MULTI-BODY DYNAMICS AND VIBRATION ANALYSIS OF CHAIN ASSEMBLY IN ARMoured FACE CONVEYOR

Jiang, S. B.^{*}; Zhang, X.^{*}; Gao, K. D.^{*}; Gao, J.^{**, #}; Wang, Q. Y.^{*} & Hidenori, K.^{***}

^{*} College of Mechanical and Electronic Engineering, Shandong University of Science and Technology, Qingdao 266590, China

^{**} College of Electronics, Communication and Physics, Shandong University of Science and Technology, Qingdao 266590, China

^{***} PSC Co., Ltd., Kanagawa-ken, 225-0013, Japan

E-Mail: imgj@sust.edu.cn ([#] Corresponding author)

Abstract

For armoured face conveyors (AFC) in coal mines, fracture and jam phenomena of the chain are common failure patterns in the chain assembly. These failure patterns are caused by chain's severe vibrations from uneven loads on the conveyor. However, due to limitations of harsh operating environment in coal mines, performing underground experiments is difficult to obtain real vibration signals. Multi-body dynamic simulation is an efficient approach to analyse the complex dynamic behaviour of chain assembly in the AFC. In order to determine the actual dynamic properties of chain assembly in the AFC under different operating conditions, multi-body simulation was used to analyse the vibration properties of the chain assembly. In this study, theoretical analysis of contact force between horizontal and vertical rings, and between sprocket wheel and rings in the chain assembly was initially performed. Rigid and rigid-flexible coupling models of the chain assembly were then established. Dynamic simulations through two types of models, that is, rigid and rigid-flexible coupling models were conducted under full-, half- and empty load conditions using the ADAMS software. Trends of contact force, stress, and velocity of horizontal and vertical rings under various working conditions were obtained, and vibration properties of the chain assembly were analysed based on the corresponding curves. Results indicate that the maximum velocity is 1.75 m/s in the rigid simulation, whereas the maximum velocity is 3.0 m/s in the rigid-flexible coupling simulation. The rigid-flexible coupling method is proven to be more accurate and feasible in describing the dynamic properties of the chain assembly than the rigid method. The proposed method should be preferentially utilized in performing multi-body dynamic simulation of the chain assembly in the AFC. This study provides references for the structural optimization and design of the chain assembly in mining AFC. (Received in January 2017, accepted in May 2017. This paper was with the authors 2 months for 1 revision.)

Key Words: Vibration Properties, Multi-Body Dynamics, Chain Assembly, Armoured Face Conveyor

1. INTRODUCTION

The armoured face conveyor (AFC) is one of the important coal transporting means in coal mines. It also plays a significant and unique role in intelligent coal mining equipment group in mechanized coal mining face [1, 2]. The reliability and stability of the AFC can determine the efficiency and safety of a coal mine to a great extent. Compared with ordinary chain assembly, the chain assembly in AFC is a more complex system with multi-bodies that comprise numerous horizontal and vertical rings, and two sprocket wheels, that is, head and tail sprocket wheels. Under working conditions in coal mines, rings are in contact with nearby rings and sprocket wheels, thus causing complex contact problems. These contact problems can then influence the dynamic and vibration properties of an AFC.

Statistical data on malfunctions in mining face show that AFC malfunctions account for a large proportion, most of which involving chain assembly malfunctions [3]. In chain assembly failures in the AFC, most malfunctions mainly involve jam and fracture of the chain. The jam

and fracture failures of the chain assembly are caused by complex operating conditions in coal mines, that is, the uneven and unbalanced loads on the conveyor. The uneven loads at different times lead to the irregular vibration of the chain. The jam and fracture failures of the chain can be ascribed to the dynamic properties of the chain assembly. Thus, investigating the dynamic properties of the chain assembly under real working conditions is helpful to overcome jam and fracture failures. Current approaches include underground experiment and numerical simulation [4-6]. However, on the one hand, limited by the harsh explosion-proof environment in coal mines, most sensors cannot be directly installed on the ring or sprocket wheel in the chain assembly. As such, the dynamic signals of the chain assembly under real operating conditions cannot be accurately acquired. On the other hand, the length of the chain assembly, which is usually hundreds of meters, is difficult to model in 3D software. The entire flexible simulation analysis is time consuming and has low efficiency. Therefore, an efficient and detailed simulation of the multi-body dynamic properties of chain assembly in the AFC is essential to analyse its complex dynamic behaviour.

2. STATE OF THE ART

Regarding the dynamic simulation and experiment of chain assembly in the AFC, plenty of scholars have conducted significant and thorough studies. Schaefer [7] established mutual movement theory of chain and sprocket wheel, and indicated that the deformation of rings on a chain can influence the meshing condition between the chain and sprocket wheels. However, the research of Schaefer was theoretical, and the established models remained unverified by experiments or dynamic simulations. Świder et al. [8] conducted numerical simulations and analysed uneven load properties of the AFC. The co-simulation method using multi-body systems with MATLAB Simulink was proven to be useful in determining dynamic properties under different statuses, but the dynamic properties were indirectly reflected by the current signal. Morro and Benati [9] investigated flexible chain systems and established a model with a number of degrees of freedom to describe the rotation and flexibility of the chain system. Their research helped establish the explicit dynamic equations for well-defined geometrical parameters. However, if the geometrical parameters were not well provided, then the model was inapplicable. Kreimer [10] investigated the chain conveyors of loaders and indicated that the service life of chain assembly in the AFC was short under complex working conditions. Dynamic loads were attributed to the oscillation of the driving head, and oscillation is an important characteristic of the AFC. Suweken and Van Horssen [11] considered the weakly nonlinear transversal vibrations of a conveyor belt. Based on the approach of Kirchhoff, they derived a single motion equation from a coupled system to describe the longitudinal and transversal vibrations of the belt. Conveyor belt studies were significant for the chain conveyor.

Furthermore, Jiao et al. [12] used MSC.ADAMS software to perform simulations under different working conditions to determine the dynamic characteristics of chain assembly in the AFC, and investigated the maximum contact force and its corresponding position. The proposed method provided a reference for finite element analysis and structural optimization. Their research was significant; however, they did not perform stress and strain analysis. He et al. [13] analysed the longitudinal dynamic behaviour of chain assembly in the AFC using the Abaqus software. Dynamic simulations under full-load start-up and chain jam were performed, and the dynamic tension and velocity of the chain were determined. However, their research did not involve flexible or rigid-flexible simulations. Thus, the results of rigid simulations were inaccurate to some extent. Luo [14] conducted fatigue failure analysis of the round link chain in the AFC and concluded that the inner side of the ring between curved and cylindrical sections was in tensile stress status under pure tensile conditions. This weak point

was partly caused by the chain vibrations. Moreover, this result could provide a reference for one specification of the chain. However, for other specifications, the proposed method was inapplicable. Wu et al. [15] presented a novel conveyor with actuated parallel manipulators and proposed a method to determine the maximum dynamic load-carrying capacity of the conveyor. However, this method was only from kinematic, not dynamic perspective; stress and strain information was not obtained.

The abovementioned achievements indicate that most studies on simulation approaches are incomplete and scholars tend to perform rigid, not flexible or rigid-flexible simulations, to determine the dynamic properties of the chain assembly in the AFC. Meanwhile, contact problems in the chain assembly simultaneously exist between horizontal and vertical rings, between rings and sprocket wheels, and between rings and chutes. With regard to this chain system, a multi-body dynamic contact analysis is efficient and helpful to describe the complex vibration behaviour during different working conditions [16, 17]. Therefore, theoretical analysis of contact force in the chain assembly is conducted in this study. Rigid and rigid-flexible coupling models are respectively constructed to perform dynamic simulations under full-, half-, and empty load conditions in the ADAMS software. The contact force curves of the horizontal and vertical rings under different working conditions are obtained and analysed to thoroughly investigate the dynamic properties of the chain assembly.

The remainder of this study is organized as follows. Section 3 establishes the 3D model of the chain assembly, describes the mathematical equation model of rings, and performs a theoretical analysis of contact force between horizontal and vertical rings and between sprocket wheel and ring. Section 4 presents the simulation results for full-, half-, and empty load conditions using rigid and rigid-flexible coupling methods. Section 5 summarizes the conclusions.

3. METHODOLOGY

3.1 Modelling of chain assembly in the AFC

The type SGZ1000/1400 AFC is used to conduct the analysis. The chain assembly in the AFC comprises many horizontal and vertical rings, and two sprocket wheels. The horizontal and vertical rings are individually connected. A horizontal ring is processed by welding technology with a circle section. Meanwhile, a vertical ring is processed by forging technology with a non-circular section. The contact area between the chute and the vertical ring is usually forged as a plane to enhance the wear performance and smoothness of the vertical ring when moving in the chute. For modelling convenience, the section of a horizontal ring must be equivalent to a circle. The sprocket wheel model with seven teeth is created according to standards.

Figure 1: Models of the rings and sprocket wheel.

Figs. 1 a-c show the models of the horizontal and vertical rings and a sprocket wheel in the chain assembly, which are respectively established using the SolidWorks software. Fig. 2 shows the 3D model of the type SGZ1000/1400 AFC. Compared with real conveyor, the established SGZ1000/1400 model in this study is shortened and simplified to ensure an efficient analysis.

Figure 2: Chain assembly model of the type SGZ1000/1400 AFC.

3.2 Theoretical analysis of contact force between horizontal and vertical rings

Fig. 3 shows the knot geometry of a ring, which consists of two semicircle rings and two cylinders. A Cartesian coordinate system is also established at the centre of the ring. The spatial mathematical equation can be obtained as below in (1).

Figure 3: Force and geometry of a ring knot in the chain assembly.

$$\left\{ \begin{array}{l} \left[\left(x - \frac{t-a}{2} \right) - \frac{\left(x - \frac{t-a}{2} \right) R}{\sqrt{\left(x - \frac{t-a}{2} \right)^2 + y^2}} \right]^2 + \left[y - \frac{yR}{\sqrt{\left(x - \frac{t-a}{2} \right)^2 + y^2}} \right]^2 + z^2 = r^2 \quad x > \frac{t-a}{2} \\ \left[\left(x + \frac{t-a}{2} \right) - \frac{\left(x + \frac{t-a}{2} \right) R}{\sqrt{\left(x + \frac{t-a}{2} \right)^2 + y^2}} \right]^2 + \left[y - \frac{yR}{\sqrt{\left(x + \frac{t-a}{2} \right)^2 + y^2}} \right]^2 + z^2 = r^2 \quad x < \frac{t-a}{2} \\ y = \sqrt{z^2 + r^2} + \left(r + \frac{a}{2} \right) \quad -\frac{t-a}{2} < x < \frac{t-a}{2} \\ y = -\sqrt{z^2 + r^2} - \left(r + \frac{a}{2} \right) \quad -\frac{t-a}{2} < x < \frac{t-a}{2} \end{array} \right. \quad (1)$$

where, x , y , and z are coordinates of a point on the ring; r and R are the outer and middle radii of the semicircle ring, respectively; a is the inner diameter of the semicircle ring; l is the half-length of the cylinder; and t is the total inner length of a ring. A chain comprises numerous horizontal and vertical rings that are individually connected. If the set contact point between a horizontal and vertical ring is the coordinate origin, then the equations of a pair of contact semicircle rings can be established according to Eqs. (2) and (3), as follows.

$$\left[(x+R-r) - \frac{(x+R-r)R}{\sqrt{(x+R-r)^2 + y^2}} \right]^2 + \left[y - \frac{yR}{\sqrt{(x+R-r)^2 + y^2}} \right]^2 + z^2 = r^2 \quad (2)$$

$$\left[(x-R+r) - \frac{(x-R+r)R}{\sqrt{(x-R+r)^2 + y^2}} \right]^2 + \left[z - \frac{zR}{\sqrt{(x-R+r)^2 + y^2}} \right]^2 + y^2 = r^2 \quad (3)$$

The type of contact between horizontal and vertical rings is called an elastic contact, and no rigid movement occurs between horizontal and vertical rings. Thus, force distribution and pressure on the contact point can be calculated by Hertz contact theory. Given that the curved section geometry of a horizontal ring is strictly the same as that of a vertical ring, distributions of contact force and pressure are significantly symmetrical along the y - and z -axes. The contact problem between horizontal and vertical rings can then be converted to a contact problem between a neutral plane and a semicircle ring. The equation of the curved section of a horizontal ring can be expressed as follows.

$$x = \sqrt{r^2 - y^2 - z^2 + R^2 + 2R\sqrt{r^2 - z^2}} \quad (4)$$

If $x = F(y, z)$, then:

$$F(y, z) = (R+r) - \sqrt{r^2 - z^2 - y^2 + R^2 + 2R\sqrt{r^2 - z^2}} \quad (5)$$

where $F(y, z)$ is the function of x . According to the Taylor formula, $F(y, z)$ can be expanded to the second order. Given that $F(0, 0) = 0$ and $\left. \frac{\partial F}{\partial z} \right|_{z=0} = \left. \frac{\partial F}{\partial y} \right|_{y=0} = 0$, the following equation can be obtained.

$$F(y, z) = \frac{1}{2} [z^2 F_{zz}(0, 0) + 2zy F_{zy}(0, 0) + y^2 F_{yy}(0, 0)] = \frac{y^2}{2r} + \frac{z^2}{2(R+r)} \quad (6)$$

Based on the assumption that $\alpha = 2r$ and $\beta = 2(R+r)$, the previous equation can be simplified as follows.

$$x = \frac{y^2}{\alpha} + \frac{z^2}{\beta} \quad (7)$$

This equation represents an ellipse with its vertex in $(0, 0)$, where the contact area is also an ellipse. According to Hertz theory and contact mechanics [18], the maximum stress in the contact area is $\varepsilon_{0\max}$, which is expressed as follows.

$$\varepsilon_{0\max} = \sqrt{\frac{\frac{F}{B} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2} \right)}{\pi \left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2} \right)}} = \frac{3F}{2\pi\alpha\beta} \quad (8)$$

where, F is the contact force between two rings; ρ_1 and ρ_2 denote the radius of curvature of the two rings; E_1 and E_2 are the elastic moduli; B is the length of the contact line; and μ_1 and μ_2 are the Poisson's ratios.

According to the previous equation, the contact stress distribution in the contact areas can be expressed as follows.

$$\varepsilon_{(y,z)} = \varepsilon_{0\max} \sqrt{1 - \left(\frac{y^2}{\alpha} + \frac{z^2}{\beta} \right)} \quad (9)$$

When a ring moves in the chute, the elastic strain will be produced by the pull of the nearby rings. The stress distribution status of a ring at this time can be obtained as follows: the outer areas of two semicircle rings endure the pull force, whereas the inner areas endure compression and shear forces; the outer areas of cylinders are compressed, whereas the inner areas of cylinders are expanded. The stress concentration position lies inside the curved section.

3.3 Theoretical analysis of contact force between sprocket wheel and ring

In the chain assembly of the AFC, the dynamic relationship between the chain sprocket wheel and the ring is actually a contact mechanics problem between the horizontal ring and sprocket wheel given that the vertical ring does not come in contact with the sprocket wheel. Relative sliding occurs during the meshing process between the horizontal ring and sprocket wheel nest; meanwhile, elastic deformation occurs. Therefore, the contact force timely changes during the rotation of the sprocket wheel.

The nonlinear elastic deformation relationship between the horizontal ring and sprocket wheel nest can be described by Hertz contact theory. Based on the assumption that the curvature radii of the contact point of the horizontal ring and sprocket wheel nest are R_{h1} , R_{h2} , R_{l1} , and R_{l2} , the following equations can be derived:

$$\Sigma \rho = \frac{1}{2} \left(\frac{1}{R_{h1}} + \frac{1}{R_{h2}} + \frac{1}{R_{l1}} + \frac{1}{R_{l2}} \right) \quad (10)$$

$$F(\rho) = \frac{\left[\left(\frac{1}{R_{h1}} - \frac{1}{R_{h2}} \right) + \left(\frac{1}{R_{l1}} - \frac{1}{R_{l2}} \right) \right]}{\left(\frac{1}{R_{h1}} + \frac{1}{R_{h2}} + \frac{1}{R_{l1}} + \frac{1}{R_{l2}} \right)} \quad (11)$$

where $\Sigma \rho$ is the sum of curvature radii, and $F(\rho)$ is the curvature radii difference.

According to Hertz contact theory, the relationship between Hertz contact force and elastic deformation along the normal line of contact point between ring and sprocket wheel can be expressed as follows:

$$\delta = \frac{\delta^*}{2} \sqrt[3]{\left[\frac{\Sigma \rho}{2} \left(\frac{1 - \mu_1^2}{E_1} + \frac{1 - \mu_2^2}{E_2} \right) F_C \right]^2} = \sqrt[3]{\left(\frac{F_C}{K_C} \right)^2} \quad (12)$$

where δ is elastic deformation; F_C is Hertz contact force; K_C is contact stiffness coefficient; and δ^* is the function of $F(\rho)$. Therefore, the nonlinear normal contact F_n force between the ring and sprocket wheel can be expressed as follows:

$$F_n = K_C \delta + \left(\delta_n^{ij}, 0, 0, \delta_{\max}, c_{\max} \right) \frac{\delta_n'^{ij}}{\delta_n'^{ij}} \left| \delta_n'^{ij} \right|^{m_1} \cdot (\delta_n^{ij})^{m_2} \quad (13)$$

where δ_n^{ij} is the relative displacement along the normal direction of the contact point; $\delta_n'^{ij}$ is the normal relative velocity; $\delta_{\max}$ is the maximum contact elastic deformation; m_1 and m_2 are the nonlinear relationship and damp correction index, respectively; and $c_{\max}$ is the maximum contact damp when the penetration value is up to $\delta_{\max}$.

3.4 Load definitions of simulations

In a mechanized mining face, the load of the AFC changes with position of the shearer. Therefore, three condition representatives, that is, full-, half-, and empty loads are selected to simulate the dynamic properties of the chain assembly. The material density of scrapers and

friction coefficient between scrapers and chutes are determined to simulate different working conditions and achieve the dynamic properties of chain assembly.

(1) Full-load condition

The static calculation of the type SGZ1000/1400 AFC shows that its linear density under full-load condition is 0.42 t/m. The full length of the type SGZ1000/1400 AFC is more than 200 m, which is harmful and unnecessary for the simulation. Therefore, the model length is shortened to ensure an efficient simulation. After the simplification, the density of each scraper should be adjusted to $1.51 \times 10^5 \text{ kg/m}^3$. The friction coefficients between scraper and plate/backplane must be defined to simulate the different resistances of the load and non-load sides.

(2) Half-load condition

When the shearer moves to the middle position of the AFC, the AFC is under its half-load condition. Furthermore, its linear density of the AFC is still 0.42 t/m, whereas its transmission length is shortened to half. Therefore, the density of each scraper should be adjusted to $0.86 \times 10^5 \text{ kg/m}^3$.

(3) Empty load condition

When the shearer moves to the head position of the AFC, the transmission length is shortened, and the AFC is under its empty load condition. At this time, the density of each scraper should be adjusted to $0.177 \times 10^5 \text{ kg/m}^3$.

The ADAMS software is used to perform the contact simulation. The simulation time is set to 2 s, with each step equivalent to 0.004 s. Given that contact simulation is strictly nonlinear and discontinuous, G-stiff is selected as the solution integrator type, whereas I3 is selected as the formulation type. Integration error is set to 0.01. These measures can ensure the efficiency and stability of contact simulation. For the remaining parameters, such as maximum iteration and correction type, their default settings are used. With these settings, simulations under different conditions are conducted to obtain the contact dynamic properties of the chain assembly.

4. RESULT ANALYSIS AND DISCUSSION

4.1 Rigid simulation analysis under full-load condition

In a rigid dynamic simulation, the contact force is an important element used to describe the dynamic properties of the chain assembly. Contact force curves change with time under a full-load condition can be directly obtained by the PostProcessor in the ADAMS software. The filtering operation is then performed to eliminate interferences.

Figure 4: Curves of contact force between horizontal rings and head sprocket wheel's nest under a full-load condition.

Figure 5: Curves of contact force between horizontal rings and tail sprocket wheel's nest under a full-load condition.

The contact force curves between horizontal rings and the nest of head sprocket wheel are presented in Fig. 4. The curves indicate that during the rotation of horizontal rings around the sprocket wheel, the contact force on a horizontal ring increases first. The contact force then reaches its maximum when its subsequent near-horizontal ring starts to mesh with the head sprocket wheel when the value is approximately 300 kN. After rotating on one tooth, the contact force value then decreases and maintains a value of 100 kN. With the rotation of the sprocket wheel, the value continues to decrease. When the horizontal ring separates from the mesh with the head sprocket wheel, the contact force then decreases to 0 kN. The contact force curves of the horizontal rings with the sprocket wheel evidently appear periodically.

Fig. 5 shows the contact force curves between horizontal rings and the nest of tail sprocket wheel. This figure also shows that the trend of the tail contact force curve is evidently distinct from that of the head. The maximum contact force on the horizontal ring is 185 kN, which then decreases to 0 kN. This decrease is caused by the minimum tension between the chain rings in its tail part. Considering the head sprocket wheel, if tension exists between tail rings, then the contact force is concluded to decrease to a nonzero value and is maintained for a certain time. The value will then decrease to 0 kN. Therefore, tension plays a significant role in the contact force between rings and sprocket wheel. Adjusting the contact force becomes possible by adjusting the tension between tail and sprocket wheel.

Figure 6: Curve of the contact force between horizontal and vertical rings under full-load condition.

The curve of the contact force between horizontal and vertical rings during the simulation is illustrated in Fig. 6. Given that the chain is loose, the contact force value remains at 0 kN. With the rotation of the sprocket wheel, the chain becomes tensile, and the contact force increases to approximately 300 kN. This value is maintained for approximately 0.75 s. As the ring starts to separate from the sprocket wheel, the contact force decreases and maintains a value of approximately 120 kN, wherein the rings are in a state of return strike. The contact force value maintains a value of approximately 120 kN afterward due to the tension between horizontal and vertical rings.

4.2 Rigid simulation analysis under half-load condition

The load parameters are modified to conduct the simulation under half-load condition. After the filtering operation, the contact force curves of the head and tail are obtained, as shown in Figs. 7 and 8.

Figs. 7 and 8 show that the trends of the contact force curves of the head and sprocket wheel under half-load condition are similar to those under the full-load condition, as shown in Figs. 4 and 5. The differences lie in the maximum force values. The maximum contact force in the head sprocket wheel is approximately 170 kN, whereas the maximum contact force in the tail sprocket wheel is approximately 120 kN.

Figure 7: Curves of contact force between horizontal rings and head sprocket wheel nest under half-load condition.

Figure 8: Curves of contact force between horizontal rings and tail sprocket wheel nest under half-load condition.

4.3 Rigid simulation analysis under empty load condition

A simulation under empty load condition is similarly performed, and the curves of contact force are also obtained. Given no significant difference between contact force of the head and sprocket wheels under this condition, only the contact force curve of the head sprocket wheel is obtained, as shown in Fig. 9.

The curves of the contact force between horizontal and vertical rings under empty load condition are illustrated in Fig. 10. Figs. 9 and 10 show that the maximum contact force between the horizontal ring and head sprocket wheel is measured at 38 kN, while the maximum contact force between horizontal and vertical rings is measured at 43 kN. Under this condition, when a horizontal ring starts to rotate around the sprocket wheel, a relative rotation between horizontal and near-vertical rings occurs. During this process, the velocity of the sprocket wheel changes into kinetic energy of the horizontal ring, thereby increasing the contact force between rings. After the horizontal ring separates from the sprocket wheel, the contact force decreases and maintains stability.

Figure 9: Curves of contact force between horizontal rings and head sprocket wheel nest under empty load condition.

Figure 10: Curves of contact force between horizontal and vertical rings under empty load condition.

4.4 Multi-body rigid-flexible coupling dynamic simulation analysis of chain assembly

In the above subsections, all the rings and sprocket wheels are regarded as rigid bodies when performing multi-body dynamic simulations. The deformations of all parts during the

simulation, which do not show stress and strain, are ignored [19]. The yield stress of a ring is less than that of a sprocket wheel, and contact areas are usually small. This phenomenon can influence the accuracy of multi-body simulations. Therefore, a rigid-flexible dynamic simulation, that is, the primary parts are regarded as flexible bodies whereas the secondary parts are regarded as rigid bodies, is essential [20]. This simulation is significant in increasing the accuracy and efficiency of the simulation.

In this study, the AutoFlex Module in the ADAMS software is used for the discrete analysis of one horizontal ring and two vertical rings by the finite element method. Rigid parts of the three rings will be replaced by flexible parts, while other parts in the chain assembly remain as rigid bodies. The rigid-flexible coupling model is shown in Fig. 11. One horizontal ring (#21) and two vertical rings (#20 and #22) are flexible bodies, whereas the other rings and the sprocket wheel are rigid bodies.

Figure 11: Rigid-flexible coupling model of the chain assembly.

In rigid-flexible coupling analysis, boundaries are similar to those in rigid simulations. Furthermore, the contact type between flexible horizontal and vertical rings should be notably set as “flex body to flex body” in the ADAMS software, whereas the contact type between flexible horizontal rings and rigid vertical rings/sprocket wheel should be set as “flex body to solid”.

The von Mises stress of the rings changes with time, as shown in Fig. 12. The maximum stress of a horizontal ring is larger than that of a vertical ring because vertical ring geometry optimization decreases the stress concentration. These results coincide with real-life practice, wherein the conditions of a horizontal ring are worse than those of a vertical ring.

Figure 12: Curves of maximum stress on rings #20, #21, and #22 with time.

The velocities in the X- and Y-directions of a flexible horizontal ring (#21) and those of a rigid horizontal ring (#16) in rigid simulations are shown in Figs. 13 and 14, respectively, to further investigate the vibration and shock properties of the chain assembly. Figs. 13 and 14 respectively show that the horizontal ring separates from the sprocket wheel and the velocities in the X- and Y-directions severely vibrate when the time is 1.5 s. By comparison, the results of rigid-flexible simulations are more accurate and reliable than those of rigid simulations. During the rotation time of the horizontal ring around the sprocket wheel, velocity fluctuation is smoothened. However, velocity in the Y-direction severely vibrates during translational movement. This phenomenon can be attributed to two reasons: the horizontal ring is unsupported by the chute, and the polygon affects the chain assembly. Furthermore, velocity fluctuation causes deformations, which result in evident vibration and a larger amplitude in a flexible ring than that in a rigid ring. Therefore, rigid-flexible simulations should be selected first for dynamic properties of chain assembly in the AFC.

Figure 13: Velocities in the X- and Y-directions in the flexible horizontal ring #21 (rigid-flexible simulation).

Figure 14: Velocities in the X- and Y-directions in the rigid horizontal ring #16 (rigid simulation).

5. CONCLUSION

The chain assembly model of an AFC was established, and multi-body rigid and rigid-flexible coupling simulations of the chain assembly were performed in the ADAMS software to determine its dynamic properties under different operating conditions. Results of rigid and rigid-flexible coupling simulations were analysed and compared. The following conclusions are drawn.

(1) In multi-body rigid simulations, the contact force trend is similar under full- and half-load conditions. However, the maximum contact force is different under these two load conditions. Under the empty load condition, the vibration properties are obviously different from those under full- and half-load conditions.

(2) When the horizontal ring starts to rotate around the sprocket wheel, the relative rotation between the horizontal and the near-vertical rings has significant influences on the vibration properties of the chain assembly.

(3) In multi-body rigid-flexible dynamic simulation, the maximum stress of a horizontal ring is larger than that of a vertical ring because vertical ring geometry optimization decreases the stress concentration.

(4) The maximum velocity of the horizontal ring by rigid-flexible coupling simulation is 3.0 m/s, which is close to the actual velocity data obtained through observation and is 71.4 % larger than the maximum velocity (1.75 m/s) of the horizontal ring by rigid simulation. This

finding indicates that the results of rigid-flexible simulations are more accurate and reliable than those of rigid simulations.

The proposed rigid-flexible coupling simulation method is proven to be efficient and feasible. Researches in this study provide references for structural optimization and vibration analysis of chain assembly in an AFC. Compared with a full rigid simulation, the rigid-flexible coupling simulation should be preferentially used when performing multi-body dynamic simulations. However, limited by the current hardware condition, a full flexible simulation can produce massive element data, which involves a considerable amount of calculation time. Therefore, a full flexible simulation with appropriate simplification should be performed to obtain accurate results and improve the calculation efficiency.

ACKNOWLEDGEMENT

The work was supported by the National Natural Science Fund of China (Grant No. 51375282 and 51406106), the Science and Technology Development Program of Shandong Province (Grant No. 2014GGX103043), the Postdoctoral Science Foundation Funded Project of China (Project No. 2016M592214), and the Postgraduate Innovation Fund of Shandong University of Science and Technology (Grant No. SDKDYC170106).

REFERENCES

- [1] Nie, R.; He, B. Y.; Yuan, P. F.; Zhang, L. H.; Li, G. P. (2015). Novel approach to and implementation of design and analysis of armored face conveyor power train, *Science China Technological Sciences*, Vol. 58, No. 12, 2153-2168, doi:[10.1007/s11431-015-5895-z](https://doi.org/10.1007/s11431-015-5895-z)
- [2] Hu, K.; Zhang, C. J.; Wang, S.; Han, S. (2017). Optimization of scraper conveyor extensible tail PID control system based on improved TLBO algorithm, *Journal of Central South University (Science and Technology)*, Vol. 48, No. 1, 106-111, doi:[10.11817/j.issn.1672-7207.2017.01.015](https://doi.org/10.11817/j.issn.1672-7207.2017.01.015)
- [3] Wang, J. H.; Yu, B.; Kang, H. P.; Wang, G. F.; Mao, D. B.; Liang, Y. T.; Jiang, P. F. (2015). Key technologies and equipment for a fully mechanized top-coal caving operation with a large mining height at ultra-thick coal seams, *International Journal of Coal Science & Technology*, Vol. 2, No. 2, 97-161, doi:[10.1007/s40789-015-0071-4](https://doi.org/10.1007/s40789-015-0071-4)
- [4] Ordin, A. A.; Metel'kov, A. A. (2015). Analysis of longwall face output in screw-type cutter-loader-and-scraper conveyor system in underground mining of flat-lying coal beds, *Journal of Mining Science*, Vol. 51, No. 6, 1173-1179, doi:[10.1134/S1062739115060452](https://doi.org/10.1134/S1062739115060452)
- [5] Yao, Y. P.; Meng, W. J.; Kou, Z. M.; Liao, Z. C. (2015). Resistance analysis for bulk materials in the vertical section of pipe scraper conveyor, *Proceedings of China Modern Logistics Engineering*, Lecture Notes in Electrical Engineering, Vol. 286, 327-335, doi:[10.1007/978-3-662-44674-4_31](https://doi.org/10.1007/978-3-662-44674-4_31)
- [6] Wang, X. W.; Wang, S. P.; Long, R. S.; Yang, Z. J.; Liu, G. P. (2016). Rigid-flexible coupled dynamic contact analysis for a chains drive system of a heavy scraper conveyor during loading start-up, *Journal of Vibration and Shock*, Vol. 35, No. 11, 34-40, doi:[10.13465/j.cnki.jvs.2016.11.006](https://doi.org/10.13465/j.cnki.jvs.2016.11.006)
- [7] Schaefer, W. (1976). Spacing links of round steel chains from the chain sprockets axis, *Glueckauf-Forschungshefte*, Vol. 37, No. 6, 253-258
- [8] Świder, J.; Herbuś, K.; Szewerda, K. (2015). Analysis of load unevenness of chain conveyor's driving motors on the basis of numerical simulations, *Journal of Achievements in Materials and Manufacturing Engineering*, Vol. 73, No. 2, 165-175
- [9] Morro, A.; Benati, M. (1988). A model for the dynamics of flexible chain systems, *Mathematical and Computer Modelling*, Vol. 11, 1093-1096, doi:[10.1016/0895-7177\(88\)90663-2](https://doi.org/10.1016/0895-7177(88)90663-2)
- [10] Kreimer, V. I. (1965). Investigation of chain conveyors of loaders, *Soviet Mining Science*, Vol. 1, No. 2, 129-135, doi:[10.1007/BF02501939](https://doi.org/10.1007/BF02501939)
- [11] Suweken, G.; Van Horssen, W. T. (2003). On the weakly nonlinear, transversal vibrations of a conveyor belt with a low and time-varying velocity, *Nonlinear Dynamics*, Vol. 31, No. 2, 197-223, doi:[10.1023/A:1022053131286](https://doi.org/10.1023/A:1022053131286)

- [12] Jiao, H. Z.; Yang, Z. J.; Wang, S. P. (2012). Contact dynamics simulation analysis for sprocket transmission system of scraper conveyor, *Journal of China Coal Society*, Vol. 37, No. S2, 494-498, doi:[10.13225/j.cnki.jccs.2012.s2.028](https://doi.org/10.13225/j.cnki.jccs.2012.s2.028)
- [13] He, B. Y.; Sun, Y. H.; Nie, R.; Li, G. P. (2012). Dynamic behavior analysis on the ring chain transmission system of an armoured face conveyor, *Journal of Mechanical Engineering*, Vol. 48, No. 17, 50-56
- [14] Luo, Q. J. (1997). Analysis of fatigue failure of high strength round link chain for mine use, *Journal of China Coal Society*, Vol. 22, No. 4, 429-433, doi:[10.13225/j.cnki.jccs.1997.04.020](https://doi.org/10.13225/j.cnki.jccs.1997.04.020)
- [15] Wu, J.; Chen, X. L.; Wang, L. P.; Liu, X. J. (2014). Dynamic load-carrying capacity of a novel redundantly actuated parallel conveyor, *Nonlinear Dynamics*, Vol. 78, No. 1, 241-250, doi:[10.1007/s11071-014-1436-8](https://doi.org/10.1007/s11071-014-1436-8)
- [16] Wittenburg, J. (2008). *Dynamics of Multibody Systems*, 2nd edition, Springer-Verlag, Berlin
- [17] Wittenburg, J. (author); Likins, P. (reviewer) (1978). Dynamics of systems of rigid bodies, *Journal of Applied Mechanics*, Vol. 45, No. 2, 458, doi:[10.1115/1.3424342](https://doi.org/10.1115/1.3424342)
- [18] Wriggers, P. (2006). *Computational Contact Mechanics*, 2nd edition, Springer-Verlag, Berlin, doi:[10.1007/978-3-540-32609-0](https://doi.org/10.1007/978-3-540-32609-0)
- [19] Pereira, M. S.; Proenca, P. L. (1991). Dynamic analysis of spatial flexible multibody systems using joint co-ordinates, *International Journal for Numerical Methods in Engineering*, Vol. 32, No. 8, 1799-1812, doi:[10.1002/nme.1620320816](https://doi.org/10.1002/nme.1620320816)
- [20] Li, X. F.; Wang, Y. B. (2012). Analysis of mixed model in gear transmission based on ADAMS, *Chinese Journal of Mechanical Engineering*, Vol. 25, No. 5, 968-973, doi:[10.3901/CJME.2012.05.968](https://doi.org/10.3901/CJME.2012.05.968)