Numerical Modeling of an Offshore Small Floating Platform with Flexible Connections

Using Elastic Force and Wave Force in Scaled Model Experiments

By Bingting Zha • Guangsong Chen • Kefeng Xie • Fukun Gui

Marine resources, rights and interests have garnered a great deal of attention within the scientific community; an array of different offshore platforms have been developed for target detection and rights protection. Unlike other large platforms, offshore small floating platforms (OSFPs) are characterized by small size, portability, and highly functional design. An OSFP freely floats on the sea, so its movement is deflected by strong waves and currents. There has been little research on the motion response of OSFPs, though it is necessary to adjust the movement of the small platform according to this response.

Many studies have applied numerical and experimental methods to investigate the motion response of other offshore platforms. Miles and Gilbert (1968), Garrett (1971), Yeung (1981), and Sabuncu and Calisal (1981) collected extensive data on the hydrodynamic coefficients of vertical cylinders in waters of finite depth. Xi (1984) completed linear and nonlinear vibrational analyses for free-floating cylindrical offshore platforms in regular waves. Others have explored massive large floating platforms; Xie et al. (1999) used linear 3D potential theory to calculate the hydrodynamic interaction between two bodies floating in waves, and Gou et al. (2004) used a boundary integral equation method to examine the hydrodynamic interactions between waves and two connected floating structures. Wu et al. (2003) studied the influence of the distance between floating bodies on the hydrodynamic coefficients of floating multi-body systems. Diamantoulaki (2010) evaluated the effects of wave frequency and the array of floating breakwaters per responses in the frequency domain. Chen (2016) proposed a raft-type wave energy dissipator supported by a mathematical hydrodynamics model; Wang (2014) performed modal analyses in the frequency domain to elucidate the motion response of hinged multiple floating bodies in regular waves. Cen (2015) analyzed the relationship between mooring line diameter and the dynamic performance of a multi-body system.

Although previous studies have extensively investigated the dynamic response of floating platforms, these works have focused on large-sized, hinged, or single floating platforms. There has been no effective method proposed to date for evaluating the wave-induced forces and motion of OSFPs with flexible connections. Li (2007), Zhao (2007), and Gui (2014) built a numerical model of a flatfish cage via lumped mass method and rigid body kinematics, thereby resolving the wave forces acting on a small straight floating pipe with the Morison equation. Jin (2011) simplified the platform into a multi-DOF rigid body and studied the effect of wave loads. Li (2013) used 3D potential flow theory and Green's function method to establish a calculation model. Again, there have been few such studies on small multi-floating systems with flexible connections—those studies that have been conducted center around numerical models or experimental tests alone.

The remainder of this paper is organized into three sections. Section 1 describes the forces acting on the platform in addition to flexible force, and provides numerical models thereof. Section 2 discusses the physical model tests that were conducted to validate the effectiveness of the numerical model. Conclusions are drawn in section 3.

Numerical Modeling of OSFP

Principle of OSFP

The OSFP provides a base for the load system, which serves as a base for detecting and tracking targets. Fig. 1(a) shows the specific composition of the OSFP. The OSFP contains a power system, a stabilization system, a measurement system and a floating system. The forces acting on the platform include weight, buoyancy, wave force and elastic force. All forces apart from weight vary with the motion of the OSFP. Fig. 1(b) shows a sketch of the simplified numerical model of the OSFP. It mainly consists of a floating gasbag, a cylinder and a flexible connection system. A top view of the model is shown in Fig. 1(c). The following assumptions are made: 1) elastic deformation of the floating gasbag is ignored; 2) the influence of the floating platform on the wave field is ignored; 3) the wave propagates in the forward direction, and the floating platform is arranged symmetrically in the water. Thus, the motion of the platform is simplified into 2D under linear wave conditions.
The cylinder and the floating gasbag are subjected to the action of weight, buoyancy, wave force, and elastic force as depicted in Fig. 2. The fixed coordinate system $o-xyz$ and body coordinate systems $o_f-xyz$ and $o_p-xyz$ are established accordingly, and the wave direction is along the $x$ axis. The centroid of the floating gasbag is $o_f$, and the centroid of the cylinder is $o_p$. The axes of the body frames correspond to the axes of the fixed coordinate system when at rest.

**Calculation of forces on cylinder**

**Calculation of buoyancy and moment**

The buoyancy of the cylinder is determined by the volume of the part which is submerged in the water. The coordinates of the centroid of cylinder are $(x_n, 0, z_n)$, and its pitch angle is $\theta_n$, as shown in Fig. 3. The wave surface elevation $\eta$ can be calculated under wave theory. The inclination angle of the wave is assumed as $\alpha$, and the buoyancy $F_b$ acting on the cylinder is calculated as follows:

$$F_b = \rho A_f \left( \eta - z_n + h \right) \frac{\sin(\alpha + \theta_n)}{2} \sin(\alpha + \theta_n), \quad (1)$$

where $A_f$ denotes the cross-section area of cylinder, and $h$ denotes the height between the centroid and the bottom.

The moment of buoyancy includes two parts. The centroid of the right triangle is the point of force of Part I. The arm of Part I's buoyancy is:

$$L^I = -\frac{\eta - z_n}{\sin(\alpha + \theta_n)} \frac{\sin(\alpha + \theta_n)}{2} \left( 1 - \tan(\alpha + \theta_n) \frac{\sin(\alpha + \theta_n)}{2} \right), \quad (2)$$

where $D$ denotes the diameter of the cylinder.

The moment $M^I_b$ is calculated as follows:

$$M^I_b = \rho A_f L^I \frac{1}{2} \tan(\alpha + \theta_n) D, \quad (3)$$

The point of force of Part II is set on the axis of the cylinder. The moment $M^H_b$ is calculated as follows:
Fig. 3 Cylinder coordinates

\[ M''_h = \rho A_h l (l/2 - h + l) \sin \theta_n, \]  

(4)

where

\[ l = \frac{\eta - z_n}{\sin(\alpha + \theta_n)} + h - l_i = l_i = \frac{1}{2} \tan(\alpha + \theta_n) D, \]  

(5)

The moment \( M''_h \) can then be calculated as follows:

\[ M''_h = M'^{''}_h + M^{''}_h, \]  

(6)

Calculation of wave forces and moment

Wave-induced drag and lift forces acting on the floating platform change with the wave. According to Li (2007) and Gui (2014), a modified Morison equation can be used to calculate the wave force on a small inclined floating cylinder. This requires first defining the impact angle as the minimum angle between the wave direction and cylinder axes, which always varies within \([0, \pi/2]\). Combined with the features of OSFP, we define the impact angle \( \theta \) here as that between the normal wave direction and cylinder axes as shown in Fig. 4. As the direction of the pitch is from the cylinder axes to the positive z-direction clockwise, the impact angle is positive.

There are many impact factors of the hydrodynamic coefficients. Properly accounting for these factors makes the analysis rather complex. In a numerical simulation, the relationship between the hydrodynamic coefficients and Re is considered, and certain corrections on the hydrodynamic coefficients are made as necessary.

We divided the cylinder into micro-segments and then calculated the wave force on each micro-segment individually. The wave force on the cylinder can then be obtained by integrating these calculations along the cylinder axes. The force acting on a micro-segment is shown in Fig. 4. The velocity and acceleration of the wave particles in x- and z-directions can be calculated under the corresponding wave theory. The lift force is neglected due to only slight oscillatory action in the transverse direction. The velocity \( \mathbf{u} \) of the water particle can also be broken down into normal component \( u_n \) and tangential component \( u_t \), and the hydrodynamic forces of each direction can be calculated as follows:

\[
\begin{bmatrix}
u_n \\
u_t
\end{bmatrix} = \begin{bmatrix}
\cos \theta & \sin \theta \\
\sin \theta & \cos \theta
\end{bmatrix}\begin{bmatrix}
u_s - U_s \\
u_t - U_t
\end{bmatrix},
\]

(7)

where \( U_s \) and \( U_t \) are the velocity components of the micro-segment.

The corresponding central angle \( \omega \) of the projected
chord-length on the surface is:

\[
\omega = \begin{cases} 
2\arccos \left( \frac{l_1}{d} \right) & l_1 < x < l + 2l_1 \\
2\pi & 0 < x < l_1
\end{cases}
\]

The wave force on micro-segment is mainly composed of drag force \( f_s \) and inertial force \( f_i \):

\[
f_s = \frac{1}{2} C_D \rho A u \left| u \right|, \quad f_i = \frac{1}{2} C_M \rho A u \left| u \right|
\]

where

\[
A_s = \int_0^{l + l_1} \sqrt{\left( \frac{l + l_1 - x}{\tan(\alpha + \theta_m)} \right)^2} \, dx = \frac{\pi d^2}{4} \left( l + l_1 \right)
\]

\[
A_i = \int_0^l \left( \frac{d \omega - d^2}{4} \tan \omega \right) \, dx = \frac{\pi d^2}{4} \left( l \right)
\]

and

\[
l_1 = \frac{l + l_1 - x}{\tan(\alpha + \theta_m)}, \quad l_1 = \frac{l + l_1 - x}{\tan(\alpha + \theta_m)}.
\]

\( C_D \) is the normal drag coefficient, which does not change significantly in any small smooth cylinder (Hou and Gao, 1998). \( C_M \) is the tangential coefficient, which is usually regarded as a constant. The relationship between \( C_D \) and \( C_M \) can be written as \( C_D = C_M \), \( C_M = 0.02 \), for a small, smooth cylinder (Wang, 1995; Takagi, 2002). For the small floating cylinder, the hydrodynamic coefficient under wave conditions may be more complex (Teng and Li, 1991). According to Yucheng Li (2007), the coefficients can be revised by combining the numerical simulation with experimental data.

According to the coordinate transformation relation, the wave forces in a fixed coordinate system are:

\[
\begin{bmatrix} f_{i_1} \\ f_{i_2} \end{bmatrix} = \begin{bmatrix} \cos \theta & -\sin \theta \\ \sin \theta & \cos \theta \end{bmatrix} \begin{bmatrix} f_s \\ f_i \end{bmatrix}
\]

The direction of \( f_i \) is consistent with the cylinder axes, so the moment by \( f_i \) is 0. The moment by \( f_s \) is:

\[
M_{i_2} = \int_0^{l + l_1} z f_i \, dx
\]

where \( z \) is the distance between the centroid of the cylinder and the micro-segment.

According to Brebbia and Walker (1979), inertia forces in x- and z-directions can be calculated by:

\[
f_{i_1} = \rho V_0 a \left( \frac{1}{2} \omega - \frac{d^2}{4} \tan \omega \right) \left( l + l_1 \right) \cos \theta \sin \theta, \\
f_{i_2} = \rho V_0 a \left( \frac{1}{2} \omega - \frac{d^2}{4} \tan \omega \right) \left( l + l_1 \right) \cos \theta \sin \theta,
\]

where \( V_0 = \left( \frac{\pi d^2}{4} \left( l + l_1 \right) \sin \omega \right) \left( l + l_1 \right) \tan \omega \), \( a = 1 \).

The inertia moment on the cylinder is:

\[
M_{i_2} = \int_0^{l + l_1} \left( f_{i_1} \cos \theta + f_{i_2} \sin \theta \right) \, dx
\]

and the resultant force on cylinder in the fixed coordinate is:
so the resultant wave moment on cylinder is:

\[ M_{Rc} = M_{Hy} + M_{Hu} \]  

(20)

**Calculation of forces on the floating gasbag**

The motion of the platform is simplified into a 2D model, but the forces acting on the floating gasbag still need to be analyzed in 3D. A schematic diagram of the forces acting on the floating gasbag is shown in Fig. 5. The body coordinate system \( n\tau \) is defined in micro-segments along the circumferential direction; \( n \) and \( \tau \) are the normal and tangential directions of the micro-segment, respectively. The loads acting on the floating gasbag include wave force, buoyancy and elastic force. Elastic force is discussed in the section below.

![Diagram](image)

**Fig. 5. Schematic diagram of forces acting on floating gasbag**

**Calculation of wave forces**

The velocity and acceleration of water particles are broken down along the \( x \) axis and \( z \) axis in the fixed coordinate system. The direction of \( u_x \) in \( n\tau \) is \( \{\varphi_n, \varphi_n, \varphi_n\} \), and the direction of \( u_z \) in \( n\tau \) is \( \{\varphi_n, \varphi_n, \varphi_n\} \). The directions of \( u_x \) and \( u_z \) are determined by the movement of the floating gasbag and the locations of the micro-segments.

The floating platform is arranged symmetrically, and the wave force in the \( y \) direction micro-segment are canceled out. \( u_x \) and \( u_z \) in \( n\tau \) can be obtained as follows:

\[ u_x = u_n \cos \theta = u_n \cos \beta \cos \alpha \]
\[ u_{x\tau} = u_n \cos \theta = u_n \sin \beta \cos \alpha \]
\[ u_{x\nu} = u_n \cos \theta = u_n \sin \alpha \]
\[ u_{zn} = u_n \cos \varphi_n = u_n \cos \beta \cos \alpha \]
\[ u_{z\tau} = u_n \cos \varphi_n = u_n \sin \beta \cos \alpha \]
\[ u_{z\nu} = u_n \cos \varphi_n = u_n \sin \alpha \]

Eqns. (21) and (22) indicate that \( \{u_n, u_\tau, u_\nu\} \) in \( n\tau \) can be obtained by:

\[
\begin{aligned}
  u_n &= u_{xn} + u_{zn} \\
  u_\tau &= u_{x\tau} + u_{z\tau} \\
  u_\nu &= u_{x\nu} + u_{z\nu} 
\end{aligned}

\]

(23)

According to Brebbia and Walker (1979), the wave forces on a micro-segment in the floating gasbag can be calculated by the Morison equation. The Morison equation can be modified to suit wave conditions as follows:

\[
f = \frac{1}{2} C_i \rho A_n \left| u - \bar{U} \right| \left[ u - \bar{U} \right] + \rho V_a a + C_{al} \rho V_0 |a - \bar{U}| \]

(24)

where \( \bar{u} \) and \( \bar{U} \) are the velocity vectors of the water particle and the micro-segment' \( \bar{a} \) and \( \bar{U} \) are the acceleration vectors of the water particle and the micro-segment. \( C_i \) and \( C_{al} \) are the drag and inertial coefficients.

The drag forces acting on a micro-segment are:

\[
\begin{aligned}
  f_n &= C_i \rho A_n \left( u - \bar{U} \right)^2 \\
  f_\tau &= C_i \rho A_n \left( u_\tau - \bar{U}_\tau \right)^2 \\
  f_\nu &= C_i \rho A_n \left( u_\nu - \bar{U}_\nu \right)^2 
\end{aligned}
\]

(25)

\( (i = n, \tau, \nu) \) is the drag coefficient in the \( i \) direction, and \( A_n (i = n, \tau, \nu) \) is the corresponding effective projected area in the \( i \) direction.

The effective projected area depends on inclination \( \alpha \), azimuth \( \beta \), and the corresponding wave slope angle \( \gamma \) of the micro-segment. The center coordinate of the
The depth in water of the micro-segment in the \( n \) direction can be calculated as follows:

\[
d_n = r - (z_i - \eta) \cos \gamma,
\]

where \( r \) is the radius of the floating gasbag.

The projected chord-length can be obtained by:

\[
d_v = 2\sqrt{r^2 - (r - d_v)^2},
\]

and the corresponding central angle of the projected chord-length can be calculated by:

\[
\phi = 2 \arccos \left(1 - \frac{d_v}{r}\right),
\]

The cross-section area in water is:

\[
A_n = d_n dl
A_r = \frac{1}{\pi} r \phi dl
A_v = d_v dl
\]

The wave forces are converted to a fixed coordinate system as:

\[
\begin{bmatrix}
  f_{\xi} \\
  f_{\zeta}
\end{bmatrix} =
\begin{bmatrix}
  \cos \theta_n & \cos \theta_r & -\cos \theta_v \\
  \cos \omega_n & \cos \omega_r & \cos \omega_v
\end{bmatrix}
\begin{bmatrix}
  f_n \\
  f_r \\
  f_v
\end{bmatrix}
\]

The inertial forces acting on the micro-segment are:

\[
\begin{bmatrix}
  f_{\xi} \\
  f_{\zeta}
\end{bmatrix} =
\begin{bmatrix}
  \rho s_l a_x + C_{n\omega} \rho s_l (a_x - U_x) \\
  \rho s_l a_z + C_{n\omega} \rho s_l (a_z - U_z)
\end{bmatrix}
\]

where \( C_{\omega} \) and \( C_{n\omega} \) are the inertia coefficients. The complexity and many influence factors in this system allow \( C_{\omega} \) and \( C_{n\omega} \) to be considered equal.

The wave forces on the floating gasbag in the fixed coordinate are:

\[
F_x = \int_0^{2\pi} (f_{\xi} + f_{\zeta}) dl,
\]

\[
F_z = \int_0^{2\pi} (f_{\xi} + f_{\zeta}) dl,
\]

and the resultant wave moment on the floating gasbag is:

\[
M_{\xi\eta} = \int_0^{2\pi} (f_{\xi} + f_{\zeta}) \xi dl.
\]

**Calculation of buoyancy**

The buoyancy of a micro-segment, as shown in Fig. 6, is:

\[
f_{\beta} = \rho g s l_i.
\]

The moment of the floating gasbag is:

\[
M_{\eta\xi} = \int_0^{2\pi} f_{\xi} \xi dl.
\]

**Calculation of elastic force of connection**

The cylinder is connected to the floating gasbag by four lightweight, 5-mm-diameter flexible connections. Only the elastic force was considered during simulation. Flexible connections initially have certain pretension forces, and
there is consistent tension on the cylinder in movement. The tension of the flexible connections is related to their level of extension, which can be calculated by the connection point at both ends (the floating gasbag and the cylinder). The coordinates can be obtained according to the movement of the platform.

The connection points are \( A_{1}, A_{2}, A_{3}, \) and \( A_{4} \), as shown in Fig. 7. The elastic force can be obtained by:

\[
F_{11} = K_{1} \left( \frac{A_{1} - A_{2}}{l_{0}} \right),
\]

\[
F_{12} = K_{1} \left( \frac{A_{1} - A_{3}}{l_{0}} \right),
\]

where \( K_{1} \) is the elastic modulus of the flexible connection.

The angles between the cylinder axes and elastic connections are \( \theta_{1}, \theta_{2} \).

\[
\theta_{1} = \arccos \left( \frac{l_{f} \cdot A_{4} \cdot A_{3}}{l_{f} \cdot A_{4} \cdot A_{3}} \right),
\]

\[
\theta_{2} = \arccos \left( \frac{l_{f} \cdot A_{4} \cdot A_{3}}{l_{f} \cdot A_{4} \cdot A_{3}} \right).
\]

The elastic forces are broken down into the body coordinate system \( o-xyz \).

\[
\begin{align*}
F_{11}^{\prime} & = \begin{bmatrix} \cos \theta_{1} \\ \cos \theta_{2} \end{bmatrix} F_{11}, \\
F_{12}^{\prime} & = \begin{bmatrix} \cos \theta_{1} \\ \cos \theta_{2} \end{bmatrix} F_{12}.
\end{align*}
\]

(44)

(45)

The elastic force torques on cylinder is:

\[
M_{x} = F_{11}^{\prime} \frac{d}{2} - F_{12}^{\prime} \frac{d}{2} + F_{21}^{\prime} x_{c} + F_{22}^{\prime} x_{c},
\]

where \( x_{c} \) is the distance between the connection point and the centroid of the cylinder.

At this point, the elastic forces and torques acting on the cylinder are known, and the elastic forces and torques acting on the floating gasbag are the action with and reaction against the elastic forces and torques acting on the cylinder.

**Establishment of numerical model**

The forces and torques of the cylinder, the floating gasbag, and the flexible connection were obtained as described above. As mentioned in Section 2.1, the two parts have different motions and are defined with generalized DOF. Based on Newton’s second law, the numerical model of the floating platform in the temporal domain is:
\[
\sum_{j=1}^{k} M_{kj} \ddot{x}(t) = F_k(t), k = (1, 2, \ldots, 6)
\]  

(49)

where \( M_{kj} \) is a generalized mass matrix, \( F_k(t) \) is the resultant force (or torque), \( \ddot{x}(t) \) is the generalized acceleration of the floating platform, \( k = (1, 2, 3) \) represents the pitch, vertical shift, and surge of the cylinder, and \( k = (4, 5, 6) \) represents the pitch, vertical shift, and surge of the floating gasbag.

The motion response of OSFP can be obtained by solving Eq. (46) through the fourth-order Runge-Kutta method.

**Numerical Results and Experimental Verification**

To verify the validity of the numerical model, we conducted a series of scaled model experiments in a wave flume at the Key Laboratory of Marine Engineering Technology, Zhejiang Ocean University, China. The wave flume is 130 m long, 6 m wide, and 4 m high. It is equipped with a rocker flap wave-maker and a wave damping apparatus.

We arranged the experimental model symmetrically in the flume 25 m away from the wave-maker. Wave data was collected by a wave-height gauge around the floating platform. The swing angle of the cylinder and the gasbag were measured by an attitude heading reference system (AHRS) fixed alongside them with an acquisition frequency of 100 Hz. The elastic force of the connection was measured with a sensor connected at the middle of the elastic connections. The OSFP model configuration is shown in Fig. 8 and Fig. 9 (see supplemental information).

The geometrical similarity scale of the model to the simulation setup is 1:4. The center of gravity and the moment of inertia were adjusted during the experiment by the internal weight. The parameters of the floating platform are shown in Table 1, and the experiment is described in detail in Fig. 10 (see supplemental information). The coefficients \( C_n \) and \( C_r \) are 1.2, and \( C_f \) is 0.7. The experiment was conducted in pure wave conditions. The elastic force of the connection and the swing angle were selected for comparison against the numerical data, as shown in Figs. 11-14 (see supplemental information).

As shown in Fig. 11 and Fig. 12, the elastic forces of the connection per the simulated results were consistently in close agreement with the experimental data. The force of the wave side and lee side was not symmetrical. Discrepancies may have arisen from three unavoidable sources: 1) the length of the connection and preload were not exactly the same; 2) the influence of flow was neglected in the numerical model; 3) the gasbag was assumed to be a rigid body in order, while there was some flexibility in the scaled model experiment.

A comparison of the swing angle values between the simulation and experiment is shown in Fig. 13 and Fig. 14. Again, the results were altogether in close agreement. The amplitude error was less than 5%, and the angle error was 12.3%. In short, the experimental curves validate the numerical simulation.

We obtained the motion responses of the numerical simulation and experiment by changing the intrinsic parameters of the platform. As shown in Fig. 15, under a different center of gravity of the platform and gasbag radius, the results of the elastic force and angle response were consistently in accordance. We also found that the elastic force and angle response decreased as the center of gravity improved; in effect, the radius can also influence the motion response. Taken together, these results provide a useful set of guidelines for parameter optimization.

**Conclusions**

In this study, we calculated the elastic force and wave force of a typical OSFP to deduce a numerical model based on the differential element method. We conducted scaled model experiments to validate the numerical model. Our conclusions can be summarized as follows.

1) The elastic force and platform swing angle calculated via the proposed method are in accordance with the
experimental data, despite some differences in the waveform due to parameter selection and model error.

2) There are very small changes in the flexible forces as wave height and period increase. The swing angle grows larger as these parameters increase as well. There are more flexible forces on the wave side than the lee side due to the connection length, preload and influence of flow.

3) The flexible force and swing angle decrease as gasbag radius decreases; they also decrease as the gasbag’s center of gravity increases.

This study marks the first time that a numerical model for OSFPs with flexible connections has been established and verified. Our results suggest that the proposed numerical model is suitable for simulating OSFP motion responses. Further research on such responses will help to better our understanding of the parameter characteristics of OSFPs.

Supplemental Information
See supplemental pages for Figs. 8-15 and Table 1.

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References
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Supplemental Information

Fig. 8. OSFP model configuration

Fig. 9 Sketch of model cross configuration

Table 1 OSFP parameters

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<th>Part</th>
<th>Parameter</th>
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Fig. 10 Scaled model experiment

Fig. 11 Elastic force of connection; calculated vs. experimental data (H=10 cm, T=2.0 s)

Fig. 12 Elastic force of connection; calculated vs. experimental data (H=26 cm, T=2.4 s)
Fig. 13 Swing angle; calculated vs. experimental data (H=10 cm, T=2.0 s)

Fig. 14 Swing angle; calculated vs. experimental data (H=26 cm, T=2.4 s)

Fig. 15 Motion response of platform; calculated vs. experimental data with different centers of gravity