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Application Of Multibody Dynamic Method (Mbd) And Mechanical Principle To The Cylinder Torque Calculation

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***Abstract*—In this report, the multibody dynamic method and the mechanical principle is used to calculate the cylinder torque when it is working. The experimental equipment is also set up to measure the torque with different angle of working cylinder. The results obtained from the mechanical principle completely matched with that of multibody dynamic method. But the results obtained from the experimental test and the mechanical principle had a constant deviation values. Therefore, the analytical equation based on the mechanical principle will add a correct factor to revise the friction effect not considered in the analytical developments. The results can match with each other between the experimental data and that of the analytical method with a correct factor.**

Keywords—Cylinder Torque, Multibody Dynamic Method, Mechanical Principle, Correct Factor

1. Research Background

In this paper, the cylinder torque at different pressures arising from different working angles is studied. In order to design a better cylinder, the central axis at different distances or different air pressure can be applied to derive a basic theory. It is also calculated by the multibody dynamic finite element software and tested by the experiment to verify the results of theoretical analysis. In experimental design, the torque is measured for only 0 degrees, 45 degrees, and 90 degrees that are found in these three angles to do a better measure, because the rest of the angle is difficultly fixed to get the data. In order to observe the torque curve for the cylinder operation, the mechanical principles and computer-aided engineering software (CAE) [1-5] are used to do analyses for all angles

2. Cylinder geometric shape

Cylinder geometric shape is drawn by using a CAD program shown in Figure 1 which basically is complicated. Therefore the actual model will be simplified, only the major parts connected to the main part affect the results of the analysis shown in Figure 2. Cylinder diameter is 250mm, the vertical distance between the cylinder center to a rocker arm is 95mm, slider skew angle of the original design is 15 degrees, but the actual skew angle found is only 14.1 degrees, and skew angle will continuously change due to the cylinder process. The length, width and height of the slider are, respectively, 42mm, 30mm, and 17.5mm. The cylinder system is made of carbon steel. The friction is not included in the analytical formula and MBD analysis that may have a deviation from the experimental tests. In the multibody dynamic finite element analysis, all the structural components in the model are assumed to be the rigid body [2, 3] that matched with the mechanical analysis assumption.

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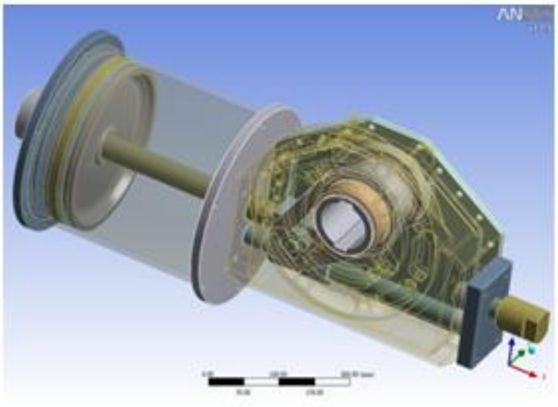

Fig.1 Original cylinder model

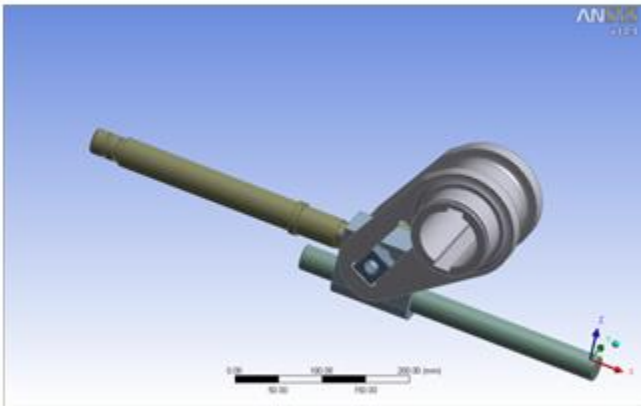

Fig.2 Simplified cylinder model

3. Experimental Test

The design of the experiment equipment is shown in Figure 3. A torque meter is used to measure torque for the slider skew angle of 0 degrees, 45 degrees and 90 degrees. First 0.5 m of iron bar above the cylinder is fixed to do support, a torque wrench is fixed near the cylinder to measure the torque. Then the cylinder begins to move when the cylinder reached the force balance, i.e., the torque is reading on the torque meter and recorded. The main thrust of 5 Bar is applied, the torque of 4861 (Nm) with 0 degrees, 2058 (Nm) for 45 degrees, and 2837 (Nm) for 90 degrees are observed, by the way the cylinder pressure is also recorded. All the results are compared with each other based on the different thrust forces because the cylinder movement is more stable for these experimental tests. The torque is measured for these three angles because the torque meter is easily fixed and measured. The cylinder motion with slider skew angle of 90 degrees is shown in Fig.4. The torque measured from the other angle is unstable because the angle measuring device is hard to fix. Therefore mechanical principle and finite element simulation analysis are needed to render the relationship between slider angle and torque.

4. Torque theory for cylinder

The relationship between skew angle and torque for the cylinder is derived to use mechanical principle [7-9]. Static equilibrium is plotted in Fig. 5 that F_2 is the applied forces, L is a minimum arm length of 0.095 m. F_1 is a back force due to the angle change and perpendicular to the cylinder axis of rotation.

Fig.3 The torque measuring equipment Fig. 4 The cylinder motion with slider skew angle of 90 degrees

5. Torque theory for cylinder

The relationship between skew angle and torque for the cylinder is derived to use mechanical principle [7-9]. Static equilibrium is plotted in Fig. 5 that F_2 is the applied forces, L is a minimum arm length of 0.095 m. F_1 is the back force due to the angle change and perpendicular to the cylinder axis of rotation.

F_2=F_1xcos(45^o-Θ_0-Θ) (1)

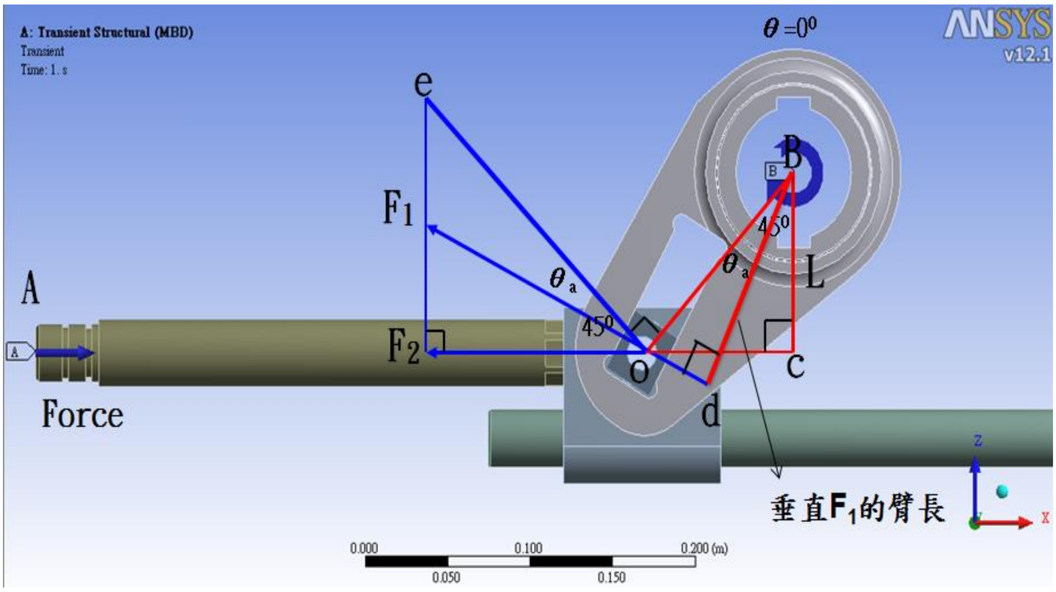

Fig. 5 Force components for static equilibrium

F_2 is the main thrust forces, F_1 is the back force due to the angle change. The skew angle Θ_0 is shown in Fig. 6.

$\theta_0 = \sin^{-1}[\sqrt{2}\sin(14.1)\cos(\theta - 45)]$ (2)

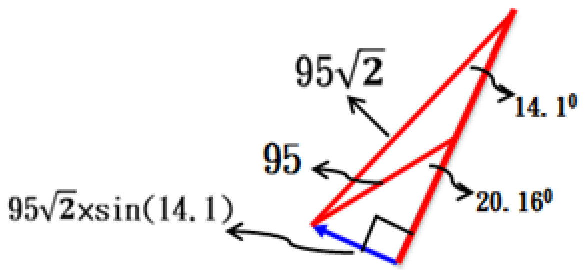

Fig.6 The skew angle decomposition

After F1 is found, the torque can be calculated from the following equations:

Torque= $F_1 \times [\mathrm{L}/\cos(45° - \theta)] \times \cos\theta_0$ (3)

In the above equation (2), the slider starts to move from 0 degree to 90 degrees and can calculate the skew angle. The structure is assumed to be a rigid body in theoretical and multibody dynamic analysis that cannot reflect the real working conditions, a correction factor that consider the friction is introduced to adjust the deviation. The factor is defined to be 3% to 20% of the thrust force [3]. The thrust force for the cylinder without consideration of friction is F_{th}=AP which P is the applied pressure and A is the cross section area. If the friction is considered, the thrust force will become the following:

F_n=AP-F_{RZ}

F_{RZ} is the friction force that is defined to be a correction factor.

6. Multibody dynamic analysis

Finite element analysis (FEA) of the main cylinder torque is studied from 0 degrees to 90 degrees and is performed with every 15 degrees. FEA can get a more holistic point of view than that of the experiment. It also can get a complete cylinder motion curve. Finite element analytical process is as follows: First the geometrical model created from CAD is imported into

ANSYS/Workbench and simplified to delete some parts that will not affect the analytical results, then the contact elements between the bodies are assigned. The thrust force (pressure) and torque are assigned to reach a static equilibrium, the detailed analytical steps are shown in Fig. 7. Multibody dynamic (MBD) method [4,5] in FEA is used to do analysis. A rigid body is selected to do analysis because friction force is neglected and can reduce the intricately contact conditions.

The deviation for the torque obtained from finite element analysis with 1 Bar pressure with that of the experimental data is 2.4% at the skew angle of 90 degrees, and the maximum deviation occurs at 0 degrees is18.9%. For the pressure of 2~6 Bar, the deviation for the torque obtained from FEA with that of the experimental data is 5.2% at the skew angle of 90 degrees, and the maximum deviation occurs at 0 degrees is18.9%. The major deviation came from the analysis without the friction. Follow-up will be multiplied by a correction factor to make the theoretical results match with the realistic situation. After this, the results obtained from the FEA are the same as that of the experimental data, it also can prove that the finite element analysis is correct.

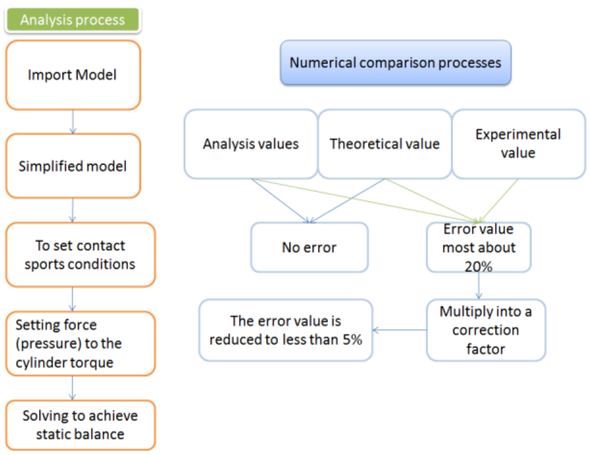

Fig.7 Analytical flow chart

7. Results and comparison

The results obtained from theoretical derivation and FEA are almost the same, but the deviation is up to 20% to compare with the experimental data. If theoretical result multiplied by a correction factor with friction consideration, deviation will be reduced to be less than 5%.

The maximum deviation of the experimental and theoretical values is 18.9% and that of minimum is 2.4% for 1 bar pressure from Table 1. The maximum deviation after correction factor considered becomes 23.4% and a minimum is 2.5%.The deviation become big, the possible reason is that the smaller the pressure measured only 1Bar, resulting in inaccurate measurement results.

The results listed in Table 2 are for 2 bars. A maximum deviation is 17% and minimum deviation is 6.3%. The maximum deviation after the correction was reduced to be 8.6%, and the smallest deviation is 2.5%, a significant decrease in apparent than in the case of 1 bar. After correction factor is considered, the maximum deviation is still at 8.6% and 5% short of our expectations of the 3.6%, so the situation here although it is not ideal to reduce deviation and the reason is the same as that of 1 bar.

Table 1 The results for 1 bar

Pressure	Angle	Simulation analysis and Theoretical value (NM) (1)	Multiply into a correction factor (NM) (2)	Experimental value (NM) (3)	(1)(2) error (%)	(2)(3) Error (%)
1Bar	0^0	1160	997	941	18.9	5.6
	15^0	706	631			
	30^0	514	469			
	45^0	434	397	407	6.2	2.5
	60^0	425	383			
	75^0	490	428			
	90^0	694	576	711	2.4	23.4

Table 2 The results for 2 bars

Pressure	Angle	Simulation analysis and Theoretical value (NM) (1)	Multiply into a correction factor (NM) (2)	Experimental value (NM) (3)	(1)(2) error (%)	(2)(3) Error (%)
2Bar	0^0	2320	1995	1926	17.0	3.5
	15^0	1413	1262			
	30^0	1028	937			
	45^0	868	793	813	6.3	2.5
	60^0	850	766			
	75^0	981	856			
	90^0	1388	1151	1250	9.9	8.6

The results shown in Tables 3 to 6, it can find the maximum deviation is 19.8% and the minimum error is 5.2% for 6 bars. After adding the correction factor, the maximum deviation for these four cases is reduced to be 4.1% for 3 bars and the smallest deviation is only 0.2% for 4 bars.

This showed that after adding the correction factor meets the test environment. It also found the analysis value will close to the measured value when the working conditions are more stable. When the pressure (the thrust force) is larger (3 bars or more), the experiments are more easily controlled. The results from the theoretical derivation and FEA are almost the same, it proves that the analytical results are correct.

Table 3 The results for 3 bars

Pressure	Angle	Simulation analysis and Theoretical value (NM) (1)	Multiply into a correction factor (NM) (2)	Experimental value (NM) (3)	(1)(2) error (%)	(2)(3) Error (%)
3Bar	0^0	3479	2991	2930	15.8	2.0
	15^0	2119	1893			
	30^0	1543	1406			
	45^0	1303	1191	1225	6.0	2.9
	60^0	1276	1150			
	75^0	1471	1284			
	90^0	2083	1727	1798	13.7	4.1

Table 4 The results for 4 bars

Pressure	Angle	Simulation analysis and Theoretical value (NM) (1)	Multiply into a correction factor (NM) (2)	Experimental value (NM) (3)	(1)(2) error (%)	(2)(3) Error (%)
4Bar	0^0	4639	3989	3969	14.4	0.5
	15^0	2826	2524			
	30^0	2057	1875			
	45^0	1737	1588	1642	5.5	3.4
	60^0	1701	1533			
	75^0	1962	1712			
	90^0	2777	2303	2298	17.2	0.2

Table 5 The results for 5 bars

Pressure	Angle	Simulation analysis and Theoretical value (NM) (1)	Multiply into a correction factor (NM) (2)	Experimental value (NM) (3)	(1)(2) error (%)	(2)(3) Error (%)
5Bar	0^0	5799	4986	4861	16.2	2.5
	15^0	3532	3155			
	30^0	2571	2344			
	45^0	2171	1985	2058	5.2	3.7
	60^0	2126	1916			
	75^0	2452	2140			
	90^0	3471	2878	2837	18.3	1.4

Table 6 The results for 6 bars

Pressure	Angle	Simulation analysis and Theoretical value (NM) (1)	Multiply into a correction factor (NM) (2)	Experimental value (NM) (3)	(1)(2) error (%)	(2)(3) Error (%)
6Bar	0^0	6959	5983	5870	15.6	1.9
	15^0	4238	3786			
	30^0	3085	2812			
	45^0	2605	2381	2470	5.2	3.7
	60^0	2551	2299			
	75^0	2942	2567			
	90^0	4165	3454	3342	19.8	3.2

The torque related to the skew angles is shown in Figs. 8-13 that the trend of three curves is the same. The maximum torque from zero degrees is observed, and then it slowly drops to a lowest values at 45 degrees, and then back to a higher values at 90 degrees. Three analytical curves with correction values are close to the experimental data. From the trend of these curves, it can estimate the torque that is not measured in the tests.

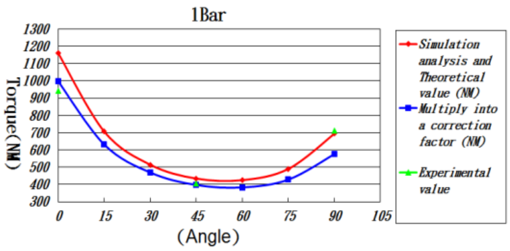

Fig.8 The relationship between torque and angle for 1 bar

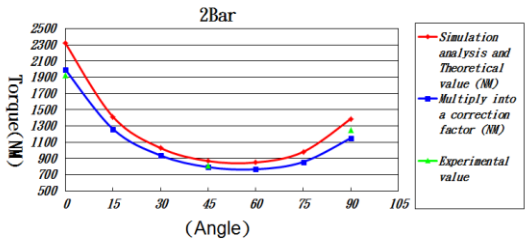

Fig.9 The relationship between torque and angle for 2 bars

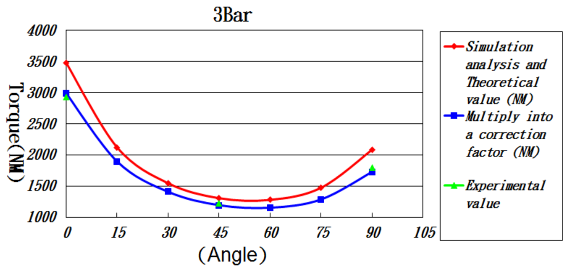

Fig.10 The relationship between torque and angle for 3 bars

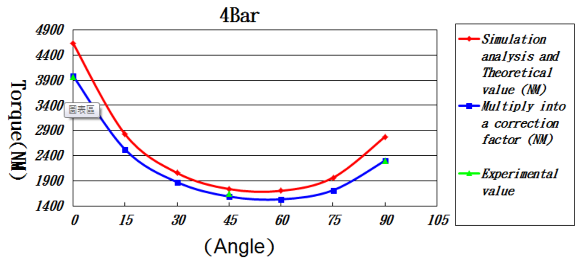

Fig.11 The relationship between torque and angle for 4 bars

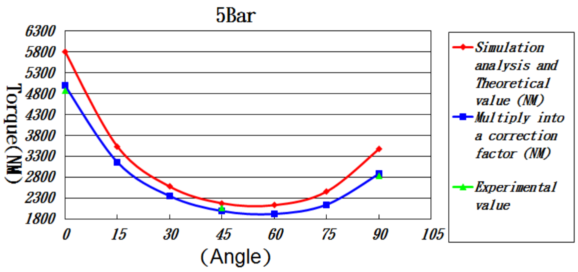

Fig.12 The relationship between torque and angle for 5 bars

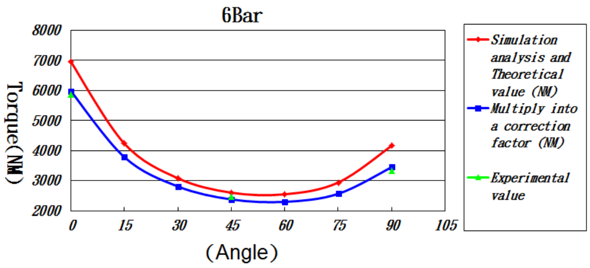

Fig.13 The relationship between torque and angle for 6 bars

8. Conclusions

This paper can get the torque of the cylinder from 1 bar to 6 bars from 0 degree to 90 degrees during the intake process, and compare with experimental data. Conclusions are as follows:

a. From the results, the results obtained from theoretical analysis and finite element analyses are the same and can fit the experimental data.

b. Before considering friction, the deviation between theoretical values and experimental data in the 0 degree and 90 degrees is up to about 20%, about 6% in the 45 degrees that the inherent of the experiments may have error and possible reason is friction. If frictions are considered into account, it will make theoretical analysis and finite element simulation becomes more complicated, so the theoretical derivation and finite element simulation does not consider the effect of friction.

c. If the theoretical formula is multiplied by a correction factor of friction (this factor made the theoretical value approach to the experimental data), such that the deviation for 3~6 bars pressure are reduced to be less than 5%.

d. It recommended that the pressure of 1 to 2 bars is to avoid the use of a cylinder, by comparison, the results of the deviation is large, it is recommended to use pressures above 3 Bars that may be more stable.

Acknowledgement

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