한국소음진동공학회 2012년 추계학술대회논문집, pp. 93~99

Optimal Design of New MR Mount for Diesel Engine of Ship
선박디젤엔진을 위한 새로운 MR 마운트의 최적설계

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Key Words : Magnetorheological Fluid(MR 유체), Engine Mount(엔진 마운트), Optimal Design(최적설계), Diesel Engine Mount(디젤엔진 마운트), Ship(선박)

ABSTRACT

This paper presents an optimal design of magnetorheological (MR) fluid based mount (MR mount in short) which can be applicable to vibration control of diesel engine of ship. In this work, a mixed-mode including squeeze mode, flow mode and shear mode is proposed and designed. In order to determine actuating damping force of MR mount required for efficient vibration control, excitation force from diesel engine is analyzed. In this analysis, a model of V-type engine is considered and the relationship between velocity and pressure of gas in torque of the piston is derived. Subsequently, by integrating the field-dependent rheological properties of commercially available MR fluid with the excitation force an appropriate size of MR mount is designed. In addition, in order to achieve maximum actuating force with geometric constraints design optimization is undertaken using ANSYS software. Through the magnetic density analysis, optimal design parameters such as bottom gap and radius of coil are determined.

요 약

이 논문은 선박디젤엔진의 진동제어에 적용할 수 있는 MR 유체기반 마운트(MR 마운트)의 최적설계를 제시한다. 이 연구에서는 압착모드, 유동모드, 전단모드를 포함하는 혼합모드가 제안되었고 설계되었다. 효과적인 진동제어를 위하여 요구되는 MR 마운트의 작동 댐핑력을 결정하기 위하여 디젤엔진의 기진력이 분석되었다. 이 분석에서 V-type 엔진이 고려되었으며 피스톤의 토크에서의 속도와 가스압력간의 관계를 유도하였다. 결과적으로 상업적으로 이용 가능한 MR 유체의 장의존적 유동특성과 기진력을 통합함으로써 적절한 MR 마운트의 크기가 설계되었다. 게다가 기하학적 제한조건이 고려된 최대 구동력을 얻기 위해 ANSYS 를 이용하여 최적설계가 수행되었다. 자기밀도분석을 통해 바닥간격과 코일의 반지름과 같은 최적설계변수가 결정되었다.

1. Introduction

A mount is normally used to support an object and suppress vibration. There are many types of mount such as passive, active, semi-active and semi-active like MR mount can be widely used to isolate the vibration at low frequency range where diesel engine mainly operates, and it can have several advantages of continuous damping control and simple design. It is well-known that there are three modes of MR fluid - flow mode, shear mode and squeeze mode which make MR fluid have some special characteristics that can change viscosity immediately when magnetic field is applied. So, those have been popularized in applications such as brakes, mounts and dampers. The flow mode is a phenomenon that fluid flows as a result of pressure gradient between two stationary plate. The shear mode is a

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phenomenon that fluid flows between two plates that move relatively. The squeeze mode relates to the fluid flows between two plates that move through perpendicular direction to their planes.

Recently, there has been an increasing interest in combining modes of MR fluid to design dampers[2,7~14]. In these researches, it is noted that squeeze mode damper can suffer high load than other modes[15~16] and this mode is applied in suspension, damper and mount with other modes[15~16,17,18,19]. To research on optimal design mount, Nguyen et al[1] studied an analytical method based on quasi-static modeling and magnetic circuit analysis. Also, Nguyen et al[4] evaluated temperature in optimization of an automotive brake based on the finite element analysis model. A method to compute the multi-mode isolator using MR fluid was built by Brigley[2], but did not give an optimal solution to design. Farjoud et al[15] and Zhang et al[16] studied by testing to find characteristic of squeeze mount that can stand high load. These works are mostly studied for MR dampers and small load mount in automotive systems. On the other hand, the MR mount for engine of ship has hardly been studied yet.

Consequently, the main purpose of this work is to propose a new type of MR mount which is application for vibration control of a huge diesel engine equipped with a ship. In order to achieve the research goal, the excitation from the diesel engine is analyzed by adopting V-type engine. Then, using the field-dependent rheological properties of MR fluid the damping force required to suppress the vibration is calculated. The initial design parameters such as length of the magnetic pole and diameter of the piston rod are determined using the governing of MR mount. In addition, in order to maximize the damping force with the geometry constraint, an optimization process is undertaken using the commercial software APDL (ANSYS Parametric Design Language). Computer simulation is undertaken in order to investigate the actuating force with respect to the distance of the piston-bottom.

2. Excitation forces from diesel engine

There are two modes in the operation of diesel engine: rigid mode and flexible mode. The rigid mode includes phenomena as rolling, pitching, bouncing, and yawing. The flexible mode has two modes, which we should concentrate on to deal with vertical bending mode and horizontal bending mode. It is noteworthy that the vertical bending mode is the most important mode to affect the ability of operation system.

There are some phenomena in operation of diesel engine which we do not desire because they have bad effect on longevity of diesel engine and human's health. Fig.1 shows the phenomena of engine. As shown in Fig.1, there are forces and moments. F_V is the vertical external force (vertical unbalanced force) and F_H is the horizontal external force (horizontal unbalanced force). M_V is the vertical external moment (vertical unbalanced moment) and M_H is the horizontal external moment (horizontal unbalanced moment). To calculate and find out the common features between force and moment transmitted to the mount, it is essential to know where a force appears and what the consequence of the moment is. A model of V-type engine for convenient computational forces is shown in Fig.2. From Fig.2, the inertia forces due to reciprocating parts of piston 1 and piston 2 along the line of stroke are expressed as[5]

$$F_{if1} = m \cdot \omega^2 \cdot R[\cos(\theta-\alpha) + \frac{R}{L_c}\cos 2(\theta-\alpha)] \qquad (1)$$

$$F_{if2} = m \cdot \omega^2 \cdot R[\cos(\theta+\alpha) + \frac{R}{L_c}\cos 2(\theta+\alpha)] \qquad (2)$$

where m is mass of reciprocating parts per cylinder (kg), ω is angular velocity of crank (rad/s), R is radius of crank

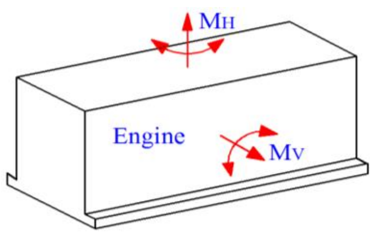

Fig.1 External forces and external moments of a diesel engine

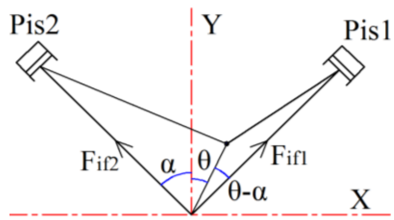

Fig.2 Model of V-piston of diesel engine

(m), θ is inclination of crank to the vertical (rad), L_c is length of connecting rod (m).

It is noted that V12 engine has 12 pistons and its arrangement is placed by 6 couples of pistons with same distance. Belonging to engine's company, these parameters are varied and confidential. So, we should find the solution to deal with this problem by using mathematical model and eliminate parameters involved with this model.

In the Eq.(1) and Eq.(2), the vertical components and horizontal components are defined as follows

$$\begin{aligned} F_V &= 2m\cdot\omega^2\cdot R(\cos\theta\cdot\cos^2\alpha\cdot\sum_1^6\cos\phi_i - \sin\theta\cdot\cos^2\alpha\cdot\sum_1^6\sin\phi_i \\ &+ \frac{R}{L_c}\cos2\theta\cdot\cos2\alpha\cdot\cos\alpha\cdot\sum_1^6\cos2\phi_i \\ &- \frac{R}{L_c}\sin2\theta\cdot\cos2\alpha\cdot\cos\alpha\cdot\sum_1^6\sin2\phi_i) \end{aligned} \tag{3}$$

$$\begin{aligned} F_H &= 2m\cdot\omega^2\cdot R(\sin\theta\cdot\sin^2\alpha\cdot\sum_1^6\cos\phi_i + \cos\theta\cdot\sin^2\alpha\cdot\sum_1^6\sin\phi_i \\ &+ \frac{R}{L_c}\sin2\theta\cdot\sin2\alpha\cdot\sin\alpha\cdot\sum_1^6\cos2\phi_i \\ &+ \frac{R}{L_c}\cos2\theta\cdot\sin2\alpha\cdot\sin\alpha\cdot\sum_1^6\sin2\phi_i) \end{aligned} \tag{4}$$

where F_V is vertical force, F_H is horizontal force, ϕ_i is firing angle of the i th-piston.

It can be easy to find the conditions for complete balance of force are applied for Eq.(4) as

$$F_V = 0;\ \ F_H = 0 \tag{5}$$

Consequently, the Eq.(5) can be rewritten as follows

$$\sum\cos\phi_i = 0;\ \sum\sin\phi_i = 0;\ \sum\cos2\phi_i = 0;\ \sum\sin2\phi_i = 0 \tag{6}$$

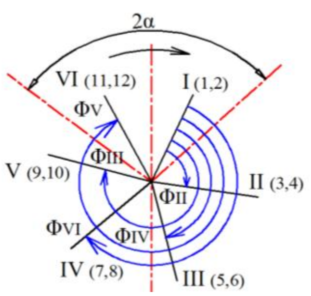

Fig.3 Operating angle of V12-type engine with V angle 2α =90

Firing angle is modeled for V12-type engine which has V angle of 90° as Fig.3. To calculate moments from forces conveniently, we can accept that the equal distance among couples of piston as shown in Fig.4. this figure shows configuration of the pistons of the V12 engine where l is distance between two couples of piston. Combining with the firing angle (Fig.3) and applying the moving force principle to the first couple of piston, vertical unbalanced moment M_V and horizontal unbalanced moment M_H can be found as

$$M_V = m\cdot\omega^2\cdot R\cdot l(-5\cos\theta + 1.72\sin\theta) \tag{7}$$

$$M_H = -m\cdot\omega^2\cdot R\cdot l(5\sin\theta + 1.72\cos\theta + 3.44\sqrt{2}\frac{R}{L_c}\cos2\theta) \tag{8}$$

To find vertical force, the relative positions between the center of the mass of engine and mounts should be noted. From this information, we can find the exact value of vertical force and use this value to design mount.

From the Fig.5, displacement of piston S is defined as

$$S = (R + \frac{R^2}{4L_c}) - R(\cos\theta + \frac{R}{4L_c}\cos2\theta) \tag{9}$$

It can be rewritten by using Maclaurin's expansion as

$$S = R(a_0 + \cos\theta + a_2\cos2\theta + a_4\cos4\theta + a_6\cos6\theta + \cdots) \tag{10}$$

Where

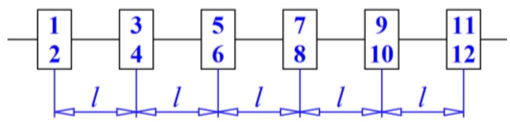

Fig.4 Diagram of distribution among couples of piston of V12-type engine

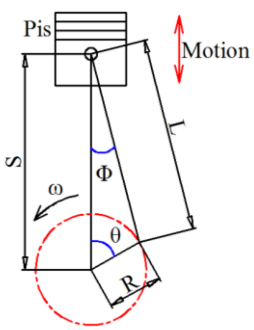

Fig.5 Model of a single piston

$$a_0 = \frac{L_c}{R}\left[1+\frac{1}{4}\left(\frac{R}{L_c}\right)^2+\frac{3}{64}\left(\frac{R}{L_c}\right)^4+\frac{5}{256}\left(\frac{R}{L_c}\right)^6\right] \quad (10a)$$

$$a_2 = \frac{L_c}{R}\left[\frac{1}{4}\left(\frac{R}{L_c}\right)^2+\frac{1}{16}\left(\frac{R}{L_c}\right)^4+\frac{15}{512}\left(\frac{R}{L_c}\right)^6\right] \quad (10b)$$

$$a_4 = -\frac{L_c}{R}\left[\frac{1}{64}\left(\frac{R}{L_c}\right)^4+\frac{3}{256}\left(\frac{R}{L_c}\right)^6\right] \quad (10c)$$

$$a_6 = \frac{L_c}{R}\left[\frac{1}{512}\left(\frac{R}{L_c}\right)^6\right] \quad (10d)$$

Relation between velocity and pressure of gas in torque T_{piston} of piston is

$$T_{piston} = P\cdot A_p\left(\frac{dS}{d\theta}\right) = P\cdot A_p\left(\frac{dS}{dt}\right)\cdot\left(\frac{dt}{d\theta}\right) = P\cdot A_p\frac{\dot{S}}{\dot{\theta}} \quad (11)$$

The Eq.(11) can be expressed as

$$T_{piston} = -P\cdot A_p\cdot R(\sin\theta + 2a_2\sin 2\theta + 4a_4\sin 4\theta) \quad (12)$$

From Eq.(12), it is easy to find the gas force to impact the mount. Again, it is noted that Eq.(7) and Eq.(12) are used for calculating dynamic force and gas force. It is assumed that the system vibration is harmonic to find acceleration and then the dynamic force acting on the system can be found out. Total vertical force acting on the mount is the sum of the values of vertical unbalanced force and gas force, which is presented in Eq.(7) and Eq.(12). To be safe for practical application, this value should be multiplied by safety factor which is range from 1.1 to 1.2.

3. Design and Optimization

3.1 Design of MR Mount

In this study, MR mount is designed based on the current rubber mount modified by adding MR piston part. This design is used to control dynamic vertical force which appears in operation of engine. A lot of configuration of the MR mount was designed to choose the best model. We chose a mount model using two coils and one is in longitudinal body of piston and the other is in bottom of housing. Parameters of two coils are equal in calculating mount. Other parameters of this MR mount are computed by utilizing initial values in mathematical model of this mount. Calculated parameters in this research are shown in Table 1 and the model for calculating damping force of MR mount is shown in Fig.6 where L_p is the length of pole coil, L is height of piston, R_{c1} is radius of first coil, R_p is radius of piston, R_{c2} is radius of second coil, R_{tc} is total radius of second coil, d_{o1} is gap between inside vertical housing and piston and d_{o2} is bottom gap between piston and housing.

The MR fluid is MR132DG with medium yield stress

$$\tau_y = 52.962B^4 - 176.51B^3 + 158.79B^2 + 13.708B + 0.1442 \quad (13)$$

where τ_y is yield stress of MR132DG fluid in magnetic field (kPa), B is magnetic flux density (Tesla). To evaluate results, it should be noted that the saturation of steel is not over 1.8 Tesla, and MR132DG fluid is less than 1.65 Tesla(4).

Damping force F_d of this mount is summed by four forces, which is defined as(2)

$$F_d = F_{re} + F_{bfm} + F_{vfm} + F_{sm} + F_{sqm} \quad (14)$$

where F_{re} is force of rubber mount which stands static load, F_{bfm} is force of flow mode of MR fluid at bottom housing, F_{vfm} is force of flow mode of MR fluid at longitudinal piston, F_{sm} is force of shear mode of MR fluid at longitudinal piston, F_{sqm} is force of squeeze mode at bottom housing.

Noting that the Eq.(14), the force from the bottom includes two forces. One is squeeze force in the initial state of MR fluid which magnitude is set up in a small time with small vibration. Another is flow force, which appears when the system has large amplitude vibration as shown in Eq.(16). The forces are expressed as(2)

Table 1 Calculated parameters of mount

Parameter	Value
Initial mass	4400kg
Type of MR fluid	MR132DG
Length of magnetic pole (L_p)	10mm
Radius of piston (R_p)	90mm
Diameter of piston rod	27mm
Current (I)	2.5A
Number of turns (N)	104
Radius of first coil (R_{c1})	85mm

$$F_{sqm} = \frac{3\pi\mu R_p^4}{2d_{o2}^3}\dot{u} + \frac{4\pi\tau_y R_{c2}^3}{3d_{o2}} sign(\dot{u}) \quad (15)$$

$$F_{bfm} = \frac{3\pi\mu R_p^4}{2(d_{o2}+u)^3}\dot{u} + \frac{4\pi\tau_y R_{c2}^3}{3(d_{o2}+u)} sign(\dot{u}) \quad (16)$$

$$F_{sm} = \frac{2\pi R_p \mu L}{d_{o1}}\dot{u} + 2\pi R_p L_p \tau_y sign(\dot{u}) \quad (17)$$

$$F_{vfm} = \frac{12A_p \cdot \mu L}{2\pi R_p \cdot d_{o1}^3}\dot{u} + \frac{2A_p \cdot L_p \cdot \tau_y}{d_{o1}} sign(\dot{u}) \quad (18)$$

where μ is off-state plastic viscosity (Pa.s), $\dot{u}$ is velocity of pistion (m/s), u is displacement of piston (m).

3.2 Optimization of MR Mount

Based on the initial results of calculation MR mount, its result will be used to optimize the values to design practical mount. The program APDL of ANSYS software is used for optimization process. First, order of optimization module is used to optimize MR design. In this module, various steepest descent and conjugate direction searches are enforced for each iteration until convergence is reached. Flow chart of optimizing MR mount is shown in Fig.7.

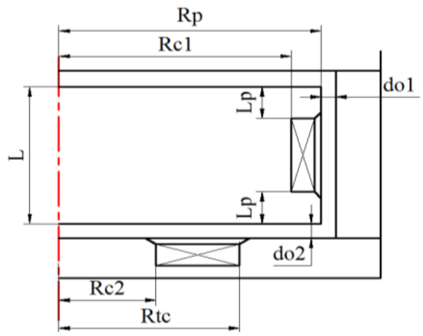

Fig.6 Geometry for calculating force of MR mount

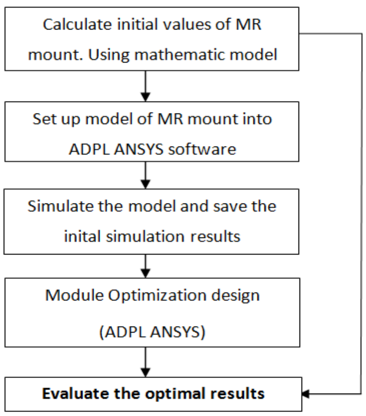

Fig.7 Flow chart of optimizing MR mount

In this research, to concentrate on the vertical forces from engine, it is important to determine distance between bottom of housing and piston. This area will create the larger force than other areas, and it is enough to maintain high force from MR fluid when current is supplied to the system. As shown in Fig.8, the total force is inversely proportional to the distance of piston-bottom housing. The best values of distance of piston-bottom housing is from 1mm to 2mm, but it is difficult to control its value. To optimize mount, data of this distance are chosen to optimize program and its values are from 1mm to 5 mm. It is noted that the larger distance piston-bottom of housing is, the smaller damping force in MR mount is.

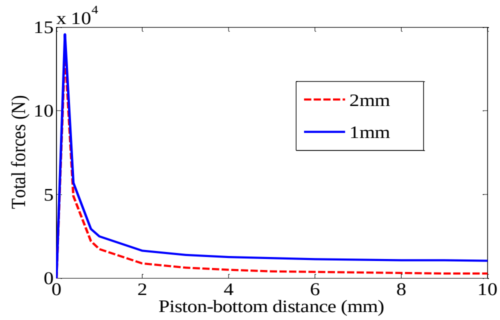

Fig.8 Relation between piston-bottom distance versus total forces

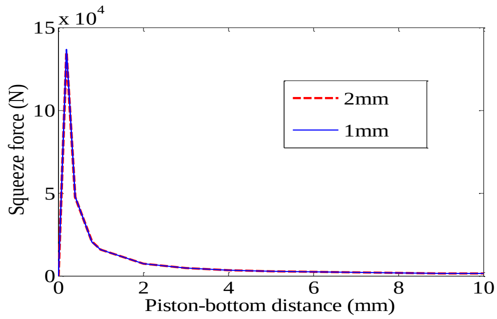

Fig.9 Relation between piston-bottom distance versus squeeze forces

Table 2 Optimized parameters

Parameter	Symbol	Min	Max	Result	Unit
Bottom gap between piston and housing	d_{o2}	1	5	2	mm
Length of second coil ($R_{tc}-R_{c2}$)	L_{c2}	20	33	20	mm
Radius of second coil	R_{c2}	47	65	60	mm
Total forces	TTF		26400	26703	N

In this MR design, the squeeze mode occurs on the bottom for a short time. Fig.9 shows relation between distance piston-bottom and squeeze mode which does not change. As mentioned, the squeeze mode does not remain during the operation of system, the next state is the flow mode as shown in Eq.(16), which is the main force for standing vertical high load and it will be used in optimization progress.

In progress to optimize mount, three main parameters which influence on progress are bottom gap between piston and housing d_{o2}, radius of second coil R_{c2}, total radius of second coil R_{tc}. Objective function in this research is related to the forces in mount which is 26400N. Function used to find optimal values and convergence can be defined by the following equation[4]

$$OBJ = \frac{1}{F_{total}} \tag{19}$$

where F_{total} is objective total forces of mount (N), $F_{total} = F_d$. In table 2, optimized force is found with 26703N which its tolerance is 1%.

4. Conclusion

In this work, a new type of MR mount for vibration control of diesel engine for ship was proposed and optimally designed. The excitation force of v-type diesel engine was analyzed and an appropriate size of MR mount which can produce actuating force over 25000N was designed. In addition, the actuating force has been maximized with the geometric constraints via optimization tool of ANSYS APDL. It is finally remarked that the optimally designed MR mount will be manufactured and its performance will be evaluated in the near future.

References

(1) Q H Nguyen, S B Choi, Y S Lee and M A Han , 2009, An analytical method for optimal design of MR valve structures, Smart Mater. Struct.18, 095032 (12pp).

(2) Mikel J.Brigley, 2006, A multi-mode magnetorheological axial isolator, Thesis of master of science, University of Maryland, USA.

(3) Frank M.White, Fluid mechanics, 2006, fourth edition, Mc-Graw Hill, New York.

(4) Q H Nguyen, S B Choi, 2010, Optimal design of an automotive magnetorheological brake considering geometric dimensions and zero-field friction heat, Smart Mater. Struct.19, 115024 (11pp).

(5) R S Khurmi, J P Gupta, 2008, Theory of machines, Eurasia publishing house.

(6) Huyndai heavy industries Co. ltd, 2012, Vibration of marine diesel engine.

(7) Guangqiang Yang, Billie F. Spencer Jr, Hyung-Jo Jung, J. David Carlson, 2009, Dynamic modeling of large-scale magnetorheological damper system for civil engineering applications. Journal of engineering mechanics, Vol.130, No.9,1107-1114.

(8) G. Yang, B.F. Spencer Jr., J.D. Carlson, M.K. Sain, 2002, Large-scale MR fluid dampers: modeling and dynamic performance considerations. Engineering Structures 24 309–323.

(9) Seung-Bok Choi, Sung-Ryong Hong, Kum-Gil Sung, Jung-Woo Sohn, 2008, Optimal control of structural vibrations using a mixed-mode magnetorheological fluid mount. International Journal of Mechanical Sciences 50 559–568.

(10) Brent J. Bass and Richard E. Christenson, 2007, System identification of a 200 kN magneto-rheological fluid damper for structural control in large-scale smart structures. Proceedings of the 2007 American Control Conference.

(11) S.R. Hong, S.B. Choi, D.Y. Lee, 2006, Comparison of vibration control performance between flow and squeeze mode ER mounts: Experimental work. Journal of Sound and Vibration 291 740–748.

(12) S.R. Hong, S.B. Choi, Y.T. Choi, N.M. Wereley, 2005, Non-dimensional analysis and design of a magnetorheological damper. Journal of Sound and Vibration 288 847–863.

(13) S.R. Hong, N.M. Wereley, Y.T. Choi, S.B. Choi, 2008, Analytical and experimental validation of a nondimensional Bingham model for mixed-mode magnetorheological dampers. Journal of Sound and Vibration 312 399–417.

(14) Billie F. Spencer, Jr., Guangqiang Yang, J. David Carlson, Michael K. Sain, 1998, Smart dampers for seismic protection of structures:a full-scale study. The Second World Conference on Structural Control, Kyoto, Japan.

(15) Alireza Farjoud, Michael Craft, William Burke And Mehdi Ahmadian, 2011, Experimental investigation of MR

squeeze Mounts. Journal of Intelligent Material Systems and Structures 22: 1645-1651.

(16) Xin-jie Zhang, Alireza Farjoud, Mehdi Ahmadian, Kong-hui Guo, Michael Craft, 2011, Dynamic testing and modeling of an MR squeeze mount. Journal of Intelligent Material Systems and Structures.

(17) Janusz Gołdasz, Bogdan Sapiński, 2011, Modeling of magnetorheological mounts in various operation modes. Acta mechanica et automatica, vol.5 no.4.

(18) K M Popp, X Z Zhang W H Li and P B Kosasih, 2009, MRE properties under shear and squeeze modes and applications. Journal of Physics: Conference Series 149 012095.

(19) P Kuzhir, M T L´opez-L´opez, G Vertelov, Ch Pradille and G Bossis. Oscillatory squeeze flow of suspensions of magnetic polymerized chains. J. Phys.: Condens. Matter 20 (2008) 204132 (5pp).