

# The Pitfalls of Combining Internal & External Equipment Isolation

## Introduction:

Almost all Make-up Air Units (**MAU**), Air Handling Units (**AHU**), and Rooftop Units (**RTU**) have internal spring-type isolation for the intake and exhaust fans. These fans typically have operating speeds that range from **400 rpm** to **1200 rpm**. Assuming an imbalance in the fan wheel, these operating speeds produce a forcing frequency (**f**) that will range between **6.67 Hz** and **20 Hz**. There are many buildings that have natural frequencies of their structures that lie between **8 Hz** and **12 Hz**.

There are a number of reasons for using internal isolation for the intake and exhaust fans. It may be intended to reduce the excitation of the building structure. It can be a feature added as a sales tool or to assist the passage of noise test requirements prior to shipment. Lastly it may help prevent fatigue failure of the cabinet. In any case, it should be noted that typically the only components that receive spring isolation in the equipment cabinets are these intake and exhaust fans.

There is a trend to use both internal and external isolation with the idea that more isolation will always reduce the transmission of noise and vibration. This is not necessarily true. In fact the use of both internal and external isolation can increase the level of noise and vibration transfer to the building's structure. The purpose of this paper is to demonstrate the conditions under which this might occur.

## Traditional Approach to Equipment Isolation:

Traditionally, external equipment isolation has been chosen by assuming that the equipment and spring isolation form a single degree of freedom system in which damping is neglected as shown in Figure G3.12.1-1 below. In this approach, the center of gravity (**C.G.**) of the equipment was assumed to lie at the geometrical center of the equipment. Also, it is usually assumed that all of the forces that would cause oscillation of the equipment act in the vertical direction. Thus, the equipment will experience only vertical motion. The stiffness and mass of the building are not taken into account by the model shown in Figure G3.12.1-1.

In Figure G3.12.1-1, **M** refers to the mass of the equipment, and **K** refers to the combined stiffness of all of the spring isolators that support the equipment. Normally the vibration isolation industry will refer to the stiffness of the springs in terms of the static deflection of the springs. The term static deflection actually entails more than the just the stiffness of the springs that support a given piece of equipment. The static deflection is a quantity that incorporates the mass of the equipment and the stiffness of the springs that support it.

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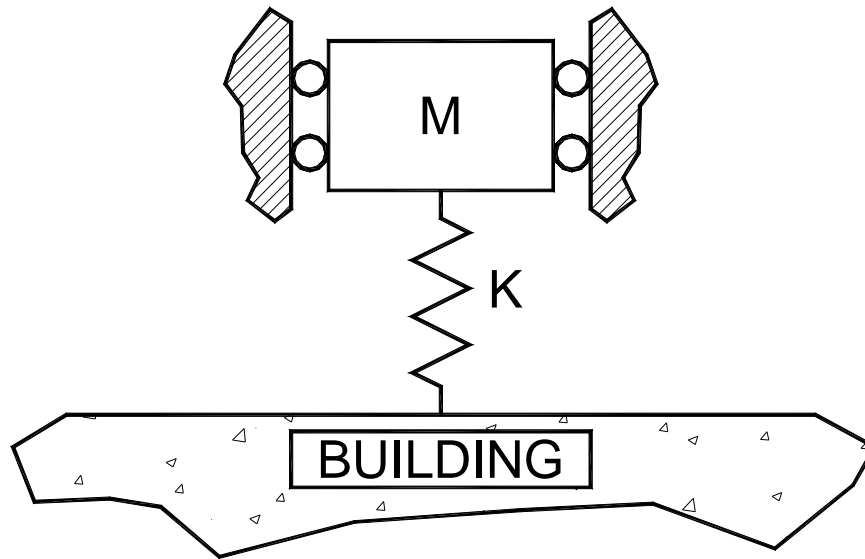
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**Figure G3.12.1-1: Basic Single Degree of Freedom Vibration Model.**

Equation G3.12.1-1 defines the natural frequency of the system shown in Figure G3.12.1-1.

$$f_N = [(M/K)^{1/2}] / (2 * \pi) = [(g/\Delta)^{1/2}] / (2 * \pi) \quad (\text{Eq. G3.12.1-1})$$

In Equation G3.12.1-1,  $f_N$  is the natural frequency of the system and  $g$  is the acceleration due to gravity, which has a value of **386.4 in/sec<sup>2</sup>**. The variable  $\Delta$  denotes the **Static Deflection** of the mass,  $M$ , which represents the equipment and the spring(s).

Unlike internal isolation springs, which are not normally “tuned” to the building structure, manufacturers of external equipment isolation will offer springs that have **1”, 2”, 3”, and 4” Static Deflections**. The natural frequencies represented by these springs are given in Table G3.12.1-1. The springs chosen for a given application will depend on the type of equipment to be supported, the position of the equipment in/on the building and the stiffness (flexibility) of the supporting building structure. The guidelines used for selection of springs to support equipment of various types are given in the Heating, Ventilating, and Air-Conditioning Applications volume of the **ASHRAE Handbook**. The standard spring families offered by Kinetics Noise Control are **1”, 2”, and 4” Static Deflections**. The improvement, over a **2” Static Deflection** spring, gained by using a **3”** static deflection spring is marginal when compared to that gained by using the **4” Static Deflection** spring. Therefore, Kinetics Noise Control chose to not offer the **3” Static Deflection** spring family as a standard product family.

If the forcing frequency ( $f$ ) of the rotating components of the equipment is close to the natural frequency of the equipment on the springs, then excess vibration and sound

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energy will be transferred to the building structure. How much vibration and sound energy is transferred is defined by the transmissibility. The transmissibility,  $T_R$ , is defined by Equation G3.12.1-2, and is discussed in more detail in Document T1.2.2. The larger the value of the transmissibility, the more vibration is transferred directly to the building structure. As the forcing frequency,  $f$ , approaches the natural frequency,  $f_N$ , the values of the transmissibility can exceed **1.0**. This is a situation where more vibration is felt by the structure than is being generated by the equipment.

**Table G3.12.1-1: System Natural Frequency vs. Static Spring Deflection.**

Static Spring Deflection $\Delta$ (in)	System Natural Frequency $f_N$ (Hz)
1	3.13
2	2.21
3	1.81
4	1.56

$$T_R = |1 / [(f/f_N)^2 - 1]| \quad \text{(Eq. G3.12.1-2)}$$

For the values of static deflection springs supplied by Kinetics Noise Control the transmissibility at fan speeds ranging from **500 rpm** to **1200 rpm** are plotted in Figure G3.12.1-2. These curves represent the performance of a piece of equipment with external isolation only. These plots assume that the fans are unbalanced, and thus there is one force per revolution of the fan.

Transmissibility,  $T_R$ , under **0.125** is desirable to minimize the amount of sound and vibration transmitted to the building. Figure G3.12.1-2 shows that this is achieved with external isolation for all instances, except for a **1" Static Deflection** at fan speeds below **600 rpm**. Of course, the building stiffness is not included in this analysis. As the building structure becomes more flexible, **2"** and **4" Static Deflection** springs would need to be used to "protect" the more sensitive building structure. It should be noted that the **1" Static Deflection** and **2"** Static Deflection curves in Figure G3.12.1-2 would apply to internal isolation only, as well. In almost any piece of rooftop equipment, it would be impractical to use springs with a static deflection of **4"** for internal component isolation.

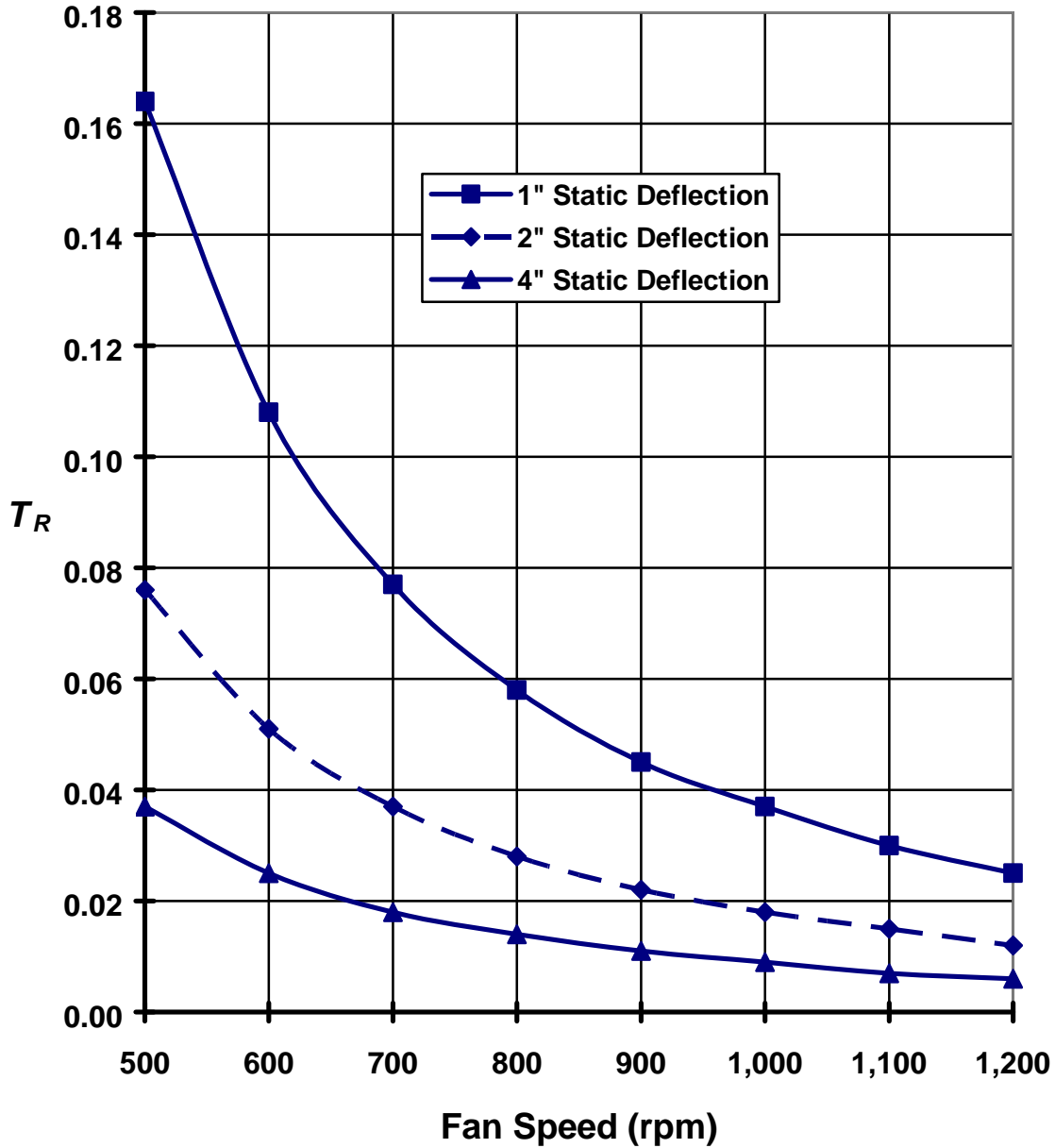
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Figure G3.12.1-2: Transmissibility vs. Fan Speed  
External Isolation Only

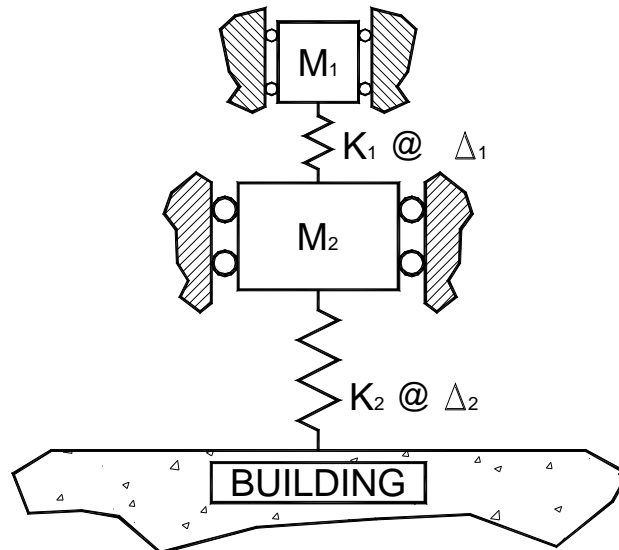


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## Using Internal and External Equipment Isolation:

As with the traditional approach, the **C.G.** of the equipment is assumed to be at its geometrical center, and damping is neglected. It is also assumed that the forces that would cause the equipment to move will act in the vertical direction only. Thus, as before, the equipment will experience vertical motion only. The stiffness and mass of the building will not be included in the model to be described below. The recommendations that are found in the ASHRAE Handbook should be followed in regard to the building stiffness as it applies to the selection of spring isolation for the various types of equipment.

Equipment with both internal and external isolation may be modeled, in its simplest form, as a two degree of freedom system. The representation of the model is shown in Figure G3.12.1-3. In this model,  $M_1$  is the mass of the internally isolated components. These components are usually the intake and exhaust fans.  $K_1$  is the stiffness of the internal isolation springs and, with the mass,  $M_1$  produces a static deflection of the internal springs equal to  $\Delta_1$ .  $M_2$  represents the mass of the rest of the equipment; that is, the non-isolated internal components and the cabinet.  $K_2$  is the stiffness of the external isolation springs and with the total mass of the equipment ( $M_1+M_2$ ) produces a static deflection of the external springs equal to  $\Delta_2$ .



**Figure G3.12.1-3: Basic Two Degree of Freedom Vibration Model.**

The system shown in Figure G3.12.1-3 has two natural frequencies where resonance will occur if the system is driven at, or close to, one or the other of those natural frequencies. The number of degrees of freedom for a given system is determined by the number of independent variables required to describe the motion of the system. For the system in Figure G3.12.1-1, only one variable will be required to describe the motion of the mass  $M$ . For the system shown in Figure G3.12.1-3, one independent variable for each of the

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masses,  $M_1$  and  $M_2$ , will be required to completely describe their motions. Thus, this system has two degrees of freedom. The mass  $M_1$  represents the internally isolated component which in our case is the fan. The remainder of the piece of equipment is represented by the mass  $M_2$ . The combined stiffness of the internal isolation springs is shown as  $K_1$ . The combined stiffness of the external isolation springs is represented by  $K_2$ .

The analysis involves determining the equations of motion for each mass. This will result in a system of two simultaneous second order differential equations. The solution for this system of equations is a classic one, and is described and demonstrated in Document T1.1.6.

The natural frequencies for systems with various configurations were determined. Three different types of units were examined. The mass ratio  $[M_1/(M_1+M_2)]$  is typically different for each of the three types of equipment. For the **MAU**, the mass of the isolated components was found to be approximately one-half the total mass of the unit  $[M_1=0.5*(M_1+M_2)]$ . The mass of the isolated components in the **AHU** were found to be roughly one-fourth the total mass of the unit  $[M_1=0.25*(M_1+M_2)]$ . Finally, the mass of the isolated components for an **RTU** was found to be about one-eighth the total mass of the unit  $[M_1=0.125*(M_1+M_2)]$ .

The literature for one of the manufacturers of roof top equipment indicates that the standard springs used for the internally isolated components would have a **2" Static Deflection**. Another manufacturer of roof top equipment used springs with a **1" Static Deflection** as the standard selection for the internally isolated components. The optional springs had a **2" Static Deflection**. The third manufacturer investigated, advertised the use of internal isolation but did not indicate the **Static Deflection** produced in the springs by the mass of the internally isolated components. It is possible that in some cases the **Static Deflection** of the internal isolation springs may be less than **1"**.

For each of the units mentioned above, several combinations of internal and external isolation springs were investigated. The cases investigated are listed below in Table G3.12.1-2. The first ( $f_{N1}$ ) and second ( $f_{N2}$ ) natural frequencies for these cases following were calculated and are given in Table 3.12.1-3.

In Table G3.12.1-3, the first natural frequency is the higher of the two. It is the one that will cause the most problems from a vibration and sound energy transfer point of view. Examination of the tabulated results above shows that there are several combinations that may experience problems with excessive vibration and/or noise. The problem combinations will show themselves more clearly using transmissibility plots similar to the one in Figure G3.12.1-2.

The transmissibility will be calculated using the first natural frequency for each case and combination. Equipment curves for each case will be plotted on the same graph along

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with the transmissibility curve for the appropriate external isolation spring. These graphs are shown below in Figures G3.12.1-14 through G3.12.1-10.

**Table G3.12.1-2: Cases Considered in This Study.**

Case No.	Static Deflection of Internal Springs $\Delta_1$ (in)	Static Deflection of External Springs $\Delta_2$ (in)	Applied to Equipment Type
1	2.00	1.00	<i>MAU, AHU, and RTU</i>
2	2.00	2.00	<i>MAU, AHU, and RTU</i>
3	2.00	4.00	<i>MAU, AHU, and RTU</i>
4	1.00	1.00	<i>MAU, AHU, and RTU</i>
5	1.00	2.00	<i>MAU, AHU, and RTU</i>
6	1.00	4.00	<i>MAU, AHU, and RTU</i>
7	0.50	1.00	<i>MAU, AHU, and RTU</i>

In Figures G3.12.1-4 through G3.12.1-10, the curve labeled **No Internal** would correspond to the use of external isolation only. Please note that in all cases the transmissibility for external isolation only is lower and, in most instances, much lower than any combination that uses internal and external isolation. The performance of the *MAU*, *AHU*, and *RTU* with both internal and external isolation approaches the performance of the units with external isolation alone, only as the fan speed increases beyond, say, **800 rpm**. Also, as the mass of the internally isolated components decreases relative to the mass of the entire unit, the performance of the unit with both internal and external isolation approaches that of the unit with external isolation alone.

Figures G3.12.1-7 and G3.12.1-10 demonstrate conditions that could be potential problems for an *MAU* installation. In both cases the external isolation springs have a **1" Static Deflection**. The *MAU* in Figure G3.12.1-7 has internal isolation springs with a **1" Static Deflection**, and the *MAU* in Figure G3.12.1-10 has a **0.5" Static Deflection** for the internal isolation springs. Figure G3.12.1-10 represents a possible worst-case scenario. For the case in Figure G3.12.1-4, the transmissibility,  $T_R$ , is about **0.9** at a fan speed of **500 rpm**. This means that roughly **90%** of the noise and vibration produced by the fan will be transmitted to the structure. For the *MAU* in Figure G3.12.1-10, the transmissibility,  $T_R$ , at **500 rpm** is **2.85**. This is the potentially dangerous condition of resonance. In this case the vibrations can grow until the *MAU* or supporting isolators fail in some manner. Could this latter condition actually occur? If the equipment manufacturer specified the internal isolation springs incorrectly, it is conceivable that the internal isolation springs could

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indeed deflect less than 1" under the weight of the internally isolated components.

**Table G3.12.1-3: First and Second Natural Frequencies for Various Types of Equipment with Both Internal & External Isolation.**

Case No.	Equipment Type	Static Deflection of Internal Springs $\Delta_1$ (in)	Static Deflection of External Springs $\Delta_2$ (in)	First Natural System Frequency $f_{N1}$ (Hz)	Second Natural System Frequency $f_{N2}$ (Hz)
1	MAU	2.00	1.00	5.06	1.93
1	AHU	2.00	1.00	3.93	2.03
1	RTU	2.00	1.00	3.52	2.11
2	MAU	2.00	2.00	4.09	1.70
2	AHU	2.00	2.00	3.13	1.81
2	RTU	2.00	2.00	2.75	1.91
3	MAU	2.00	4.00	3.59	1.36
3	AHU	2.00	4.00	2.78	1.43
3	RTU	2.00	4.00	2.49	1.49
4	MAU	1.00	1.00	5.78	2.40
4	AHU	1.00	1.00	4.42	2.56
4	RTU	1.00	1.00	3.90	2.70
5	MAU	1.00	2.00	5.07	1.93
5	AHU	1.00	2.00	3.93	2.02
5	RTU	1.00	2.00	3.52	2.10
6	MAU	1.00	4.00	4.42	2.22
6	AHU	1.00	4.00	3.75	1.51
6	RTU	1.00	4.00	3.40	1.53
7	MAU	0.50	1.00	7.17	2.72
7	AHU	0.50	1.00	5.56	2.86
7	RTU	0.50	1.00	4.98	2.98

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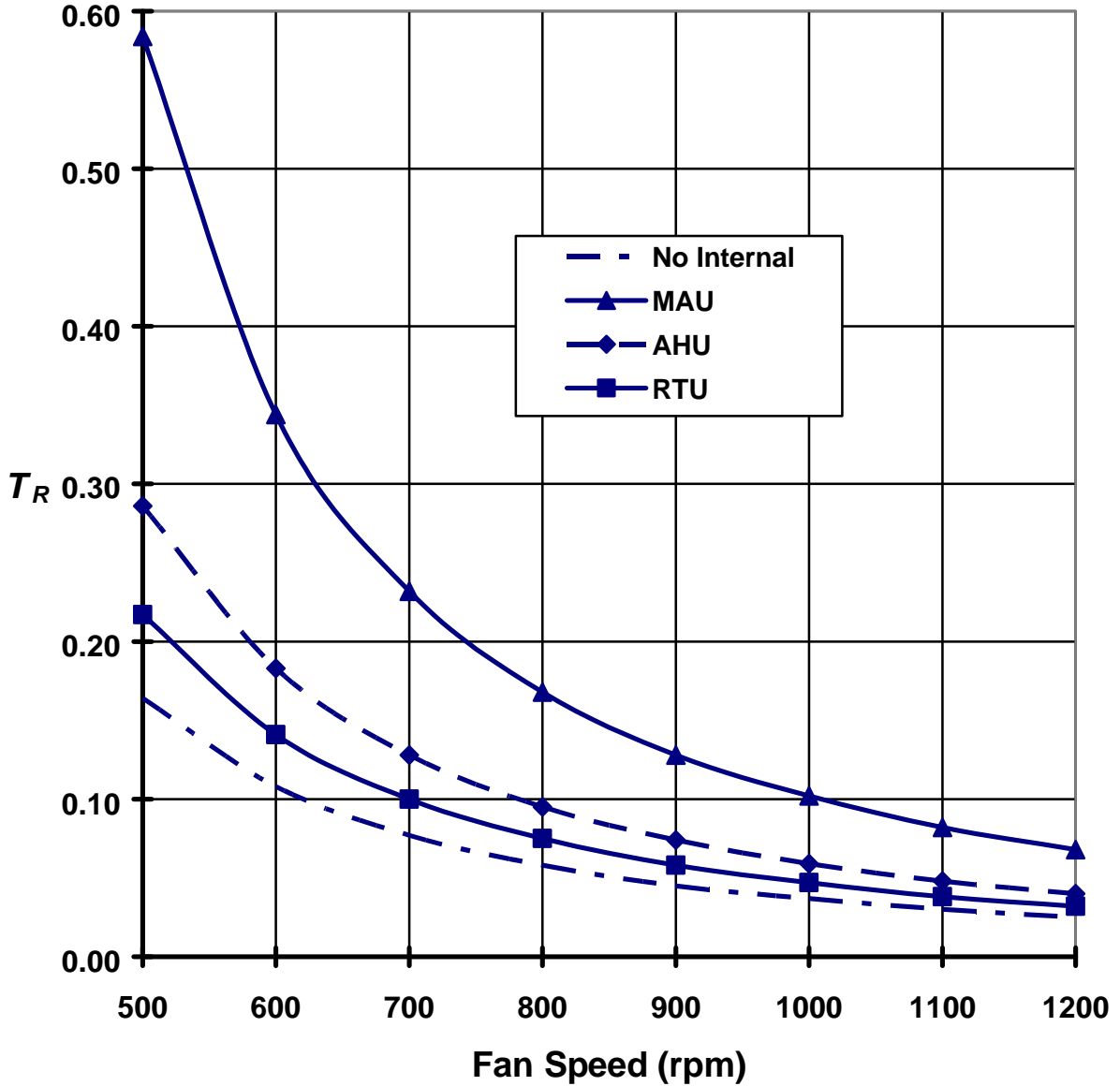


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Figure G3.12.1-4: Case No. 1 - Transmissibility vs. Fan Speed  
 Internal Isolation = 2" Static Deflection Springs  
 External Isolation = 1" Static Deflection Springs



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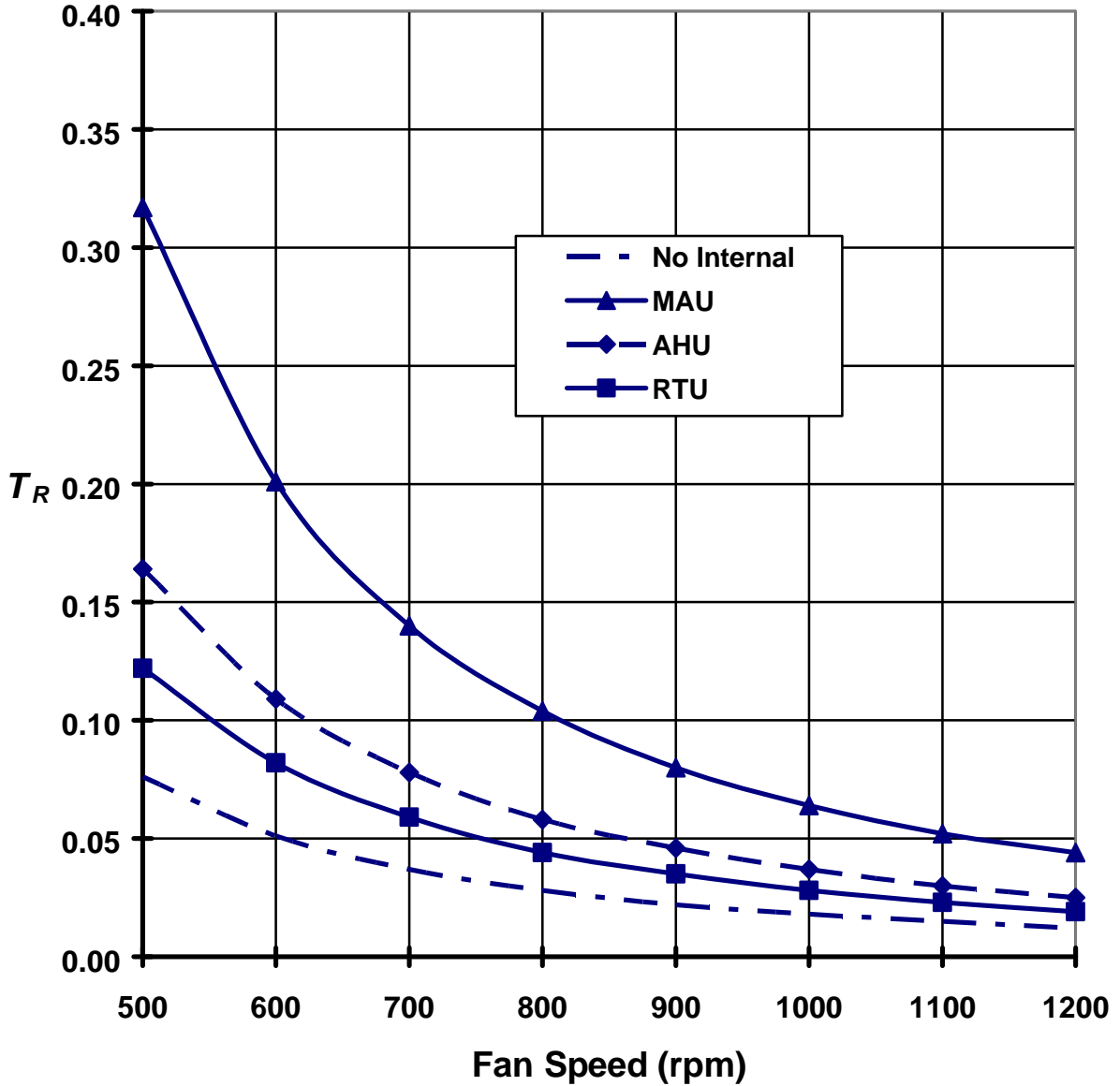
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Figure G3.12.1-5: Case No. 2 - Transmissibility vs. Fan Speed  
 Internal Isolation = 2" Static Deflection Springs  
 External Isolation = 2" Static Deflection Springs



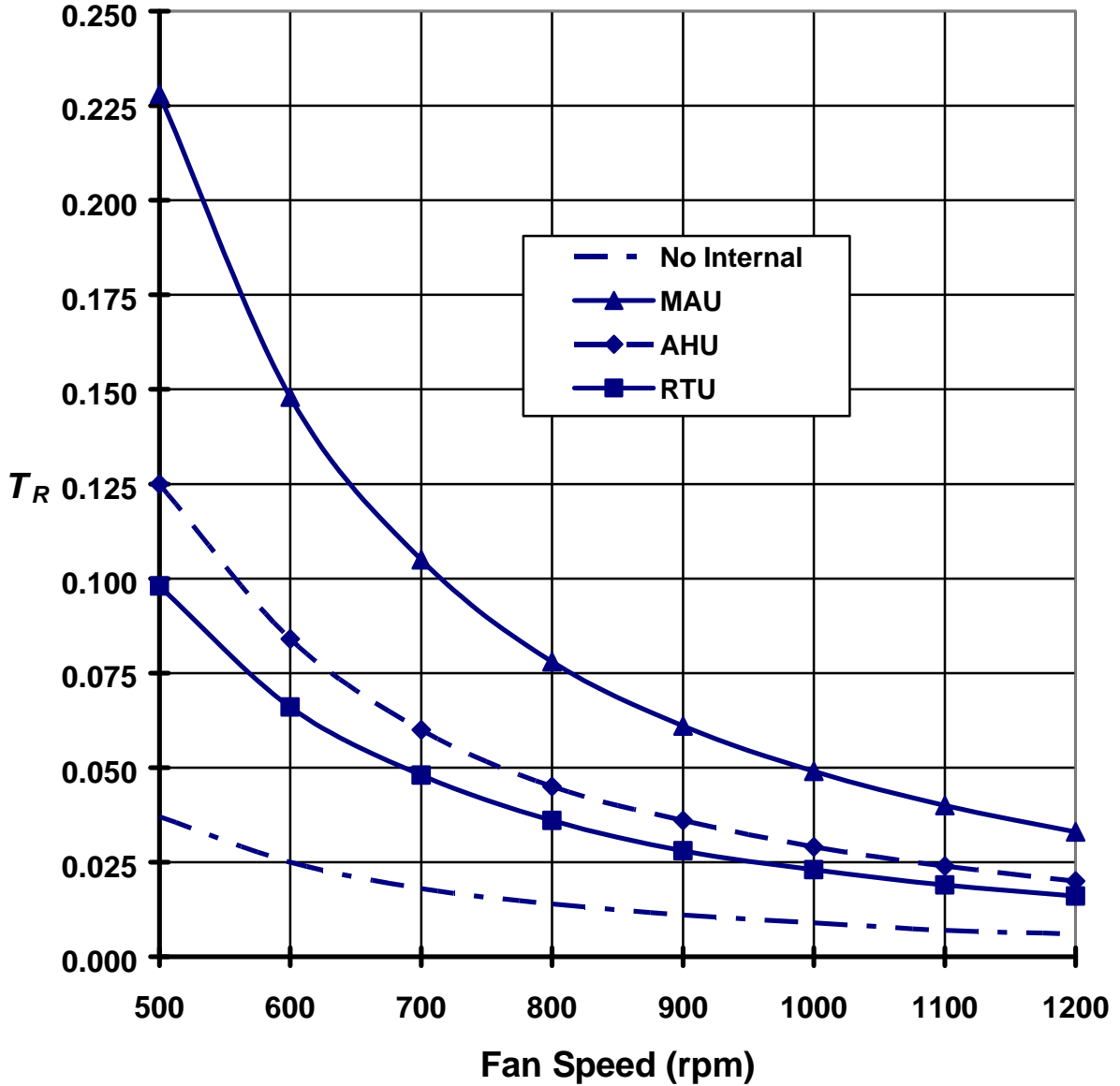
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Figure G3.12.1-6: Case No. 3 - Transmissibility vs. Fan Speed  
 Internal Isolation = 2" Static Deflection Springs  
 External Isolation = 4" Static Deflection Springs



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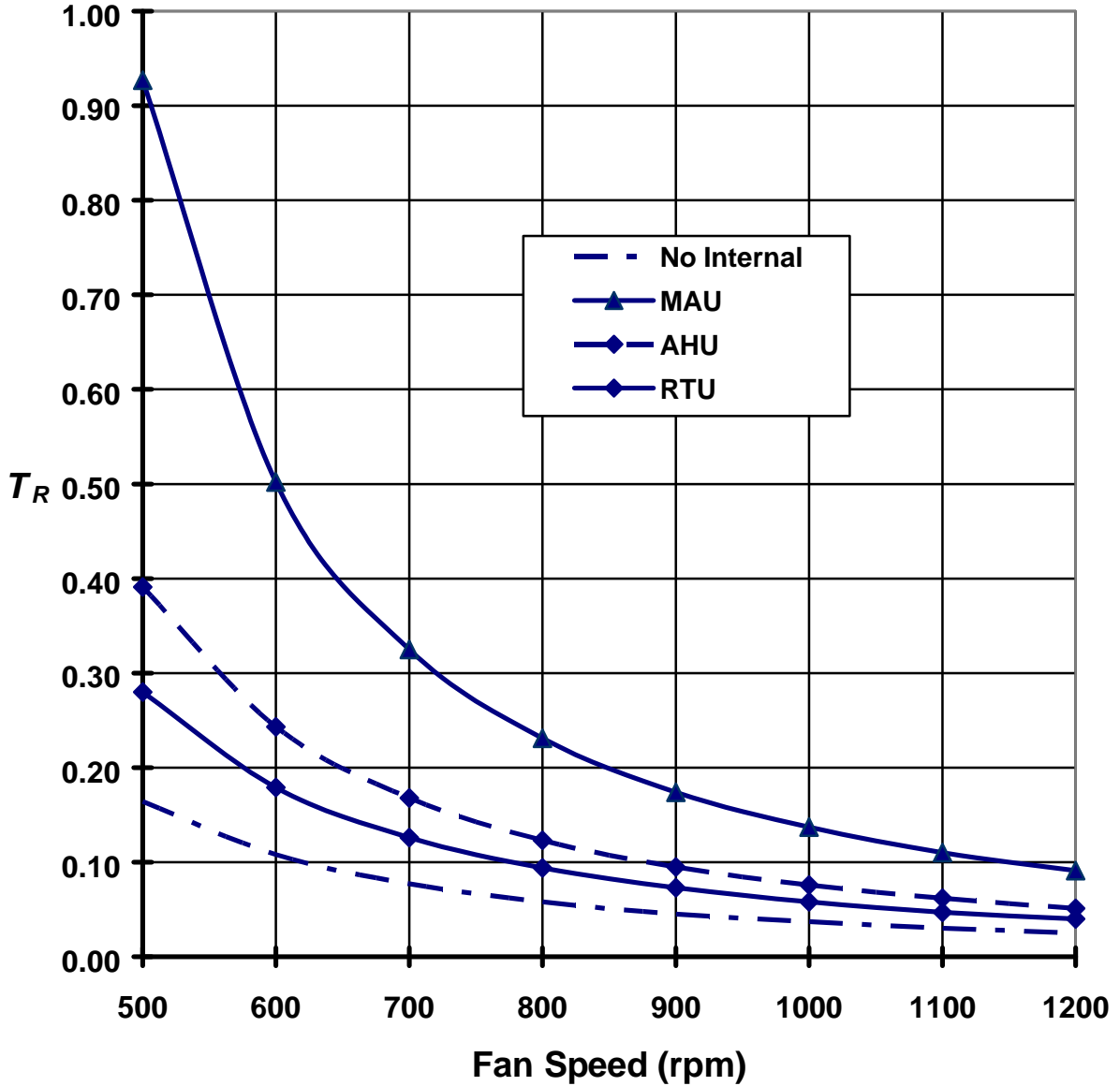
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Figure G3.12.1-7: Case No. 4 - Transmissibility vs. Fan Speed  
 Internal Isolation = 1" Static Deflection Springs  
 External Isolation = 1" Static Deflection Springs



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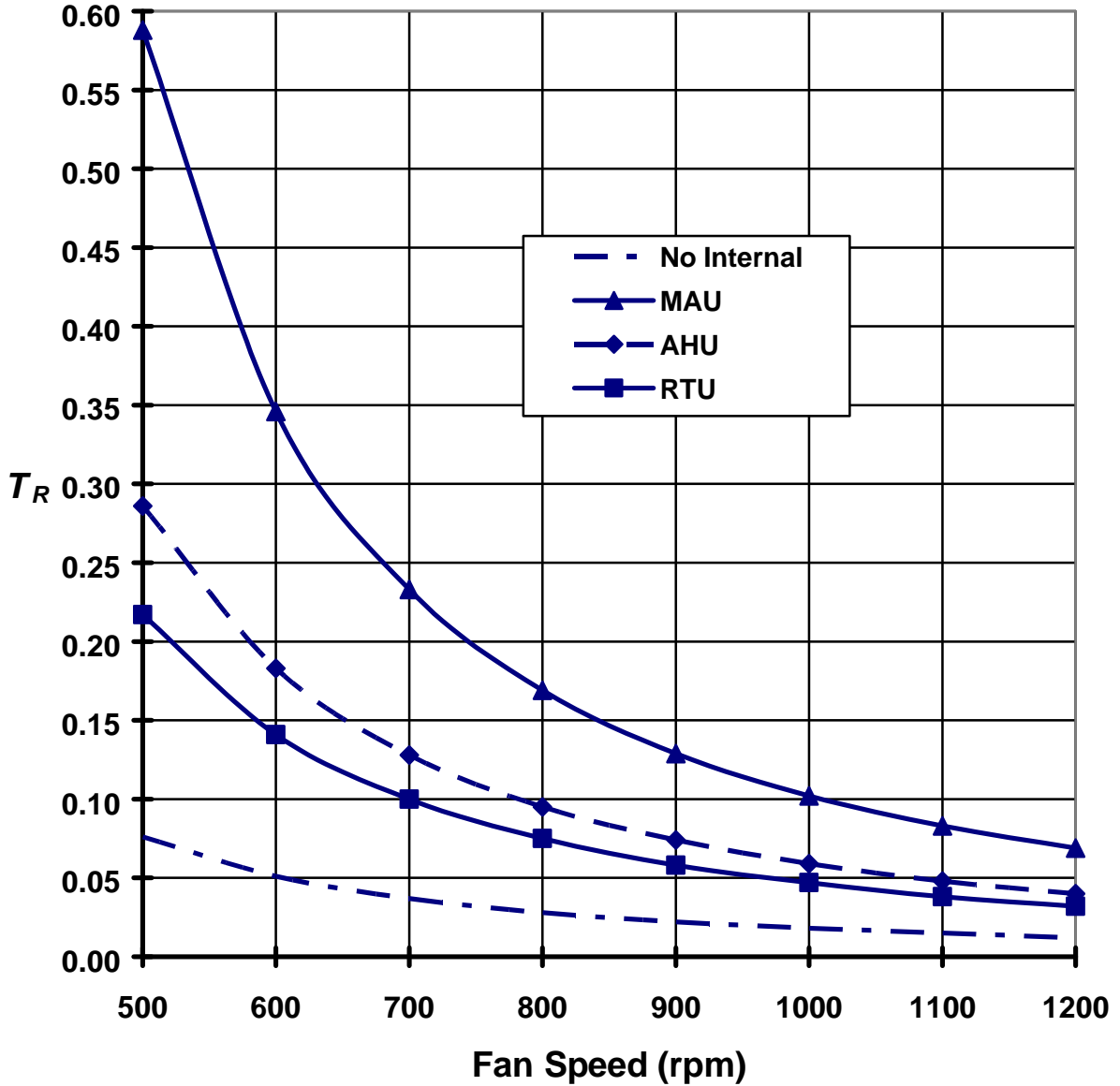
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**Figure G3.12.1-8: Case No. 5 - Transmissibility vs. Fan Speed**  
 Internal Isolation = 1" Static Deflection Springs  
 External Isolation = 2" Static Deflection Springs



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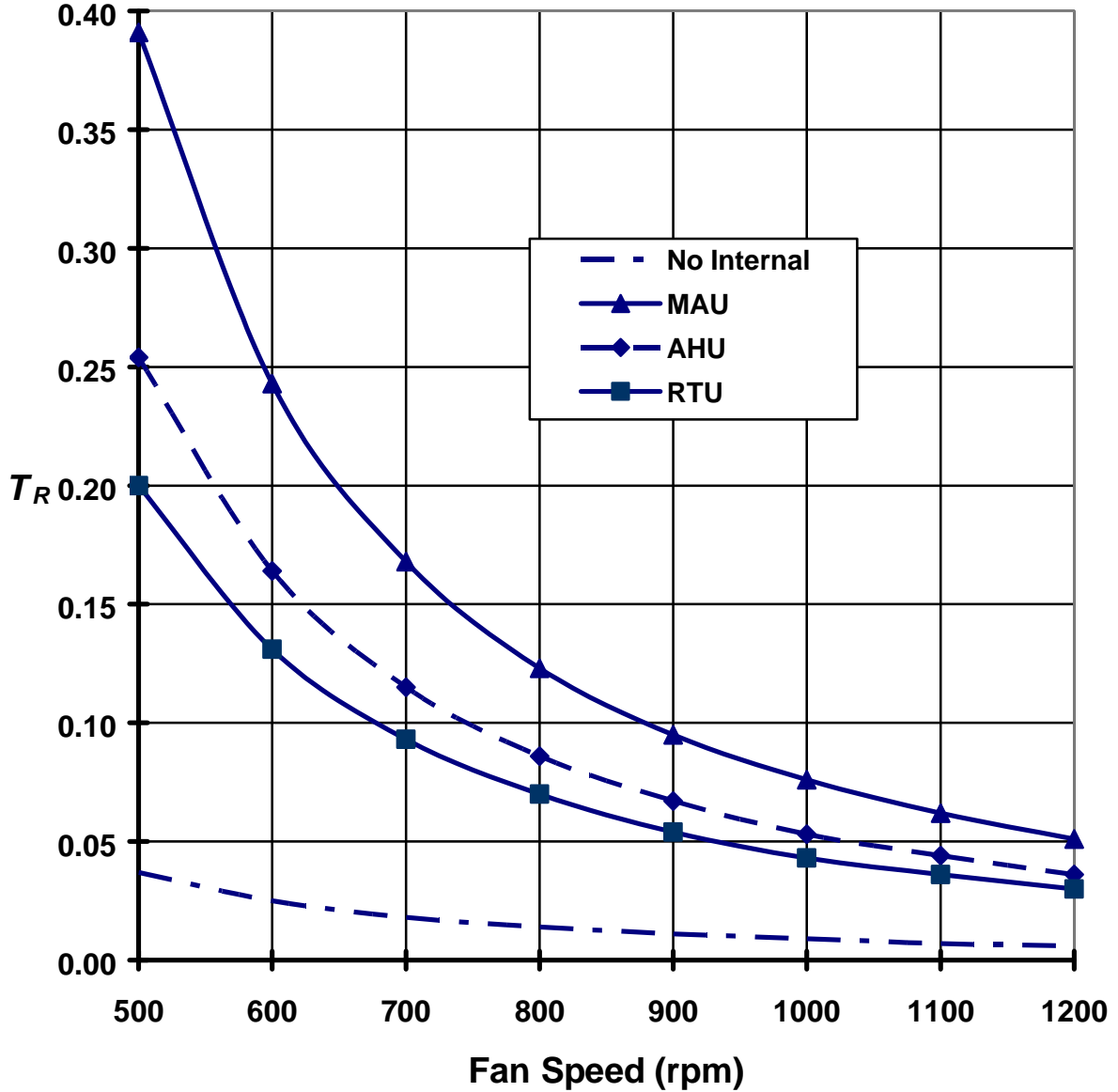
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Figure G3.12.1-9: Case No. 6 - Transmissibility vs. Fan Speed  
 Internal Isolation = 1" Static Deflection Springs  
 External Isolation = 4" Static Deflection Springs



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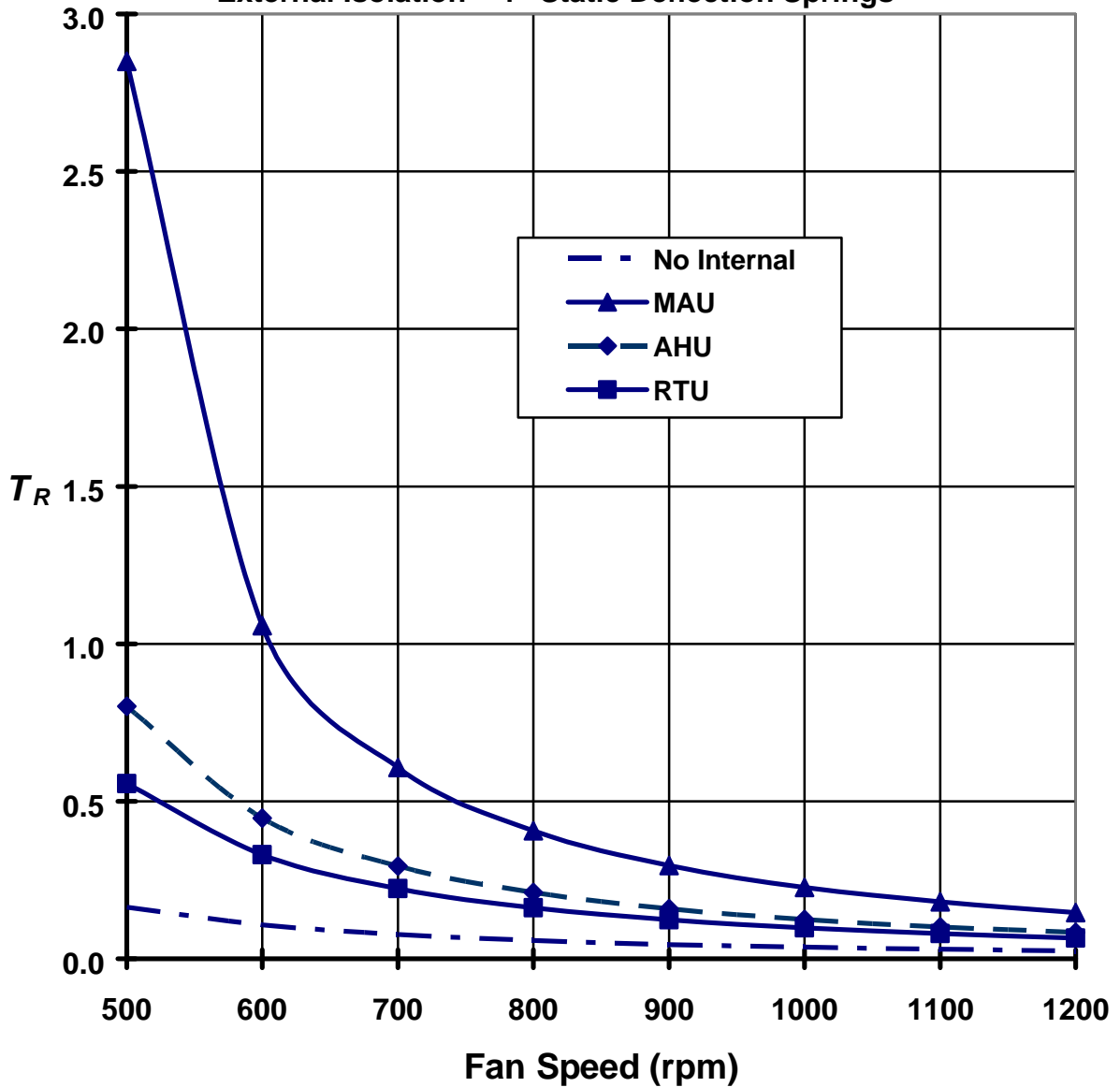


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Figure G3.12.1-10: Case No. 7 - Transmissibility vs. Fan Speed

Internal Isolation = 0.5" Static Deflection Springs  
 External Isolation = 1" Static Deflection Springs



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Figures G3.12.1-4 through G3.12.1-10 clearly demonstrate that the use of external equipment isolation alone will always produce results that will be superior to those achieved by having both internal and external isolation on the same piece of equipment. That is, the external isolation when used by it self will always have a lower value for the transmissibility,  $T_R$ , than when used in conjunction with internal isolation. This means that less sound and vibration will be transferred to the building structure when just the external equipment isolation is used. Care should be taken to ensure that the internal isolation springs are “blocked out” before operating the equipment if external equipment isolation has been installed.

**Using Only Internal Equipment Isolation:**

Many acoustical and vibration isolation consultants feel that the internal isolation provided with the rooftop unit will hold the transmitted vibration and noise to acceptable levels, and sometimes, depending on the application, this may be true. However, keep in mind that there are many instances that will require that the entire cabinet be isolated. These cases will be discussed shortly. First we should examine the curves for the transmissibility for the use of internal equipment isolation alone. Four sets of isolation springs were chosen, **0.25" Static Deflection, 0.50" Static Deflection, 1" Static Deflection, and 2" Static Deflection.** The **0.25" Static Deflection** case is included because this is the deflection range expected for a neoprene isolator. It is possible that some smaller units may in fact use this type of isolator due to space restrictions in the equipment cabinet. The **0.50" Static Deflection** case is included to show what can happen if the internal isolation springs are not properly selected. The Transmissibility vs. Fan Speed curves for these four cases are shown in Figure G3.12.1-11.

The use of **1" and 2" Static Deflection** springs will produce the same values for the transmissibility,  $T_R$ , as seen in Figure G3.12.1-2 for external isolation only. Thus, if the fans are the only source of noise and vibration, the rooftop unit would be adequately isolated for most applications. However, if there are other sources of sound and vibration in the equipment cabinet, they would not be isolated. Also, if the walls of the cabinet were to be excited by the fan, any other internal component, or by the action of the wind, then this sound or vibration would not be isolated from the building.

Internal isolation springs not properly selected will cause sound and vibration issues at fan speeds below **800 rpm** or so. The use of neoprene isolators for the fans should be avoided. They will allow excessive noise and vibration to pass on to the building. Below **600 rpm** the neoprene springs may even amplify the effects of the sound and vibration that are transmitted to the building.

Finally, for buildings where perceptible transmission of excess sound and vibration is absolutely unacceptable, the use of **4" Static Deflection** external isolation springs alone is recommended for rooftop units.

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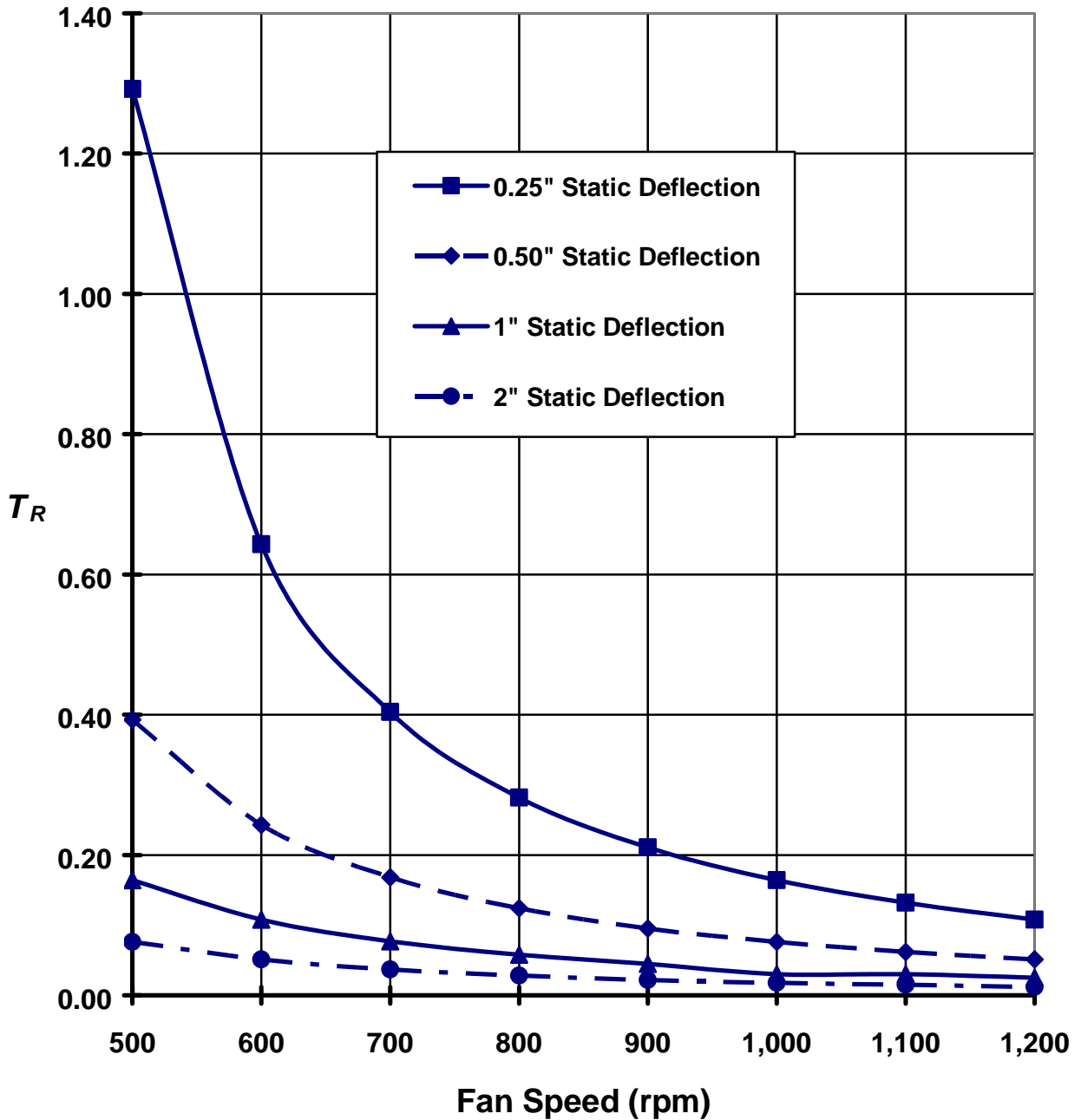


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Figure G3.12.1-11: Transmissibility vs. Fan Speed  
Internal Isolation Only



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## Conclusions:

- 1.) In all cases investigated, equipment that had external isolation only had lower transmissibility values than equipment with both internal and external isolation.
- 2.) The harmful effects of using both internal and external isolation are amplified as the operating speed of the internally isolated components decreases. The reason for this is shown in the first natural frequency indicated in Table G3.12.1-3. They are, in some instances, very close to the forcing frequency of a **500 rpm** fan which is **8.33 Hz**.
- 3.) As the mass of the internally isolated components increases relative to the mass of the entire unit, the harmful effects of the internal and external isolation become more pronounced. When the mass of the internally isolated components becomes a significant percentage of the whole mass of the unit, the use of internal and external isolation could have disastrous results when combined with the lower fan speeds.
- 4.) In some of the cases in Table G3.12.1-3, the first natural frequency of the internally and externally isolated equipment approached the natural frequency of some buildings. In these cases, significant transmission of low frequency noise and vibration could be expected.
- 5.) Any condition that moves the **C.G.** of the internally isolated components and/or the entire piece of equipment off of the geometric center of the equipment will introduce additional degrees of freedom to the system. This will result in additional natural frequencies, resonance conditions, which must be avoided. Thus it behooves the owner to reduce the number of resonance conditions by eliminating the internal or the external isolation. However, as demonstrated above, external isolation performs better than internal isolation.
- 6.) If both internal and external isolation are specified, choose external isolation springs with a set of either **2"** or **4" Static Deflection** springs based on the entire mass of the equipment. The external isolation springs with the **4" Static Deflection** are preferable.
- 7.) If the equipment installation is sensitive to the transmission of vibration and noise, internal isolation should not be used. The use of external isolation alone always produces better results than the use of both internal and external isolation. The external isolation springs recommended most often for sound and vibration sensitive applications have a **4" Static Deflections**. Springs with such high static deflections are not usually applicable for internal isolation. This is primarily due to the limited space available inside the equipment cabinet.
- 8.) For applications with external isolation only, any internal isolation springs on the fans should be "blocked out". The exceptions would be neoprene mounting grommets that are used on high speed components such as screw-type compressors, and fans with operating speeds over **1200 rpm**.
- 9.) If internal isolation is to be used rather than external isolation, the "blocking" used to immobilize the fan during shipment and installation of the rooftop unit must be removed.

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