Fluid inertia damper using MR fluid with a long spiral bypass pipe

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Abstract

An inertia damper with a long spiral bypass pipe was developed in order to obtain a series inertia effects for vibration suppression of structures in practical use. The damper comprised a cylinder, piston, and MR fluid. The gap was spirally formed around the outside of a cylinder, and acted as a long bypass pipe. MR fluid is well known for its changeable damping effect, but the mass of the fluid is the focus of this study. It is obvious from previous studies that a fluid inertia effect is caused by quick motion in the long bypass pipe, and it depends on the length of the bypass pipe, compression ratio of area, and density. A large inertia effect can be derived if the fluid has a heavy mass. In order to confirm the inertia effect, a prototype damper was manufactured, and resisting force characteristics were measured. Seismic response tests using a 3-story structure were then performed using a shaking table with the damper installed. Each story of the structure had a height of 3 m, and a weight of 6 tons with a total height of 9 m and a total weight of 18 tons. Several types of earthquakes were input, and the response acceleration, relative displacement, and reaction force were measured. Finally the effects of vibration reduction were confirmed experimentally, and a feasibility study was discussed.

Key words : Vibration control device, Vibration control, Damping, Damper, MR fluid, Series inertia mass

1. Introduction

Many vibration control devices have been developed and used in practice for the purpose of vibration energy dissipation. In such devices, fluid-type dampers using oil and air, known as oil damper and air damper, are very useful for obtaining a damping effect and are comparatively easy to use; therefore they are used in many fields. As a type of fluid, Magneto-Rheological fluid (hereafter, designated as MR fluid) is also used in dampers instead of oil. The fluid-type damper using MR fluid (known as MR damper) can have the advantage of controllable damping, which can be obtained by virtually changing the viscosity of the fluid because the yielding shear stress of the fluid can be switched by means of a magnetic field (Dyke, et al., 1996, Spencer, et al., 1997, Yang, et al., 2002). MR fluid is a magnetic colloidal solution that comprises oil and small iron particles with surfactants and it has a heavier density than other fluids. Meanwhile, the authors have developed some dampers with an inertia effect to provide a damping effect. An electromagnetic-type damper using a ball screw mechanism can have an inertia effect using a flywheel (Matsuoka, et al., 2007, Sunakoda, et al., 2009, Hiramoto, et al., 2012, Nakamura, et al., 2014), and a vibration suppression device using a fluid can utilize the fluid's inertia (Sunakoda and Matsuoka, 2006, Matsuoka and Sunakoda, 2009). The inertia effect of the damper is to act as a series inertia mass to a primary mass when the damper is installed in a normal 1DOF

system, and it can make a natural frequency shift towards low frequency and cut off vibration at the anti-resonance frequency. This concept was previously patented by Okumura (2005). Dampers with an inertia effect have been developed by many researchers, e.g., a mechanical snubber (Chiba and Kobayashi, 1984, Kawaguchi, et al., 1991), a pump damper (Kawamata and Onuma, 1986), and a rotary damping tube (Furuhashi and Ishimaru, 2004, Saito, et al., 2007). Furthermore, there are a few studies on the variable inertia effects (Ullman and Velkoff, 1979), a mechanical-type damper using a flywheel (Ohmata and Miyanaga, 1991), an electrical-type damper using impedance matching (Kato and Ito, 1990) or LCR circuits (Isoda, 2008), and a fluid-type damper (Matsuoka, 2011). In this study, the authors developed a fluid inertia damper of 20 kN capacity by utilizing the heavy density of the MR fluid in order to practically apply an inertia effect to the damper. The inertia effect is caused by the quick movement of fluid in a long bypass pipe. The technique of using an inertia effect in a long spiral bypass pipe was patented by Teramura and Yamaguchi (1999), but it has not yet been in the practical application phase and was not accomplished by using MR fluid. In this paper, a proto-type of the fluid inertia damper is designed and manufactured. The theory of inertia and damping effects in cases having a spiral bypass pipe is introduced. To evaluate the dynamic properties of the damper, resisting force characteristics were measured and compared with the theoretical results. Subsequently, to confirm the effects of the vibration reduction for structures, seismic vibration tests were performed using a shaking table at National Center for Research on Earthquake Engineering (NCREE) with the damper installed on the first floor of 3-story benchmark structure. Each story of the structure had a height of 3 m and a weight of 6 tons, with a total height of 9 m and a weight of 18 tons. The effects are estimated both experimentally and numerically, and finally a feasibility of the damper is discussed.

2. Specification

The design diagram of a fluid inertia damper is shown in Fig. 1, and its design parameters are listed in Table 1. It is expected that the damper has a maximum load capacity of 20 kN. The damper comprises a cylinder, piston, cylinder blocks, flange, and rod end. The cylinder, piston and cylinder blocks are made of SS400 structural steel. The cylinder comprises two parts: an inner cylinder and an outer cylinder. A rectangular gap that is 11 mm deep and 11 mm wide is spirally formed on the outside of the inner cylinder with 11 turns and a 20 mm pitch. The inner cylinder is inserted in to the outer cylinder with an interference fit. The spiral gap acts as a long bypass pipe, such as an orifice. The fluid, LORD Co. MRF-122EG, which is a density $2,280 \text{ kg/m}^3$, is poured into the cylinder and bypass pipe through tapered screw holes. The volume of fluid used is about 1.8 liters. Rubber seals are placed between the piston, cylinder, and cylinder block for leak proofing, and a backup ring is also inserted into the piston head to prevent compression by the weight of the piston itself. The fluid inertia effect is caused by the fluid moving quickly through the long bypass pipe when the piston is pushing fluid out of the cylinder. The damper has 90 mm stroke and weighs about 40 kg including the fluid.

3. Theory

To introduce the resisting force characteristics of the damper, a conceptual sketch of the cylinder and gap is shown in Fig. 2. The damper has the characteristics of an inertia force, damping force, and friction force. When the

Fig. 1 Design diagram of the fluid inertia damper. The damper has load capacity of 20 kN. A rectangular gap is spirally formed on the outside of the inner cylinder, shown as dotted lines, which acts as a long bypass pipe.

displacement of the piston is defined as u , the total resisting force F is given by

$$F = F_m(\ddot{u}) + F_d(\dot{u}) + F_r \text{sign}(\dot{u}) \quad (1)$$

where F_m is defined as the inertia force, which is proportional to the acceleration; F_d is the damping force, which is proportional to the velocity; and F_r is the friction force of the damper, respectively. The last term of Eq. (1) shows the sign function, which gives 1 within positive velocity and -1 within negative velocity, otherwise 0.

3.1 Inertia effect

The inertia effect was theoretically introduced in previous papers (Sunakoda, et al., 2006, Matsuoka and Sunakoda, 2009). The kinetic energy of the fluid in the bypass pipe T is defined as

$$T = \frac{1}{2} m \dot{y}^2 \quad (2)$$

where m is the mass of the fluid in the bypass pipe and y is the displacement of the fluid. From the continuity equation, a fluid volume that is pushed out from the cylinder should be equal to the volume that is introduced into the bypass pipe, which is

$$A_o y = A_p u \quad (3)$$

where A_p and A_o are the cross-sectional areas of the piston and bypass pipe and become $\pi(D_1^2 - D_2^2)/4$ and bh , respectively. D_1 and D_2 are the diameters of the cylinder and piston rod, and b and h are the width and depth of the rectangular gap, respectively. The mass of the fluid in the bypass pipe is expressed by

$$m = \rho A_o L \quad (4)$$

where ρ is the density of the fluid, and L is the length of the bypass pipe. L is geometrically given by Pythagoras' theorem and becomes

Table 1 Design parameters of the fluid inertia damper.

Diameter of the cylinder	D_1	90 mm
Diameter of the piston rod	D_2	34 mm
Pitch center diameter of the gap	D_p	123 mm
Width of the gap	b	11 mm
Height of the gap	h	11 mm
Spiral pitch	L_p	20 mm
Number of spiral turn	N	11 turns
Total length of the bypass pipe	L	4.26 m
Volume of the fluid		$1.79 \times 10^{-3} \text{ m}^3$
Stroke		$\pm 90 \text{ mm}$
Total weight		40 kg
Density of the MR fluid	ρ	$2,280 \text{ kg/m}^3$
Viscosity of the MR fluid	μ	0.28 Pas

Fig. 2 Conceptual sketch of the cylinder and gap.

$$L = N\sqrt{(\pi D_p)^2 + L_p^2} \quad (5)$$

where N is the number of spiral turns, D_p and L_p are the pitch center diameter and pitch of the spiral gap, respectively. Substituting Eqs. (3), (4), and (5) into Eq. (2) and using Lagrange's equation of motion, the inertia force F_m is given by

$$F_m = \rho A_o L \alpha^2 \ddot{u} \quad (6)$$

and the series inertia mass m_d is defined as a modulus that is related to the acceleration $\ddot{u}$ and given by

$$m_d = \rho A_o L \alpha^2 \quad (7)$$

where α is the compression ratio of the cross-sectional areas between the piston and bypass pipe, which becomes A_p/A_o . Therefore, it is noted from Eq. (6) that the inertia effect is proportional to the compression ratio squared, the density of the fluid, and the length of the bypass pipe.

3.2 Damping effect

The damping effect of MR damper using a bypass pipe was given in a past study (Sodeyama, et al., 2004), but the theory was in the case of a straight bypass pipe. In this research, a damping force in the case of a spiral bypass pipe is introduced. The damping force F_d , which is caused by a pressure drop in laminar flow is generally given by Bernoulli's equation, but it is further arranged in the case of a bent pipe and defined as

$$F_d = \frac{1}{2} \zeta \rho A_p \dot{y}^2 \quad (8)$$

where ζ is the coefficient of head loss for a whole spiral pipe. When $R_e(d/2R)^2$ is smaller than 91, the coefficient of head loss for a single bent pipe of 90° , which denotes ζ_1 , is given by (Ito, 1959)

$$\zeta_1 = 0.00873 \beta \theta \lambda_c \frac{2R}{d} \quad (9)$$

$$\beta = 0.95 + 4.42 \left(\frac{R}{d} \right)^{-1.96} \quad (10)$$

where R_e is the Reynolds number, which is $\rho d \dot{y} / \mu$; R is the bend radius of the pipe, which becomes $D_p/2$; θ is the bend angle; μ is the viscosity of the fluid; and d is a hydraulic equivalent diameter. In general, the hydraulic diameter is defined as 4 times the cross-sectional area per wetted perimeter, which becomes $4bh/(2b + 2h)$. However, the hydraulic equivalent diameter in the case of rectangular duct is given by Huebscher's equation (1948) and defined as

$$d = 1.3 \frac{(bh)^{0.625}}{(b+h)^{0.25}} \quad (11)$$

It is noted from Eq. (11) that the hydraulic equivalent diameter increases by about 10% to the initial hydraulic diameter in this case. λ_c is the friction coefficient of a bent pipe and is defined as

$$\lambda_c = \frac{21.5K}{(1.56 + \log_{10} K)^{5.73}} \lambda \quad (12)$$

where K is the Dean number, which denotes the centrifugal effects of the eddy flow in a curved pipe. λ is the friction coefficient of a straight pipe under laminar flow and is defined as $64/R_e$. The relationship between the Reynolds number R_e and Dean number K is given by

$$K = R_e \sqrt{\frac{d}{2R}} \quad (13)$$

It is assumed that the long spiral bypass pipe is a series of continuous bends. According to a previous study on loss factors in a coil pipe (Shimizu, et al., 1982), it was found that a mutual interference coefficient of bends η becomes about 0.4 in the case of continuous bends, and is given by

$$\eta = \frac{\zeta}{n\zeta_1} \quad (14)$$

where n is a number of bends, which becomes $360N/90$ in this case. It is also known that the mean flow velocity in the case of a bent pipe decreases by about 60% to the velocity of a straight pipe (Ito, 1987).

In fact, the damping effect has an approximately quadratic function relationship to the velocity, and the damping coefficient c_d is derived by $F_d/\dot{u}$.

4. Dynamic properties

To evaluate the dynamic properties of the damper, the resisting force characteristics were measured. The damper was attached to a fixed wall and a servo actuator through a load cell, and the force and displacement were measured when the actuator was forcibly excited by a sinusoidal wave with a frequency from 0.25 to 1.5 Hz and amplitude of 20 to 50 mm. In the performed tests, the perpendicular load on the damper due to its own weight was canceled by hanging it using a wire rope and eye bolt. The viscosity of the MR fluid was measured before the test using a viscometer and was determined as 0.28 Pas.

The experimental results together with the theoretical results, which were calculated by Eq. (1), are shown in Fig. 3. It is obvious from this figure that the damper can have an inertia effect, shown as a left-rising straight line that increases with increasing frequency, and that the damping effect has an oval shape. The series inertia mass m_d becomes 2,385 kg from Eq. (7), and the damping coefficient c_d becomes approximately $(143|\dot{u}| + 34.8)$ kNs/m from Eq. (8), which is linearly proportional to the velocity. The friction force F_r was comparatively small and identified as approximately 100 N from the experimental results. The experimental results agreed well with the theoretical results, and the theory of inertia and the damping effect are confirmed. The L-shaped fixed wall, which comprised steel plates of 24 mm thick and ribs, elastically deformed slightly under the heavy load. Since a harmonic rotating motion occurred under high frequency due to the centrifugal force of the fluid flow, it seems that the experimental results were partially influenced as waveforms.

Fig. 3 Resisting force characteristics. —: Exp., - -: Cal. A left-rising slope, as a broken line, indicates the inertia effect that was derived from Eq. (7). The inertia and damping effects increase with increasing frequency. The experimental results agree with the theoretical results.

5. Seismic vibration tests

In order to confirm effects of vibration reduction of the damper for real structures, seismic vibration tests were performed on a 3-story benchmark structure. The 3-story structure had height of 3 m, width of 3 m, span of 2 m, weight of 6 tons per story, and was 9 m high. Its total weight was 18 tons, as shown in Fig. 4. The structure was placed on a shaking table, which was 5 m×5 m square. Added masses were applied to each story. The damper was installed between the base and first floor only, using a chevron brace. The series inertia mass of the damper was arranged to be about 10% of the total mass of the structure, and the actual mass ratio $m_d/(m_1+m_2+m_3)$ became 0.13. These tests were operated by NCREC in Taiwan. Four types of earthquake were applied horizontally by the shaking table: Imperial Valley earthquake in 1940 (USA); El Centro north-south component, Southern Hyogo prefecture earthquake in 1995 (Japan); Japan Meteorological Agency Kobe north-south component, and two kinds of Chi-Chi earthquake in 1999 (Taiwan); TCU-84 and TCU-129, in which the maximum acceleration was normalized to be from 1 to 6 m/s². The wave of TCU-84 had a low dominant frequency and comparatively long period component. Conversely, the wave of TCU-129 had a wide band of frequency component. The response acceleration, relative displacement of each story and reacting forces of the damper were measured by accelerometers, linear variable displacement transducers, and a load cell, respectively. The tests were stopped at the acceleration level where the relative displacements between each story exceeded 50 mm.

An analytical model of the 3-story structure is shown in Fig. 5, and its physical parameters are listed in Table 2. Stiffness k_i and structural damping coefficient c_i were identified from the response acceleration of each story without the damper when a random wave with white noise was input (Nakamura, et al., 2000). The equation of motion of the 3-story structure is given in a matrix form as

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = -\{m\}\ddot{x}_0 \quad (15)$$

where x_0 is an input displacement of the shaking table, and M , K , C are the mass, stiffness, and damping matrices, which are defined as

$$[M] = \begin{bmatrix} m_b + m_d & -m_d & 0 & 0 \\ -m_d & m_1 + m_d & 0 & 0 \\ 0 & m_2 & m_2 & 0 \\ 0 & m_3 & m_3 & m_3 \end{bmatrix}, \quad [K] = \begin{bmatrix} k_b & 0 & 0 & 0 \\ 0 & k_1 & -k_2 & 0 \\ 0 & 0 & k_2 & -k_3 \\ 0 & 0 & 0 & k_3 \end{bmatrix}, \quad [C] = \begin{bmatrix} c_d & -c_d & 0 & 0 \\ -c_d & c_1 + c_d & -c_2 & 0 \\ 0 & 0 & c_2 & -c_3 \\ 0 & 0 & 0 & c_3 \end{bmatrix}$$

$$\{u\} = \{u_b \quad u_1 \quad u_2 \quad u_3\}^T, \quad \{m\} = \{m_b \quad m_1 \quad m_2 \quad m_3\}^T \quad (16)$$

where x_i , m_i , k_i , and c_i are the absolute displacement, mass, stiffness and structural damping coefficient of the i -th story, respectively. The relative displacement of the i -th story is defined as $u_i = x_i - x_{i-1}$, and $u_b = x_b - x_0$. It is assumed that the stiffness k_b is defined as the equivalent stiffness of a fixed wall in order to consider elastic deformation and is determined from the experimental results by trial and error. Moreover, x_b is the absolute displacement of an additional degree of freedom, and m_b is an auxiliary mass which becomes zero in the calculations. The analytical results were derived by Matlab using input acceleration data, which was recorded for every test. The natural frequency in the 1st mode was calculated by modal analysis and became 1.0 Hz. The modal damping ratio of the 1st natural frequency was 0.4% without the damper and 8% with the damper as a maximum. The surface temperature of the outer cylinder was measured after every test using an infrared radiation thermometer and was maintained at a uniform ambient temperature of 22°C.

The experimental results of the time history waves of seismic responses together with the analytical results at levels that are comparable with cases in which dampers installed and not installed are shown in Figs. 6, 7, 8 and 9. Peak accelerations and relative displacements, which are related to the input acceleration, are plotted in Figs. 10 and 11, and are shown in Table 3. The peak acceleration of the damper installed in the case of El Centro and TCU-129 decreased about 3/5 - 1/2 to that of no damper installed, and the relative displacement decreased by about 1/2. Conversely, the peak acceleration decreased by about 1/2, whereas the relative displacements were damped down by 2/3 - 2/5 in the cases of Kobe and TCU-84. The damper seems to be more vibration suppressive for seismic motions having a long period component like Kobe and TCU-84 than the others, which include a high dominant frequency such as El Centro

Fig. 4 Seismic vibration test setup. 3-story structure with damper set on a shaking table.

Fig. 5 Analytical model of 3-story structure.

Table 2 Physical parameters of 3-story structure.

Height		9 m
Width		3 m
Depth		2 m
Total weight		18,000 kg
Weight per story	m_i	6,000 kg
1st natural frequency		1.0 Hz
Stiffness	k_1	1.257×10^6 N/m
	k_2	1.188×10^6 N/m
	k_3	1.069×10^6 N/m
	k_b	3.5×10^6 N/m
Damping coefficient	c_1	3.386×10^3 Ns/m
	c_2	3.920×10^3 Ns/m
	c_3	4.417×10^3 Ns/m

and TCU-129. It is believed that the natural frequency of the 1st mode of the structure, which shifted to a lower frequency due to the inertia effect of the damper, is close to the dominant frequency of the input waves of Kobe and TCU-84. It worked enough to suppress vibration in the cases of Kobe and TCU-84, since the inertia effect provided a damping effect. Vibration reduction was more efficient than that in the other cases. The damper worked at a maximum force of about 20 kN, and the inertia effect was produced, as shown by the left-rising straight line in Figs. 6(d) - 9(d). The analytical results for the case where the damper is installed are generally close to the experimental results. It is believed that the structural damping is smaller than the real one in the calculations. From Fig. 10, the peak accelerations with the damper installed decrease about 1/2 - 2/3 to that of no damper at every level. Moreover, from Fig. 11, the peak relative displacements decrease about 1/2 to the ones. It was believed that the peak values were increasing linearly to the input level, and all of the tests were performed within the elastic range of the structure. Meanwhile, seismic responses in the case with the damper installed without an inertia effect, i.e., a conventional fluid-type damper, were further simulated in order to evaluate the inertia effect. The peak response values are also shown in Table 3 within brackets. It seems that the maximum displacement grew 10% - 20% in the cases of El Centro and TCU-129, which have high frequency domains. Conversely, there is not a great difference between an inertia effect and no inertia effect in the cases of Kobe and TCU-84, which have long period component, since the inertia effect is comparatively useful in a high frequency range. Specifically, the responses at the 3rd story are larger than those in the case with the inertia effect. The maximum force became 1.4 times greater due to the inertia effect. Therefore, it is pointed out that the inertia effect of the damper supports the damping effect.

6. Conclusion

In this paper, a fluid inertia damper with a long spiral bypass pipe was developed and the dynamic properties and vibration reduction for structures were investigated. The primary conclusions can be summarized as follows:

1. A gap that acts as an orifice was spirally formed on the outside of the inner cylinder, and the inner cylinder was inserted into the outer cylinder with an interference fit. By this method, it is possible to make a long and narrow bypass pipe.
2. From the dynamic property tests, the fluid inertia damper can have an inertia force due to the mass of the fluid and a damping force due to the viscosity of the fluid. The friction force is negligibly small.
3. The inertia effect depends on the compression ratio between the cross-sectional areas of the piston and bypass pipe, density of the fluid, and length of the bypass pipe. The inertia force can be derived by using a heavier density of MR fluid.
4. The damping effect in the case of a spiraled bypass pipe is theoretically introduced and becomes a quadratic function related to the velocity.
5. From the seismic vibration tests, the peak accelerations of the damper installed at each story decrease about $3/5 - 1/2$ to that of no damper installed, and the peak relative displacements also decrease about $1/2 - 2/5$ on average. It is believed that vibration reduction for a seismic motion having a long period component is more effective than that of a wide-band frequency due to the inertia effect.
6. In the dynamic property tests and seismic vibration tests, the experimental results are well matched to the theoretical and analytical results and the validity of the theory and calculations are confirmed.

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Fig. 6 Seismic response waves in the case of El Centro 3 m/s^2 . —: Damper not installed, - -: Damper installed. Peak accelerations decrease about $3/5 - 1/2$, and displacements decrease about $1/2$. Maximum load is 22 kN .

Fig. 7 Seismic response waves in the case of JMA Kobe 2 m/s^2 . —: Damper not installed, - -: Damper installed. Peak accelerations decrease about $1/2$, and displacements decrease about $2/3 - 3/4$. Maximum load is 15 kN .

Fig. 8 Seismic response waves in the case of TCU-84 1.5 m/s^2 . —: Damper not installed, —: Damper installed. Peak accelerations decrease about 1/2, and displacements decrease about 2/5. Maximum load is 13 kN.

Fig. 9 Seismic response waves in the case of TCU-129 6 m/s^2 . —: Damper uninstalled, —: Damper installed. Peak accelerations decrease about 2/3, and displacements decrease about 1/2. Maximum load is 21 kN.

Fig. 10 Peak acceleration of seismic response. $\circ$: $\ddot{x}_1$, Δ : $\ddot{x}_2$, $\square$: $\ddot{x}_3$, black is with the damper not installed, red is with the damper installed. Response accelerations increase linearly with the input acceleration.

Fig. 11 Peak relative displacement of seismic response. $\circ$: u_1 , Δ : u_2 , $\square$: u_3 , black is with the damper not installed, red is with the damper installed. A linear increase means a region of elastic deformation.

Table 3 Comparison of peak values between the experimental and calculated results. Values in brackets are in cases without the inertia effect of the damper.

Input wave	Condition	Story	Acceleration [m/s ²]		Displacement [mm]		Force [kN]	
			Exp.	Cal.	Exp.	Cal.	Exp.	Cal.
El Centro 3 m/s ²	Damper not installed	1st	4.90	3.24	50.0	48.9		
		2nd	5.62	3.85	46.2	38.6		
		3rd	8.15	5.73	39.7	32.2		
	Damper installed	1st	3.15	3.50 (2.65)	28.9	26.6 (27.6)	22.3	19.9 (14.1)
		2nd	3.80	2.72 (3.16)	26.9	23.7 (26.3)		
		3rd	4.16	3.31 (4.01)	20.8	18.6 (22.5)		
Kobe 2 m/s ²	Damper not installed	1st	5.62	3.50	41.9	45.4		
		2nd	4.42	3.97	43.5	39.6		
		3rd	6.34	5.74	31.4	32.2		
	Damper installed	1st	2.49	1.99 (1.99)	32.3	31.1 (31.5)	15.4	13.4 (13.6)
		2nd	3.51	2.28 (2.62)	33.0	30.6 (33.1)		
		3rd	4.06	3.80 (3.95)	20.3	21.3 (22.2)		
TCU-84 1.5 m/s ²	Damper not installed	1st	2.48	2.07	50.8	44.2		
		2nd	4.33	3.40	46.3	38.4		
		3rd	5.16	4.34	27.1	24.4		
	Damper installed	1st	1.94	1.19 (1.23)	27.7	24.7 (27.0)	12.5	10.2 (9.9)
		2nd	2.58	1.89 (2.24)	26.4	21.5 (25.4)		
		3rd	2.90	2.63 (3.10)	16.2	14.8 (17.4)		
TCU-129 6 m/s ²	Damper not installed	1st	5.97	3.72	40.9	31.6		
		2nd	8.08	3.66	36.1	31.1		
		3rd	6.54	4.13	30.2	23.2		
	Damper installed	1st	5.42	3.81 (3.78)	23.7	22.7 (22.8)	20.6	14.7 (10.8)
		2nd	4.85	2.54 (3.08)	22.9	18.2 (20.1)		
		3rd	4.16	2.93 (3.57)	18.2	16.4 (20.0)		

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