads was analyzed by using a three-dimensional vehicle-pavement coupled model with 14 degrees of freedom[13]. Various operating conditions were used to evaluate the acceleration-frequency characteristics of the suspension system of the heavy vehicles using nonlinear dynamic model of a two-axle heavy vehicle[14].

In this study, a quarter-vehicle dynamic model with an air suspension system is established for analyzing and evaluating the effect of the operating conditions of vehicle ride comfort and road surface friendliness. The weighted r.m.s acceleration responses of of the vertical vehicle body (a_{wb}) according to the ISO 2631:1997(E) [15] and dynamic load coefficient (DLC) are chosen as the objective functions which use Matlab/Simulink software to simulate the vehicle dynamics model and calculate the objective functions. The road surfaces and vehicle speeds are selected for analysis and evaluation.

2. VEHICLE DYNAMIC MODEL

In order to figure out the effects of the different vehicle operating conditions on vehicle ride comfort and road surface friendliness, a quarter-vehicle dynamic model with an air suspension system is established under random road excitation as shown in Fig-1.

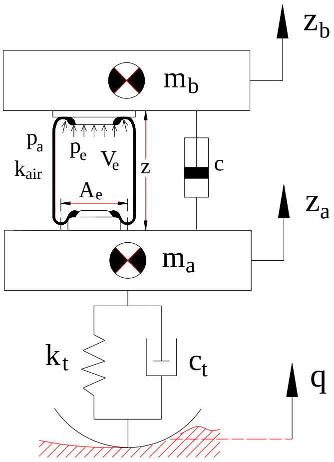


Fig-1: quarter-vehicle dynamic model with air suspension system

In Fig-1, m_a, and m_s are unsprung mass and sprung mass of vehicle; k_t and c_t are the stiffness and damping coefficients of tire; k_{air} and c are the stiffness coefficient of air bag and damping coefficient of a hydraulic damper; z_a *and* z_b are the vertical displacements of unsprung mass and sprung mass; and q is the excitation of road surface roughness; $z=z_b-z_a$ is the relative displacement between vehicle sprung mass (vehicle body) and unsprung mass (vehicle axle); p_e is the absolute pressure in the air chamber (Pa), p_a is the atmospheric pressure (Pa), and A_e is the effective area (m^2), V_e and A_e are the effective volume and area.

The equations of motion for a quarter-vehicle dynamic model using Newton's second law of motion are written as follows:

$$m_a\ddot{z}_a = k_{air}\left(z_a - z_b\right) + c\left(\dot{z}_a - \dot{z}_b\right) - k_t\left(z_a - q\right) - c_t\left(\dot{z}_a - \dot{q}\right) \tag{1}$$

$$m_b\ddot{z}_b = -k_{air}\left(z_a - z_b\right) - c\left(\dot{z}_a - \dot{z}_b\right) \tag{2}$$

Determining the stiffness coefficient of air bag of suspension system: According to Hooke's law[17], the stiffness k_{air} of air spring can be calculated by Eq.(3)

$$k_b = \frac{dF_a}{dz} = \frac{dp_e}{dz}A_e + \left(p_e - p_a\right)\frac{dA_e}{dz} \tag{3}$$

where, Fa is the air bag elastic force

From Fig. 1, there is no air exchange between the air spring component and the other system according to the thermodynamics law and ideal gas state equation given by:

$$p_eV_e^n = const \tag{4}$$

where, n is the polytropic exponent and depends on the following working conditions: n = 1 for the isothermal condition, n = 1.4 for the adiabatic condition, and 1 < n < 1.4 for the polytropic condition.

Differentiating Eq.(4) with respect to z yields:

$$\frac{d}{dz}\left(p_eV_e^n\right) = V_e^n\frac{dp_e}{dz} + np_eV_e^{n-1}\frac{dV_e}{dz} = 0 \tag{5}$$

$$\text{And} \qquad \frac{dV_e}{dz} = -A_e \tag{6}$$

From the equations above, the equivalent stiffness can be given as follows:

$$\begin{aligned} k_{air} &= n\left(p_g + p_a\right)\frac{A_e^2}{V_e} + p_g\frac{dA_e}{dz} \\ &= n\left[p_a + \left(p_0 + p_a\right)\left(\frac{V_0}{V_e}\right)^n - p_a\right]\frac{A_e^2}{V_e} + \frac{dA_e}{dz}\left[\left(p_0 + p_a\right)\left(\frac{V_0}{V_e}\right)^n - p_a\right] \end{aligned} \tag{7}$$

The stiffness of air spring at standard height (when the displacement z=0) is defined as

$$\begin{aligned} k_{air} &= n\left[p_a + \left(p_0 + p_a\right)\left(\frac{V_0}{V_e}\right)^n - p_a\right]\frac{A_e^2}{V_e} \\ &+ \left[\left(p_0 + p_a\right)\left(\frac{V_0}{V_e}\right)^n - p_a\right]\frac{dA_e}{dz}\bigg|_{z=0} \end{aligned} \tag{8}$$

where, V_0 is the initial effective volume; p_0 is the initial pressure in air bag.

The effective volume and area are defined as

$$\begin{cases} V_e = V_0 - \alpha\left(z_b - z_a\right) \\ A_e = A_0 + \beta\left(z_b - z_a\right) \end{cases} \tag{9}$$

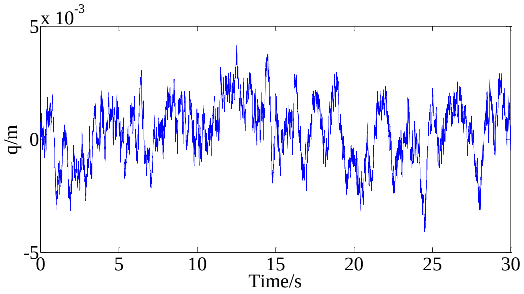
where, A_0 is the initial effective volume; α and β are the change of the effective volume and area with respect to z

Road surface excitation model[18], [20]: Many studies have shown there are several types of single-point time domain models of road irregularity excitation, namely FFT, AR/ARM, white noise filtering and harmony superposition methods. Based on the study carried out by Dodds C J and Robson J D [19], the road surface roughness is usually assumed to be a zero-mean stationary Gaussian random process and can be generated through an inverse Fourier transformation based on a power spectral density (PSD) function. The time domain excitation of the uneven road surface is generated as the sum of a series of harmonics:

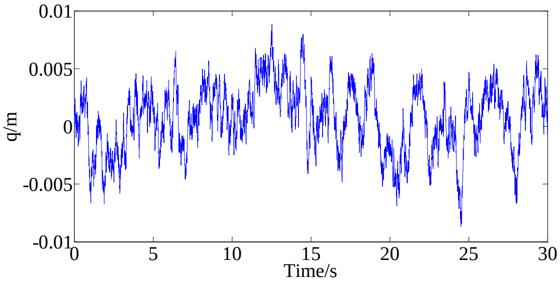
$$q(t) = \sum_{k=1}^{N}\sqrt{2G_q\left(f_{mid-k}\right)\Delta f_k}\,\sin\left(2\pi f_{mid-k}t + \varphi_k\right) \tag{10}$$

where, $G_q(f_{mid-k})$ is power spectrum, m^2/Hz; Δf is the frequency range, Hz; t is time, s; φ_k is the random phase uniformly distributed from 0 to 2π.

The road surface roughness with ISO class A and class B according to the standard ISO 8068[16] is shown in Fig-2.



(a) ISO class A



(b) ISO class B

Fig- 2: Road surface roughness according to ISO 8068.

3. EVALUATION INDICATORS

Vehicle ride comfort[1]: Currently there have been many methods to evaluate the vehicle ride comfort such as frequency-domain method, time-domain method, etc. This study is based on ISO 2631-1 (1997)[15], vibration evaluation based on the basic evaluation method including measurements of the weighted root-mean-square (rms) acceleration is defined by:

$$a_w = \left[\frac{1}{T}\int_0^T a_w{}^2(t)dt\right]^{1/2} \tag{11}$$

where $a_w(t)$ is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurements.

In this way, for indications of likely reactions to various magnitudes of overall vibration in the public transport a synthetic index-called the weighted r.m.s acceleration, a_w can be calculated from formula Eq.(11) and the r.m.s value of the vertical acceleration in vehicle would be compared with the values in Tab. 1.

Dynamic tire load[1]: Dynamic tire load would lead to stress and strain of road surface. The long-term accumulation of road surface plastic deformation causes the destruction of roads, such as cracks and rutting. In order to analyze the effects of the different vehicle operating conditions on vehicle ride comfort and road surface friendliness, the dynamic load coefficient (DLC) is chosen as objective function which is defined by a ratio of the root mean square of the vertical dynamic tire force over static load as follows:

$$DLC = \frac{F_{t,rms}}{F_s} \tag{12}$$

$F_{t,rms}$ and F_s are the root mean square of the vertical dynamic and the static tire force. The value of the DLC is in range of 0.05 to 0.3 under normal operating conditions. It may reach zero when the wheels move on a special smooth road or increase up to 0.4 when the tires of the axles spend a significant proportion of their time disconnecting the road surface [21].

Table- 1: Comfort levels related to a_w threshold values

a_w/(m.s^2)	**Comfort level**
< 0.315	Not uncomfortable
0.315÷0.63	A little uncomfortable
0.5 ÷ 1.0	Fairly uncomfortable
0.8 ÷ 1.6	Uncomfortable
1.25 ÷ 2.5	Very uncomfortable
> 2	Extremely uncomfortable

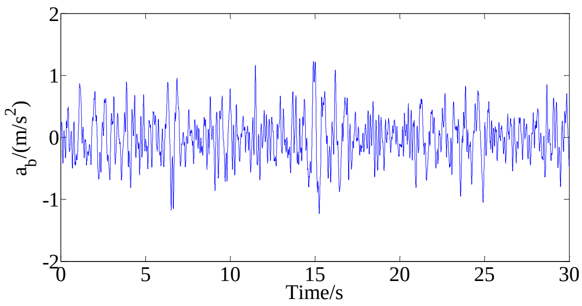
4. SIMULATION AND DISCUSSION

In order to solve the general dynamic differential equation of vehicle presented in section 2, Matlab/Simulink software is used with a set of parameters of vehicle in Tab.2.

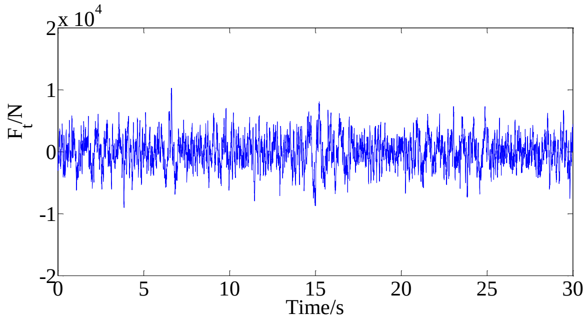
Table- 2: Parameters of the vehicle

Parameters	**Values**
m_a/kg	1554
m_b/kg	4871
c/ (N.s/m)	2.5539x104
k_t/ (N/m)	1.7933x106
c_t/(N.s/m)	2.414 x103
p_a/ Pa	0.1 x106
V_0/m^3	0.0333
A_0/m^2	0.0906
p_0/Pa	2.865 x106
n	1.4

The simulation results of the acceleration responses of the vertical vehicle(a_w) and the vertical dynamic tire load acting on road surface when vehicle moves on the ISO class B road surface at v=20m/s with fully loaded are shown in Fig.3. Fig.3 could determine the values of the weighted root mean square r.m.s of acceleration responses of the vertical vehicle body (a_{wb}) and the dynamic load coefficients (DLC) as a_{wb}= 0.3651m/s^2 and DLC=0.1644. The a_{wb} value is a little uncomfortable conditions for driver comfort according to ISO 2631-1 (1997) with the comfort levels related to a_w threshold values in Tab 1 and the DLC values are satisfied in the allowed range (in section 3) for road surface friendliness. The effects of the different vehicle operating conditions on a_{wb} and DLC values are being presented in the following in sections.



(a) Acceleration responses of the vehicle body



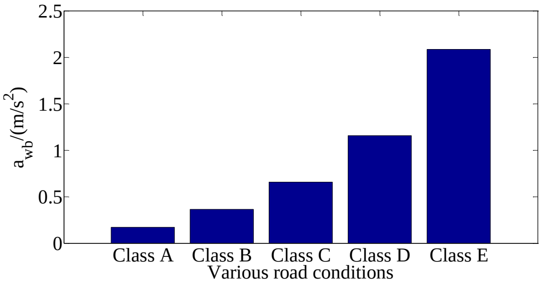
(b) Vertical dynamic tire force

Fig- 3: Acceleration responses of the vehicle body and vertical dynamic tire force

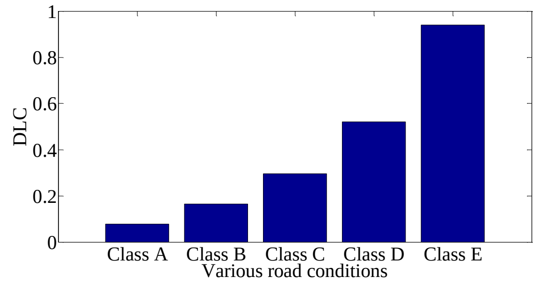
Effect of road surface roughness: Five road surface conditions from level A (very good) to level E (very poor) in ISO/TC 80686 have been considered as inputs to the vehicle-road coupled model when vehicle moves with a velocity of 20 m/s with full load. The a_{wb} and DLC values change when vehicle move various road conditions, as shown in Fig. 4.

Fig.4 shows that the a_{wb} and DLC values increase quickly when vehicle moves on the poor road surface conditions, which makes the negative effects on vehicle ride comfort and road surface friendliness. Especially, when the vehicle moves on the very good road surface condition (class A) to good road surface condition (class B) and then to average road surface condition (class C), the a_{wb} and DLC values increase from 11.90%, 80,12% and 11.86 %, 80.11%. The road surface conditions have a great influence on vehicle ride comfort and road surface friendliness.

Effect of vehicle speed: The vehicle speeds of 5 m/s, 10 m/s, 15 m/s, 20 m/s, 25 m/s, and 30 m/s were taken into account to analyze the effect of various vehicle speeds on the a_{wb} and DLC when vehicle moves on the road surface condition of the ISO level B and full loaded. The a_{wb} and DLC values change when vehicle moves in various speed conditions, as shown in Fig. 5.

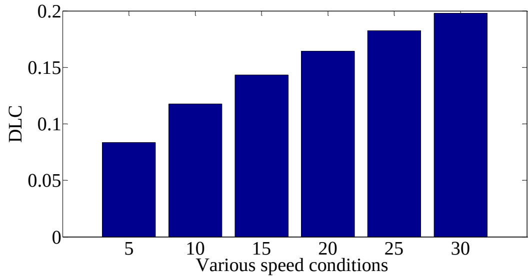


(a) a_{wb} values

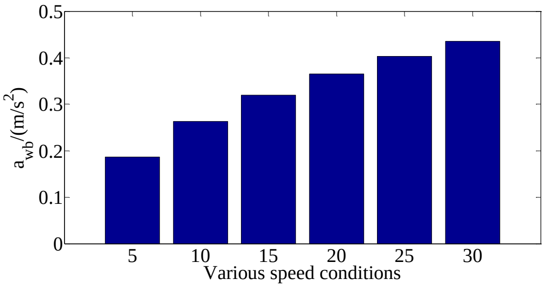


(b) DLC values

Fig- 4: a_{wb} and DLC values with various road conditions



(a) a_{wb} values



(b) DLC values

Fig- 5: a_{wb} and DLC values with various speed conditions

Fig.5 shows that the vehicle speed increases, the a_{wb} and DLC values increase quickly, making the negative effects on vehicle ride comfort and road surface friendliness. Especially, the vehicle speed increases from v=15m/s to v=20 m/s and then to v=25km/h, the a_{wb} and DLC values increase from 14.31%, 10.44% and 14.72 %, 10.95%. It is the vehicle speed conditions that have a great influence on vehicle ride comfort and road surface friendliness.

5. CONCLUSIONS

In this study, a quarter-vehicle dynamic model with an air suspension system is designed in order that the influence of the operating conditions on vehicle ride comfort and road surface friendliness can be investigated and evaluated. Here are the major conclusions drawn from the analysis and evaluation results: (1) The vehicle ride comfort and road surface friendliness is going to be worse when the road condition deteriorates and (2) The a_{wb} and DLC values increase quickly, making the negative effects on vehicle ride comfort and road surface friendliness. The study results certainly are the theoretical basis for the designing optimization of the air suspension system which will be published in the future by the authors.

Acknowledgment

This research was supported financially by Thai Nguyen University of Technology, TNUT, Viet Nam.

REFERENCES

[1] Van Quynh L., Van Cuong B., Long L.X., Van Quan D. (2020). “Effects of Suspension Design Parameters of a Semi-trailer Truck on Vehicle Ride Comfort and Road Surface Friendliness”, CERA 2019: Advances in Engineering Research and Application, pp 281-289.

[2] LX Long, LV Quynh, BV Cuong (2018). “Study on the influence of bus suspension parameters on ride comfort”, Vibroengineering PROCEDIA, Vol. 21, p. 77-82.

[3] VQ Le, J Zhang, X Liu, Y Wang (2011). “Nonlinear dynamic analysis of interaction between vehicle and road surfaces for 5-axle heavy truck”, Journal of Southeast University, Vol. 27, Issue 4, p. 405-409.

[4] Haider J. Abid, Jie Chen, and Ameen A(2015). “Nassar. “Equivalent Air Spring Suspension Model for Quarter-Passive Model of Passenger Vehicles”, International Scholarly Research Notices Volume 2015, Article ID 974020, 6 pages

[5] Le, V.Q.(2017). “Comparing the performance of suspension system of semi-trailer truck with two air suspension systems” Vibroengineering PROCEDIA, Vol.14, 220–226.

[6] Moheyeldein M M, Abd-El-Tawwab A M, Abd El-gwwad K A, and Salem M M M (2018). “An analytical study of the performance indices of air spring suspensions over the passive suspension”, Beni-Suef University Journal of Basic and Applied Sciences, Vol.7(4), p. 525-534.

[7] Long, L.X., Hong, T.T., Quynh, L.V., Van Cuong, B.(2018). Performance analysis of the hydro-pneumatic suspension system of heavy truck” Int. J. Mech. Eng. Technol. (IJMET), Vol.9(13), p.1128–1139.

[8] Van Quynh, L., Zhang, J., et al.(2013). “Influence of heavy truck dynamic parameters on ride comfort using 3D dynamic model”, Dongnan Daxue Xuebao (Ziran Kexue Ban)/J. Southeast Univ. (Nat. Sci. Ed.), Vol.43(4), 763–770 (2013).

[9] Lu Sun (2002). “Optimum design of road friendly vehicle suspension systems subjected to rough pavement surfaces”, Applied Mathematical Modelling 26, 635–652.

[10] Chuan-yin Tang and Yun-gong Li (2017). “Research on Air Suspension System Based on Genetic LQG and PID Control” International Conference on Optics, Electronics and Communications Technology (OECT 2017)

[11] Van Liem, N., Jianrun, Z., Van Quynh, L., Renqiang, J., Xin, L.(2017). “Performance analysis of air suspension system of heavy truck with semi-active fuzzy control”, J. Southeast Univ. (Engl. Ed.), Vol. 33(2), p.159–165.

[12] Nguyen, V.L., Van Quynh, L.(2019). Ride comfort performance of heavy truck with three control cases of semi-active isolation systems. Vibroengineering PROCEDIA, Vol. 22, p. 93–98.

[13] Van Quynh, L.(2017). “Influence of semi-trailer truck operating conditions on road surface friendliness”, Vibroengineering PROCEDIA 16, p.67–72.

[14] V Nguyen, V Le, R Jiao, H Yuan(2020). “Low-frequency vibration analysis of heavy vehicle suspension system under various operating conditions”, Mathematical Models in Engineering, Vol. 6, Issue 1, p. 13-22.

[15] .ISO 2631-1 (1997). Mechanical Vibration and Shock-Evaluation of Human Exposure to Whole-Body Vibration, Part I: General Requirements. The International Organization for Standardization, 1997.

[16] ISO 8068 (1995). Mechanical Vibration-Road Surface Profiles-Reporting of Measured Data. International Organization for Standardization.

[17] Xiao, P., Gao, H., Shi, P., & Niu, L. (2018). “Research on air suspension with novel dampers based on glowworm swarm optimization proportional–integral–derivative algorithm”, Advances in Mechanical Engineering, 10(8), 168781401879171.

[18] LV Quynh, BV Cuong, et al (2019). “Effect of in-wheel motor suspension system on electric vehicle ride comfort”, Vibroengineering PROCEDIA, Vol. 29, p. 148-152.

[19] Dodds C J, Robson J D (1973). “The description of road surface roughness. Journal of Sound and Vibration”, Vol.31(2), p. 175–183.

[20] HA Tan, LV Quynh, NV Liem, BV Cuong, LX Long(2019). “Influence of damping coefficient into engine rubber mounting system on vehicle ride comfort”, Vibroengineering PROCEDIA, Vol. 29, p. 112-117.

[21] Rosnawati Buhari, Munzilah Md Rohani, et al (2013). “Dynamic load coefficient of tyre forces from truck axles”, Applied Mechanics and Materials, Vol. 405, Issue 408, p.1900-1911.