THE USE OF ISOLATORS FOR ATTENUATING DYNAMIC DISTURBANCES GENERATED BY SMALL INTERNAL COMBUSTION ENGINES

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Abstract

This paper presents the factors that must be considered in selecting mounting systems for internal combustion engine installations that will be effective in providing vibration isolation and noise reduction. This includes a mounting arrangement that has proved very effective when the power take-off is by belt across the isolation system. The factors presented include the dynamic modes that are most disturbing, isolator location for the most efficient isolation system, and support structure requirements. Several case histories are presented to illustrate the practical application of the factors presented.

Preliminary Considerations

Good engine mounting can add much to the market value of a finished product. It can eliminate or reduce noise, shock, and vibration, increase operator comfort and safety, extend the fatigue life of parts, improve overall performance and assure compliance with safety and environmental regulations. But, for some manufacturers, engine mountings often have to be unduly compromised. How, then, can design compromises be avoided or minimized?

Manufacturers of internal combustion engines supply those engines with mounting points selected for ease of installation but those attachment points are in the wrong place for effective isolation. Therefore failure to begin design of the engine mounting early enough can be costly and result in redesigning the final product, more expensive, customized isolators and lost product value through compromising on a lower standard of engine isolation. Common problems that result are: mechanical shorts, failure to make sure that everything that attaches to the engine is flexible; insufficient clearance space allowed for the most desirable positioning of the isolators; brackets that are too flexible; a belt take-off from the engine going over an isolator; and modifications to the engine and its accessories after acceptance of the engine mounting design.

Nature of Disturbances

An engine fastened directly to its support frame has a direct path for the transmission of vibration and noise. When the engine is attached to its support by means of a properly selected resilient isolator, the path of vibration and noise disturbance is broken.

Two types of disturbances originate within the engine. These have to be identified in order to position isolators correctly and to select the isolators with the characteristics required for isolation. The first is generated by the firing pulse. The explosion of the fuel in the cylinder is resolved by the crank mechanism into a torsional moment acting on the stationary parts of the engine (see Figure 1a) about a longitudinal axis essentially parallel to the crankshaft centerline. This is usually referred to as the firing frequency and is the most disturbing mode. Therefore, the isolation of the firing frequency is the primary function of the mounting system.

The firing frequency can be calculated as follows: with a two-cycle engine: fd ~~(firing frequency) = RPM x no. of cyl.~~ (one power stroke per rpm). With a four-cycle engine: fd = RPM x no. of cyl. divided by 2 (one power stroke per two rpm). In practice, however, there is uneven piston firing because of uneven fuel mixture distribution. So you have to isolate the 1/2 order and 1st order disturbances as well as harmonics: 1-1/2, 2, 2-1/2, etc. (First order = 1 x RPM: second order = 2 x RPM, etc.)

The second group of disturbances are imbalance forces due to reciprocating (pistons) or rotating (crankshaft and rods) masses within the engine. These generate inertia forces and moments that react on the engine block about axes perpendicular to the crankshaft. Depending on the number of cylinders and the crankshaft construction, these forces combine and cancel each other in varying degrees. Those forces will vary from the case of a single cylinder engine, which is unbalanced in all directions in a lateral plane, to the case of a six cylinder engine which can have all of its primary and secondary forces and moments balanced internally. However, no engine less than six cylinders can be totally internally balanced and therefore, the disturbing inertia forces and moments must be considered, as well as the firing frequency, when designing an effective isolation system. Figure 1b presents the modes of primary and secondary forces and moments that have to be isolated for various engine types.

An understanding of the effect of dynamic loads on the engine is important to the proper placement of the isolators. When the engine vibrates freely under the influence of dynamic loads, it takes the course of least resistance, and therefore it wants to move along or rotate about the principal axis of minimum inertia (see figure 2). A greater amount of energy is needed to cause acceleration about or along other than the principal inertia axis. This is very important because effective engine isolation requires the placement of mounts with reference to the principal axis of minimum inertia. The mounts should be positioned to allow as much free vibration about the principal roll axis of inertia as possible (the Y axis in figure 2). Since the firing frequency (the most disturbing dynamic force produced by the engine) causes the engine to vibrate primarily about this axis, the isolation system should be very soft in the roll mode (and also because it is the axis with the minimum inertia value to react those dynamic forces).

Theory of Isolation

Any body free to move in space, has six degrees of freedom, three in translation (see Figure 3) and three in rotation (pitch, roll, and yaw). The freedom of movement in each of those modes is determined by the flexibility of the isolation system.

The ideal system would suspend the body freely in space on infinitely soft springs so that any periodic forces would be absorbed by the inertia reaction of the mass and virtually none would be transmitted through the isolators to the structure. This is obviously impractical, but it is practical to achieve isolation percentages of 80 or 90% and still maintain a relatively stable isolation system.

The ideal mounting system should provide isolation in all six natural modes throughout the operating range of the engine. In order to accomplish this, the natural frequency of the mounting system in a particular mode must be less that the $\sqrt{2}$ times the lowest disturbing frequency in that mode (see figure 4). Otherwise the mounting system will actually be amplifying the dynamic forces. A good rule of thumb is that the natural frequency should be half of the disturbing frequency in question. This will assure that at least 60% isolation is achieved.

Isolation is not solely a function of the stiffness of the isolators. A problem in the application of isolators results from a lack of understanding of the way isolators function. The misconception exists that somehow the isolators soak up the vibration like a sponge. As noted earlier, an infinity soft system allows the power plant to float in space and thus the dynamic disturbances are dissipated by doing work in overcoming the inertial reaction of the mass. In other words, the isolators allow the engine to move freely relative to the support structure. The notion that an isolated engine should not exhibit such relative motion has prompted many designers to spread the isolators as far as possible from the engine; i.e., provide a wide footprint for the mounting system in the mistaken idea that this would improve stability and reduce relative motion. This is one of the major causes of an unsuccessful isolation system.

For example, from Figure 1b, we note that for two and three cylinder engines, besides the firing frequency (roll), the major dynamic disturbances are in pitch and yaw (see Figure 2). In other words, all the rotational modes must be isolated. In isolating rotational modes the placement of the mounts is critical (see Figure 5). The rotational stiffness increases as the square of the distance between isolators. Spreading the isolators to achieve a wide footprint raises the rotational frequencies into the operating range where the dynamic disturbances are amplified rather than isolated. So the purpose of the system is defeated.

Interaction With The Support Structure and Noise Reduction

Regardless of how carefully the stiffness characteristics and isolator locations are selected, the whole exercise can be for naught if the interaction with the support structure is not considered.

A general rule for effective isolation is that the stiffness of the engine bracket and support structure should be at least 10 times greater than the isolator. This will guarantee that 90% or more of the total system stiffness is provided by the isolator and 10% or less from the support and attachment structures. A support structure that is too flexible will nullify the effectiveness of the isolator.

Today there is as much concern with noise reduction as there is vibration control. Structure borne disturbances can be grouped into two bands:

Mechanical Vibration - 0-50Hz
Noise - 50-20k Hz

Thus, noise is merely high frequency vibration and elastomeric isolators, which provide excellent low frequency isolation, should provide excellent isolation for noise as well. Under ideal conditions such isolators achieve 9 - 12 dB per octave attenuation. However, in many cases, while the mechanical vibration problem is solved by the use of elastomeric isolators, the noise problem is not diminished.

The reasons for this are:
1) Airborne and structure-born noise exist side-by-side and, since the isolators can reduce only the structure-born noise, the air-born noise, which probably predominates, is not affected.
2) Structure-born noise attenuation does not depend solely on the isolator performance characteristics but also on the rigidity in the support structure. Thus, the most common cause for poor system performance in noise reduction is structural flexibility.

<u>Belt Power Take-Off</u>

When the power takeoff is by belt, serious isolation problems develop if the belt is routed across the mounting system particularly when the conventional engine mounting points are used. With this arrangement, mounts flexible enough to provide isolation will allow engine deflections too great to maintain pulley alignment or belt tension. Isolators stiff enough to maintain pulley alignment and belt tension will not provide the desired degree of isolation.

This problem can be overcome, to a great degree, by a more judicious location of the mounting points and the selection of isolators with specific stiffness characteristics. The basic problem is that the belt pull force is amplified in the isolators when the conventional mounting feet are used because the feet are located away from the point the force is applied. Thus, stiffer isolators are needed to minimize engine deflections. By locating the isolators as close to the plane of the belt as possible, the multiplying effect of the belt force is reduced and engine deflections can be controlled. Also selecting isolators with the proper stiffness characteristics will achieve a high degree of isolation.

This mounting arrangement is shown in Figure 6 for both a vertical and horizontal belt power take-off. In both cases, a three point mounting system is used. For a horizontal belt, the front isolators are located as close to the plane of the engine pulley as possible and are elevated to the crankshaft centerline. The isolators are stiff in the lateral direction and so can absorb the belt loads with little deflection. The isolators are very flexible in the vertical direction to provide as much isolation of the firing frequency as possible. At the engine rear, there is a single isolator to act as a stabilizer. This isolator is very flexible in the lateral direction.

The same principals apply when the belt pull is vertical except that in this case the front isolators are stiff in the vertical direction and very flexible in the lateral plane. To provide isolation for the firing frequency with this arrangement, the lateral spread of the front isolators should be kept to a minimum. The rear isolator has the same characteristics and location as noted above.

These mounting systems have been successful in providing a high degree of isolation even when the belt force was 2 to 3 times greater than the mass of the engine. Case History number 3 provides data on one such installation.

Some Case Histories

The following case histories Illustrate the actual application of the principles herein stated:

Case History Number 1

A midwestern company manufactures a skid steer loader powered by a Deutz Model F2L511 two cylinder, four cycle, air cooled diesel engine. The engine drives a separately mounted hydraulic pump through a short drive shaft.

The engine is supplied with mounting brackets as shown in Figure 7 which, when used, constitutes a floor mounted system. The company used the mounting feet provided and selected standard commercially available elastomeric isolators solely on the basis of their rated static load, a common practice in the industry.

Two models of the skid steer loader went into production, but after many customer complaints of excessive noise and vibration, and several cases of equipment failure, the company decided to review their isolation system.

Analysis of the system clearly shows why a problem existed. In a two cylinder engine all the rotation modes must be isolated; i.e., the firing frequency (roll) as well as pitch and yaw which was not the case with the initial mounting arrangement. All the rotational modes were in the operating speed range of the engine - in the range of 2300 to 2700 R.P.M. Therefore, instead of isolating the firing and unbalance disturbances produced by the engine, the isolation system was actually amplifying them.

Since the vehicles were already in production, the options open to the company to solve this problem were extremely limited. Nevertheless, an isolation system with reasonable performance was achieved.

The isolators were repositioned in the horizontal plane of the crank shaft center line by the use of a cradle arrangement as shown in Figure 7. New elastomeric isolators were chosen on the basis of load-deflection characteristics as well as load ratings so that all the rotational modes were reduced by more than 50% to below 1200 R.P.M.

Thus, while no isolation was achieved for idle R.P.M., at least sufficient isolation and noise reduction was provided at operating speeds {1800 - 3000 R.P.M.) to eliminate customer complaints and other problems.

<u>Case History Number 2</u>

The manufacture of turf machinery was developing a new skid steer loader powered by a three cylinder, four cycle Teledyne Continental Model TMD 20 diesel engine driving a separately mounted hydraulic pump.

The company selected a four point mounting system for convenience of installation and selected isolators that they had previously used successfully on a four cylinder diesel engine. However, Figure 1b shows that isolation of a four cylinder engine is easier than a three cylinder. While a mounting arrangement may perform very well on a four cylinder installation, it probably will not be satisfactory for a two, three, or five cylinder engine installation. For the three cylinder engine, all the rotational modes (roll, pitch, and yaw) have to be isolated. The system initially selected by the company failed to do this. The rotational natural frequencies were close enough to the operating speeds of the engine to render the system only marginally effective. Space limitation precluded any change in the location of the rear isolator. However, isolators with more appropriate load-deflection characteristics, i.e., more flexible in the horizontal plane than the initial units used, were selected.

The front isolators were changed and relocated. They were moved backward to the mounting bracket provided on the engine, upward to the plane of the engine center line, and focused (for information on focusing isolators, see SAE paper no. 821095).

The system (Figure 8) achieved a significant improvement in vibration isolation. All natural modes fell below 800 R.P.M. which permitted the company to limit the dynamic disturbances even at idle by using a high idle (1200 R.P.M.).

Case History Number 3

A manufacturer of small utility vehicles used in turf, agriculture, golf course maintenance, etc. redesigned one of their vehicles which included changing the power plant from an 8 h.p., 4 cycle unit to a 14 h.p., Vanguard V-Twin 2 cylinder, 4 cycle engine. The engine drives a separately mounted fully automatic torque converter through a belt across the mounting system. The belt pull is over 250 lbs. while the engine weight is only 110 lbs.

The original mounting arrangement used the engine feet with stiff isolators to maintain belt tension and pulley alignment. This system was not effective in attenuating engine generated noise and vibration and the company wanted improved performance in the redesigned vehicle.

The mounting arrangement shown in Figure 5 for the horizontal belt pull was recommended. The front isolators selected were 2.5 times stiffer in the lateral direction than the vertical to absorb the belt pull with a minimum of deflection. The rear stabilizer mount is a simple stud type isolator, 5 times stiffer vertically than laterally.

This arrangement required the construction of a cradle under the engine to extend the mounting points to their most effective position. However, the performance achieved justified the addition of the cradle which actually simplified the engine mounting arrangement. The project engineer responsible for the vehicle redesign stated, "The isolation system meets every expectation we were hoping for. The vibration isolation has improved dramatically and has reduced significantly the noise associated with rattling sheet metal on the vehicle. Going from the old version to the new is like stepping out a jalopy and into a Cadillac".

Subsequent testing of the new and the old vehicles proved the truthfulness of the engineers statement. The isolation efficiency of the recommended system varied from 84% to 91% depending on operating mode while that of the old design varied from 35% to 66% under the same conditions. Noise at the operators ear was reduced from a range of 80 to 86 db on the old vehicle design to a range of 74 to 83 db on the new. While the difference between the two vehicles encompassed more than the isolation system, the majority of the reduction of noise and vibration in the redesigned vehicle can be attributed to the unique mounting arrangement.

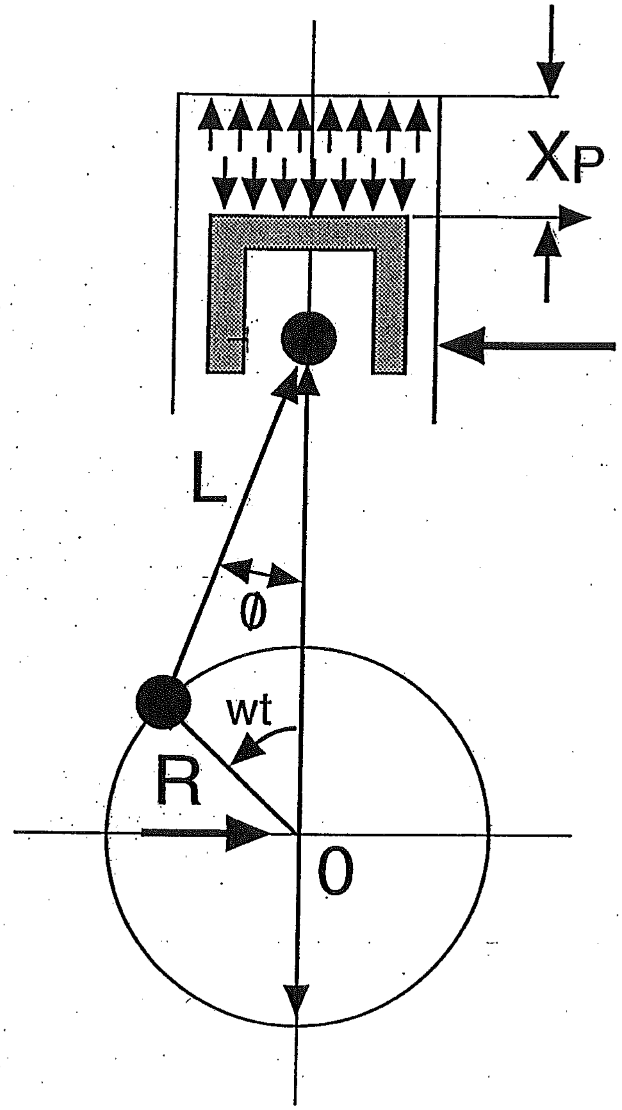
Case History Number 4

The manufacturer of a range of grass cutting machinery used by highway departments and municipal golf courses designed a new vehicle to meet high requirements of durability and longevity of service. It had to comply with the various noise regulations of the E.E.C. which required both the engine and the operator platform to be isolated. The vehicle is powered by a 51 h.p., Kubota, 4 cylinder diesel engine. To make the most economical use of the isolators and to reduce inventory costs, the customer required that the same mount be used at all attachment points on both the engine and the cab.

The most efficient mounting system is achieved when the stiffness of the isolator at each attachment point is proportional to the static load at that point, i.e., if the static load at one mounting point is twice that of anther point, then the stiffness of the isolator at the first point would be twice the stiffness of the isolator at the second, necessitating different mounts at each point. The requirement of the customer to have an efficient system and at the same time using the same isolator at all mounting places was achieved using the three point system shown in Figure 9.

Two focused, captive type isolators were used at the front of the power pant and a single unit of the same mount was used under the rear of the hydraulic pump. Captive mounts (with interlocking metal components that prevent separation in the even of rubber failure) were chosen because of the rugged operating duty cycle of the vehicle but also to meet the ROP requirement for the operator cab.

The success of the mounting arrangement is shown by the fact the machine meets the E.E.C. noise level directives 84/538 and 88/187.



Crank mechanism

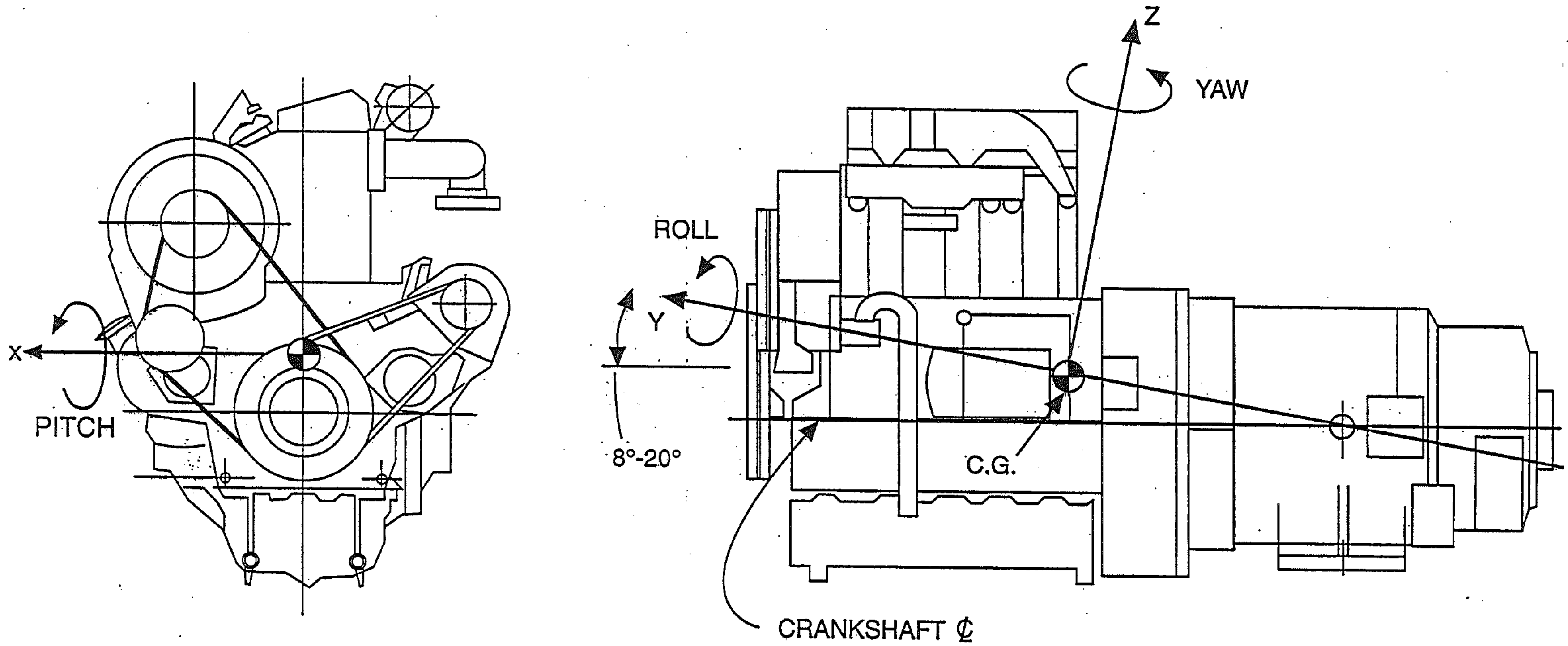
a.) Torsional moment created by crank mechanism (firing frequency)

TYPE ON ENGINE	CRANK ARRANGEMENT	INERTIA FORCES		INERTIA MOMENTS	
		PRIMARY RPM	SECONDARY 2xRPM	PRIMARY RPM	SECONDARY 2xRPM
ONE CYLINDER	1	VERTICAL HORIZONTAL	VERTICAL	NONE	NONE
TWO CYLINDER VERTICAL	1-2		VERTICAL	PITCH YAW	
TWO CYLINDER OPPOSED	1 2			YAW	YAW
2 CYLINDER HORIZONTAL	1 2	VERTICAL			
2 CYLINDER 90°V	1-2		HORIZONTAL		
2 CYLINDER 60°V	1-2	VERTICAL			
THREE CYLINDER VERTICAL	1-2-3			PITCH YAW	YAW
FOUR CYLINDER VERTICAL	1-2-3-4		VERTICAL		
FOUR CYLINDER VERTICAL	1-2-3-4			PITCH YAW	
4 CYLINDER HORIZONTAL	1, 2 / 3, 4	—		YAW	
4 CYLINDER HORIZONTAL	1, 4 / 2, 3				YAW
FOUR CYLINDER 60°V	1 4 2 3			YAW	YAW
SIX CYLINDER VERTICAL	1-2-3-4-5-6				

b.) Unbalanced inertia forces and moments

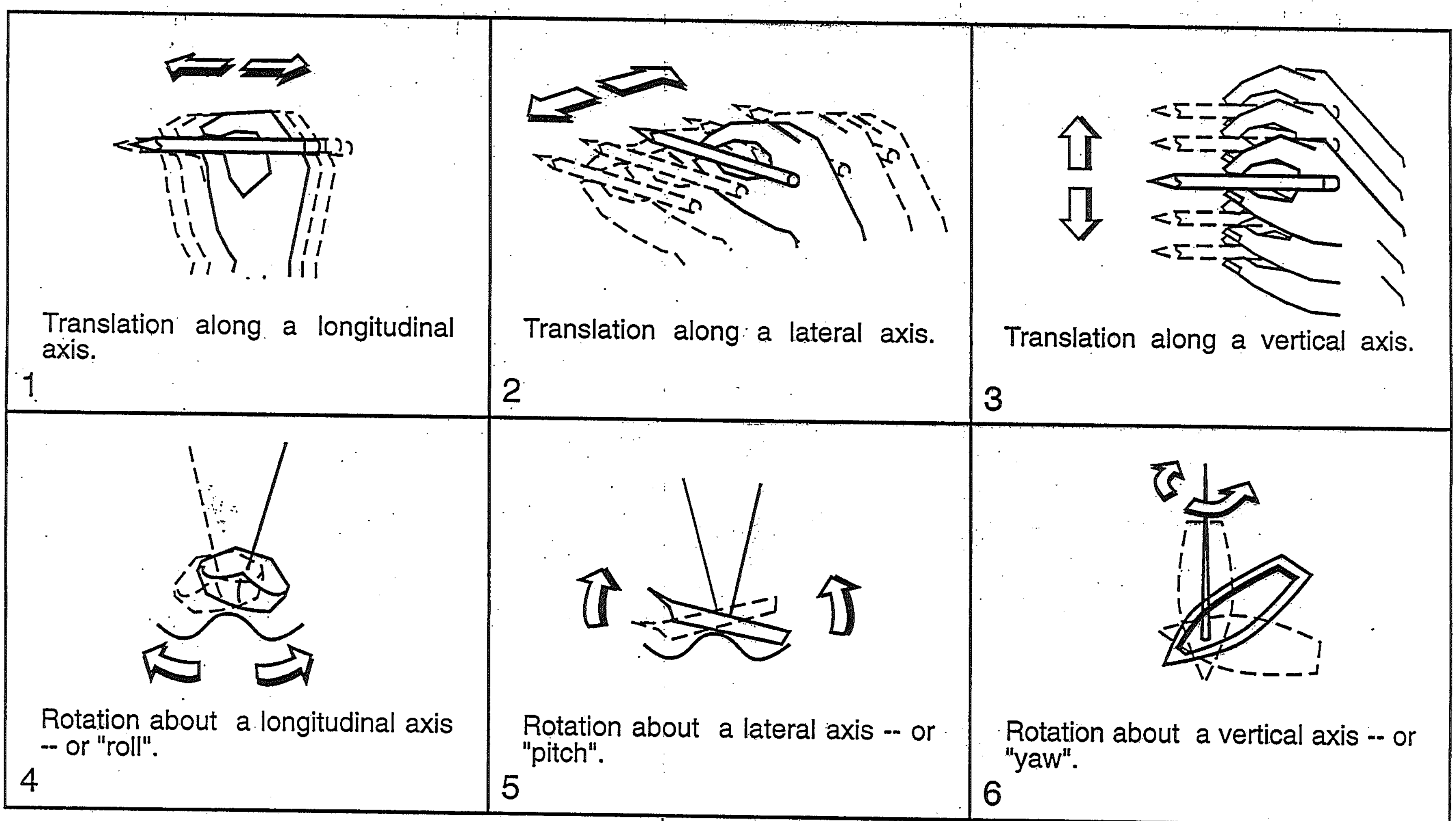
DYNAMIC DISTURBANCE GENERATED BY INTERNAL COMBUSTION ENGINES

FIGURE 1



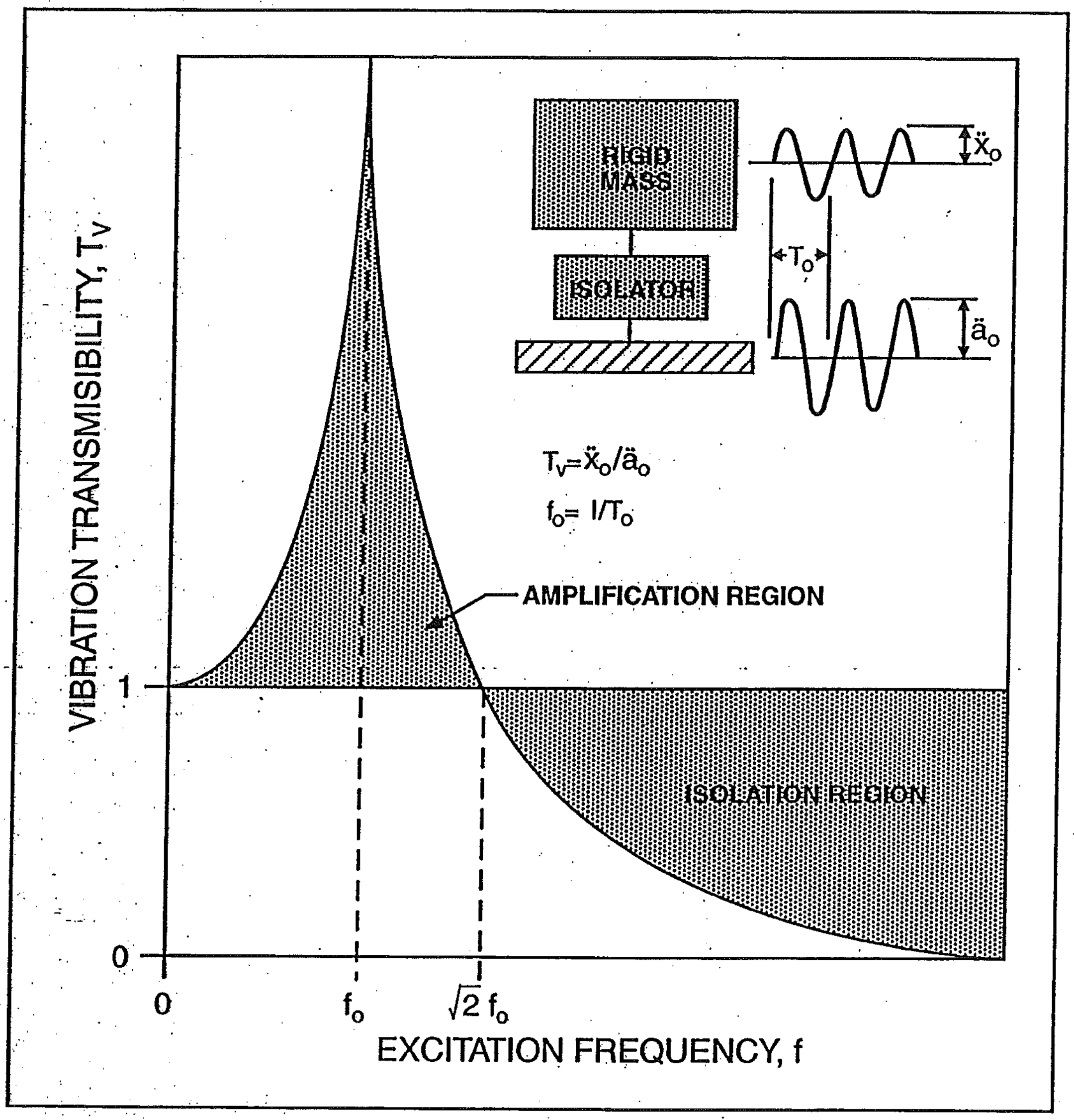
TRANSLATIONAL AND ROTATIONAL MODES OF AN ENGINE. X, Y, AND Z ARE THE PRINCIPAL AXIS OF LEAST MOMENT OF INERTIA

FIGURE 2



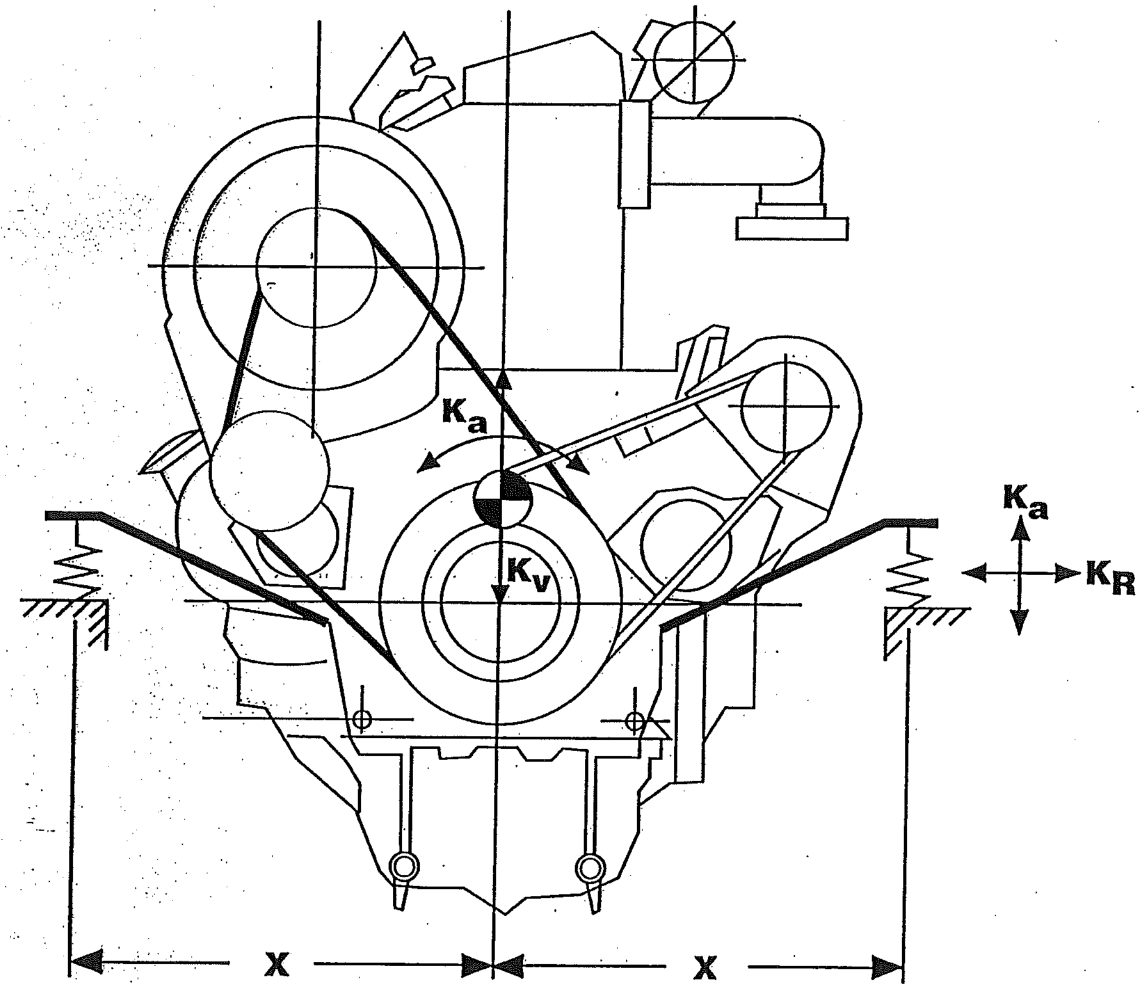
THE SIX MODES OF MOTION IN SPACE OF A RIGID BODY

FIGURE 3



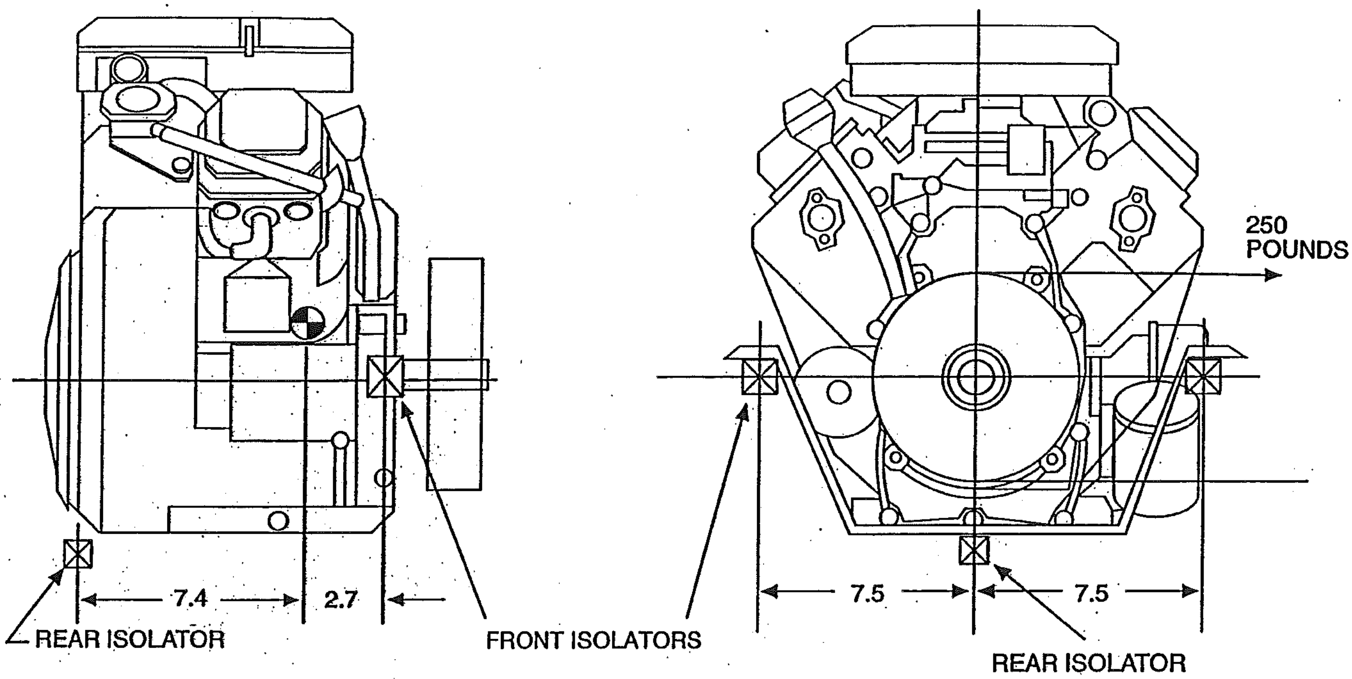
Qualitative representation of vibration tansmissibility for an undamped linear isolation system.

FIGURE 4

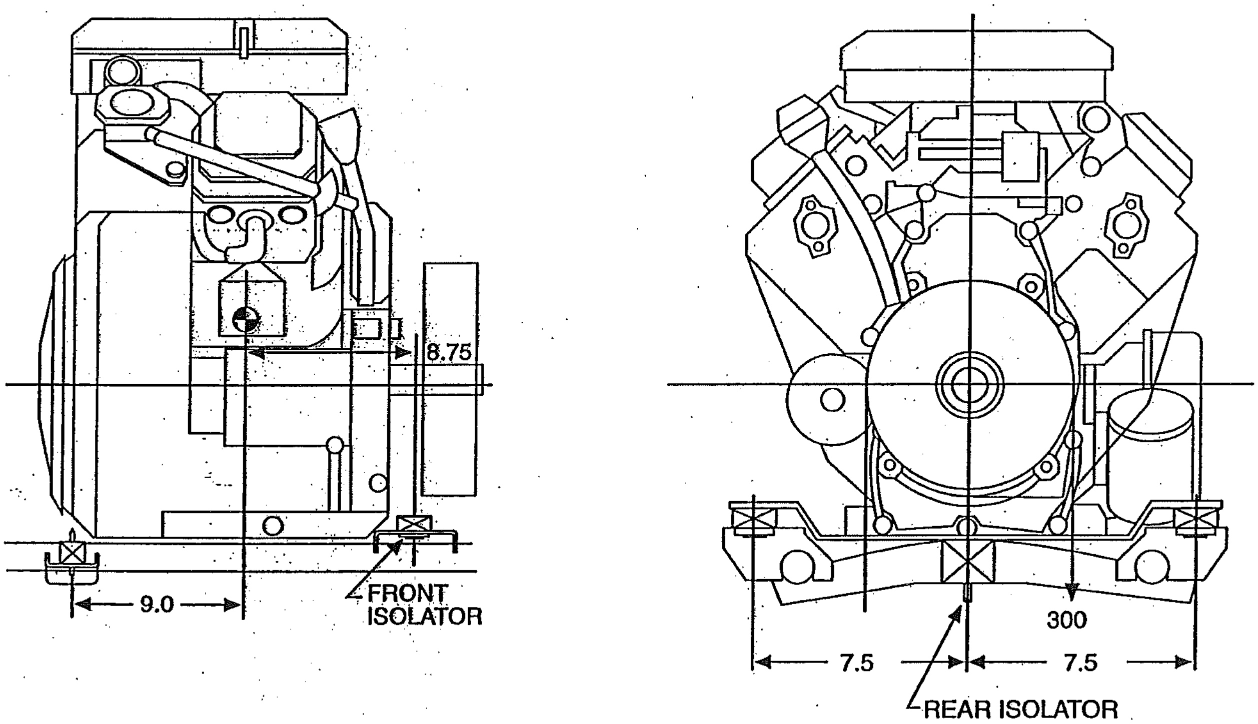


$$K_V = 2(Ka)$$
$$K_\Theta = 2(Ka)(x)^2$$

FIGURE 5



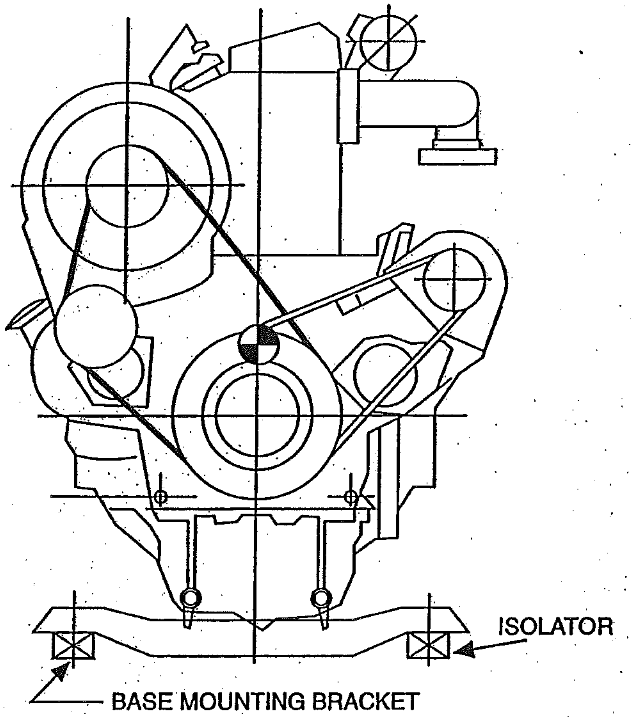
Horizontal Belt Pull



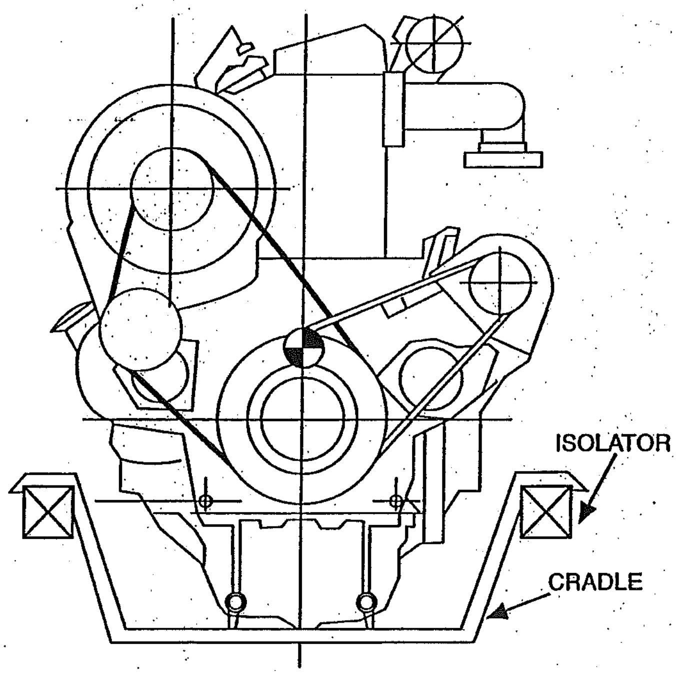
Vertical (Down) Belt Pull

SUGGESTED MOUNTING ARRANGEMENTS WHEN THE BELT POWER TAKE-OFF IS ACROSS THE MOUNTING SYSTEM

FIGURE 6



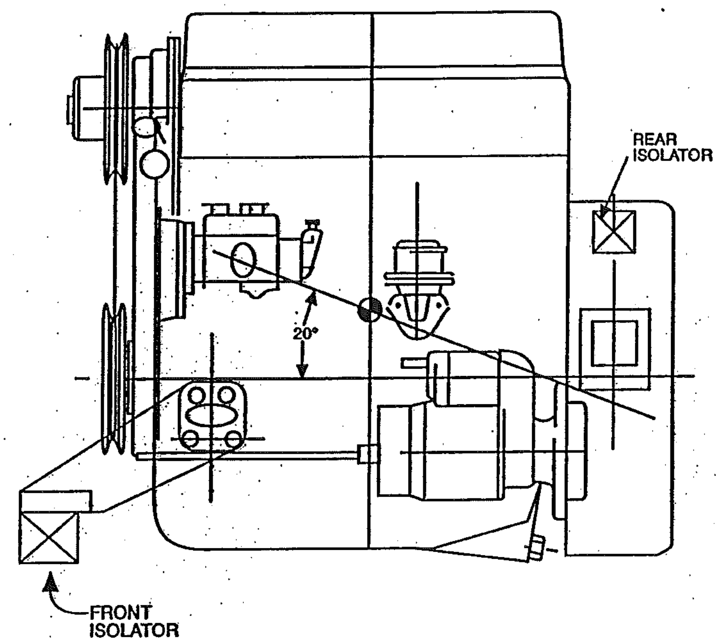
ORIGINAL SYSTEM



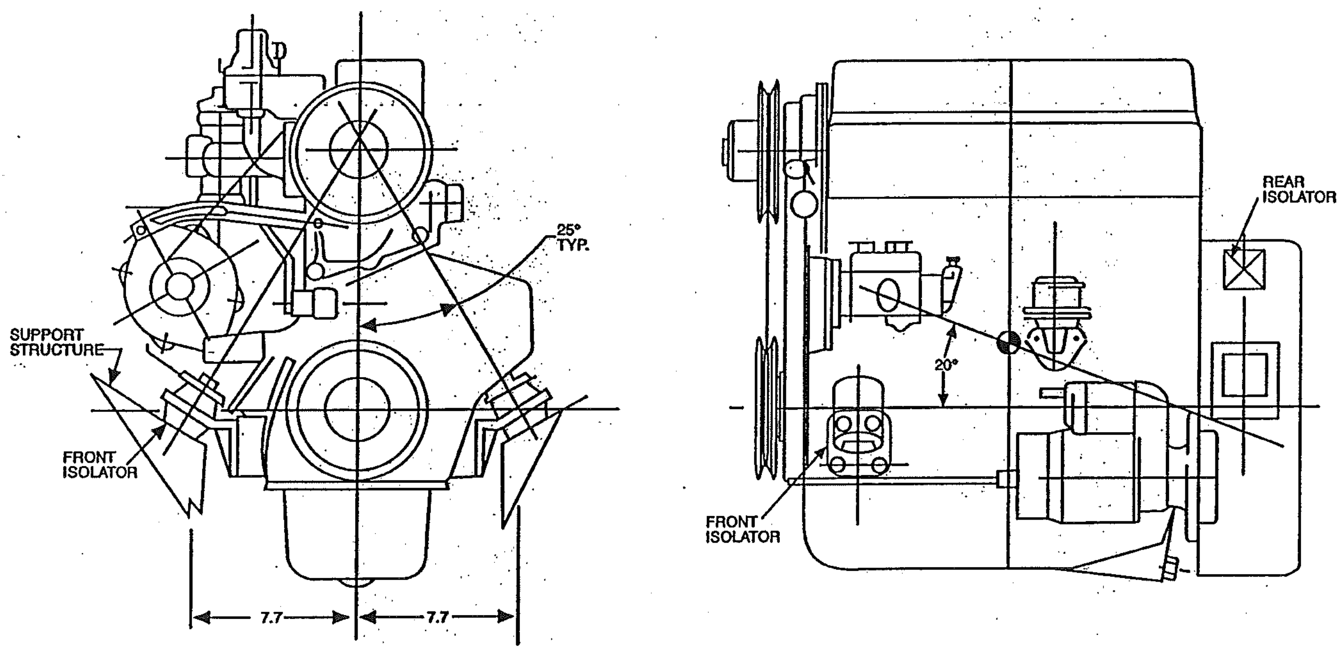
IMPROVED SYSTEM

CASE HISTORY NO.1

FIGURE 7



ORIGINAL SYSTEM



IMPROVED SYSTEM

CASE HISTORY NO.2

FIGURE 8

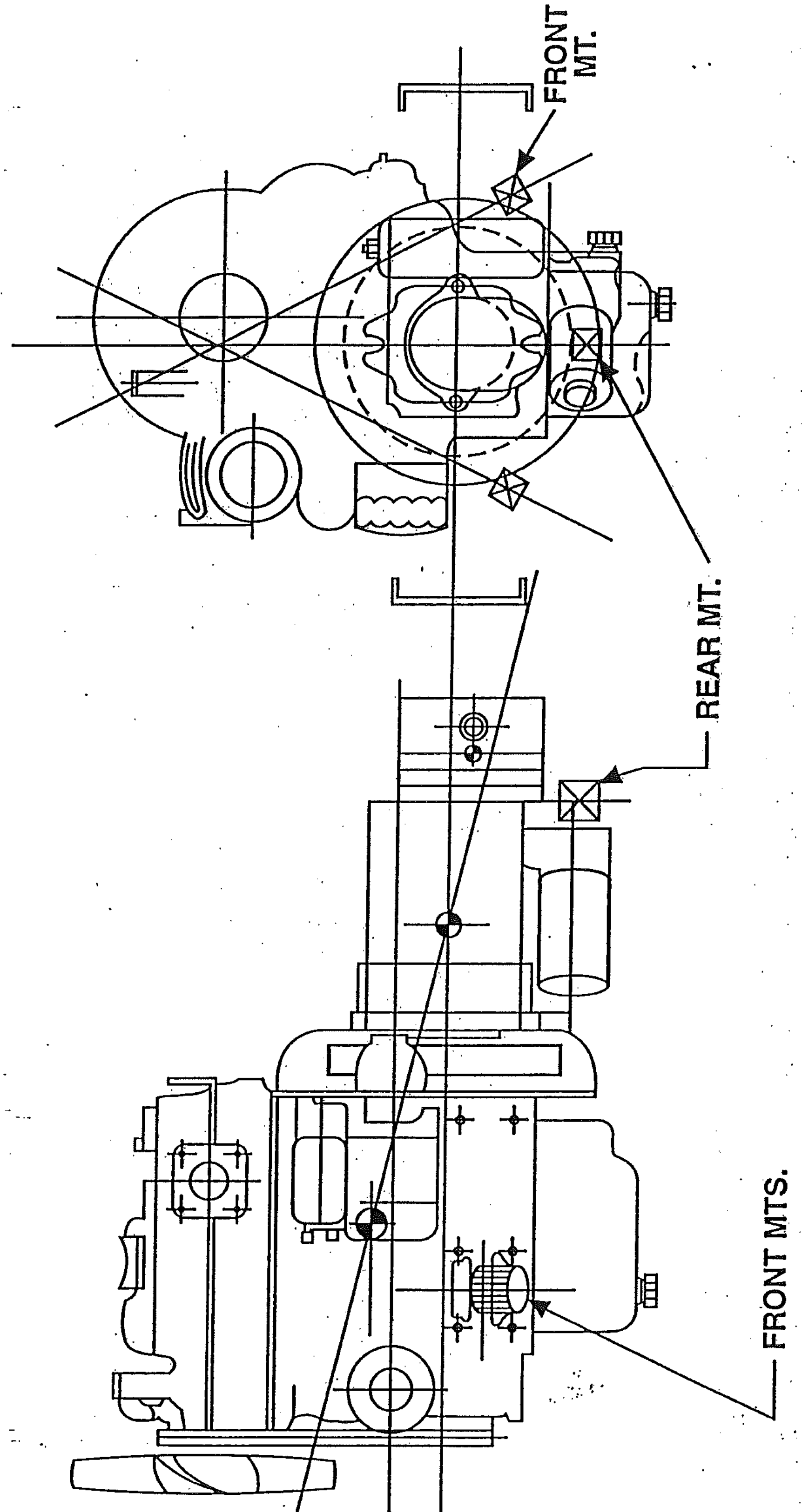


FIGURE 9