Carlyle™

Recommendations to Minimize Refrigerant Line Vibrations

Application Data - Minimizing Line Vibration

Introduction

Refrigerant leaks have always been a major concern in any air conditioning or refrigeration system. Costs associated with refrigerant replacement and the reliability of the unit have given system designers and manufacturers the incentive to keep these refrigerant losses to a minimum. Additionally, the environmental effects have created additional concerns.

The following bulletin will assist equipment manufacturers in the design of refrigeration and air conditioning systems to minimize refrigerant line vibration problems. Vibration problems can often result in broken refrigerant lines. Causes of vibration in discharge lines can be primarily separated into the following categories:

1. Structural Resonances
2. Forced Vibration
3. Acoustical Resonances

Of the above causes of vibration, structural resonances are the most common cause followed by forced vibration and acoustical resonances. Vibration can also be caused by multiple combinations of the above. This guide will address each of these potential causes for both new and retrofit refrigerant piping systems.

Structural Resonance

Structural resonances occur when the natural frequency of the discharge piping matches the frequency of the gas pulsations in the piping (excitation frequency). When the excitation frequency is close to the natural frequency of the piping, the vibration of the piping becomes greatly amplified.

Imagine a tuning fork, it has a specific frequency that it resonates at, and that is where the most vibration occurs. Thinking of a piping system as a very large tuning fork, it would have to be designed so that it did not have a structural resonance at or near the exciting frequency of the gas pulsations. The natural frequency is a function of the stiffness and mass of the line.

A piping run should be designed as stiff and light as possible to reduce the chance of vibration problems. However, in systems where start and stop kicks or other system motion occurs, flexibility must be designed into the piping system to accommodate the motion. A stiff and light line will have a higher frequency at which it resonates than one that is less stiff and/ or has more mass.

In reciprocating compressors, the gas pulsations occur at relatively low frequencies, therefore systems with higher natural frequencies will have less chance of being affected by structural resonances. In screw compressors, the gas pulsations are generally at higher frequencies which increase the risk of exciting a structural resonance. The use of variable speed drives with compressor further complicate the situation as they lead to a much wider spectrum of excitation frequencies that must be considered.

These structural resonances are notoriously difficult to predict. A thorough evaluation a piping design in a laboratory environment is the best method for identifying and mitigating structural resonances. When a design is found

to be free of structural resonances, the exact design should be used consistently in the future. Any changes to a structural resonance free design, such as moving a fitting, should lead to fully requalifying the design.

Forced Vibration

Forced vibration is the vibration caused by movement of the compressor (mechanical imbalance) and/or discharge gas pulsations (pulsation driven vibration) emitting from the compressor. The term forced vibration excludes vibration due to any piping system resonances. Compressor driven vibration is not a typical problem when the compressor is solid mounted, but when the compressor is soft mounted the chances of vibration problems are greater.

Spring or rubber mounting kits are typically used in environments where vibration transmitted from the compressor to the system frame may cause excessive noise or damage to a sensitive area. When spring or rubber mounting kits are used, the piping system (which is where most vibration problems occur) must be designed to absorb start and stop kicks and handle the additional motion of the compressor during operation. This is usually accomplished by using flexible piping (vibration absorbers) and spring piping hangers.

Although these items may reduce the transmission of movement of the compressor to the system, they may greatly enhance the effects of piping system vibration. The chances of pulsation driven vibration and structural resonances increase with the introduction of springs into the system. Spring or rubber mounting of compressors is sometimes necessary depending on environment and/ or rack or unit design. Spring mounting generally makes piping geometry more complex and increases the possibility of vibration problems. Generally, if there is no specific reason for spring mounting compressors then don't.

On solid mount applications utilizing flexible piping (vibration absorbers) the inlet and outlet of the vibration absorber should be rigidly clamped. Allowing one or both ends of the vibration absorber to flex can increase the chance and/ or magnitude of vibration problems. Carlyle doesn't recommend the use of vibration absorbers in solid mounted compressor systems unless they are securely clamped at both ends.

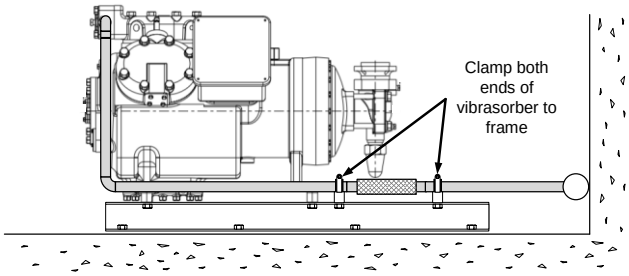


Figure 1 – Rigid Mounted Compressor & Vibrasorber

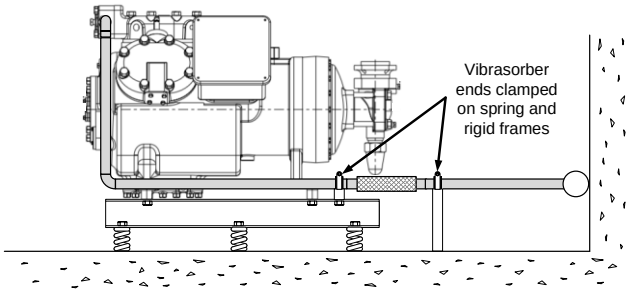


Figure 2 – Soft Mounted Compressor & Vibrasorber

On spring mounted applications, vibration absorbers should be used only in accordance with manufacturer's requirements and recommendations. Typically, vibration absorbers should be parallel to the crankshaft with the inlet clamped to the compressor or spring mounted base and the outlet clamped to the solid mounted frame. When piping spring mounted compressors, try to arrange the compressor and piping using as few bends as possible. The previous statement holds true on solid mount systems as well, but spring mounted systems usually become more of a problem because the discharge line vibration is most

effectively mitigated when the absorber is parallel to the crankshaft. Careful planning can eliminate excessive piping and bends.

Gas pulsation driven vibration is the most common cause of forced vibration. Pulsation driven vibration does NOT mean that the compressor is emitting such high pulsations that it forces the line to vibrate regardless of the piping geometry. All reciprocating compressors emit discharge gas pulsations (a reciprocating compressor generates a constant stream of pulsating flow). When discharge gas pulsations react with the piping system geometry in such a way as to set up an oscillating force, discharge pipe vibration may occur. An example of this is when the discharge line comes off the compressor service valve and enters 1, 2, 3 or more elbows. Picture the pulsating discharge gas flowing from the compressor through the first straight section of discharge pipe. The discharge gas then hits the first elbow and bounces into the next section of straight pipe. An oscillation in the gas has already started and each elbow may increase the oscillation creating a significant amount of line vibration. Designing the discharge piping as straight as possible will reduce the chances of pulsation driven vibration occurring. Another cause of forced vibration is the "Bourdon Tube Effect When discharge piping forms a U shape, the high pressure discharge gas tries to straighten the piping, resulting in vibration. Gas pulsations have little if any effect on discharge lines composed of a single straight pipe.

Acoustic Resonance

Acoustic resonances result from the specific discharge gas properties and the piping system geometry (not the structural dynamics). The effect of acoustical resonances is to amplify the gas pulsations at specific location in such a magnitude as to cause significant vibration. There are many types of acoustic resonances and most are very difficult to predict. The most common type results from longitudinal standing wave patterns.

Critical Pipe Length

A longitudinal standing wave will occur when the length of any piping run matches a critical wavelength of the gas pulsations. These critical length calculations are a function of the speed of sound of the refrigerant gas and the gas pulsation frequency, following this equation:

$$L_{critical} = \frac{C}{2f}$$

Equation 1 – Critical Length for Semi-hermetic Fixed Speed Compressors

Where;

$L_{critical}$	is the first critical piping length
C	is the speed of sound of the refrigerant gas Note that this will vary for different gases and based on the specific operating pressure and temperature of the gas.
f	is the excitation frequency based on the compressor model See Table 1.

Once this first critical length is calculated, then the piping system should be checked for any piping runs that are within ~10% of this length or any integer multiple (i.e. 1X, 2X, 3X etc) of it. Vibrations at higher multiples (4X, 5X, etc) are theoretically possible but unlikely.

Figures A1 through A11 are provided at the end of this guide for the speed of sound data of several different refrigerant gases. Contact Carlyle Application Engineering for other gases that are not provided here.

Excitation frequencies for the Carlyle compressors in fixed speed applications are shown in Tables 3-5.

Compressor	Excitation Frequencies	
	60Hz Fixed Speed	50Hz Fixed Speed
06D 2cyl 05K 2 cyl 06CC All models	58Hz	48Hz
06D 4 cyl 06E 4 cyl 05K 4 cyl	117Hz	97Hz
06D 6 cyl 06E 6 cyl 05G 6 cyl	58 & 117Hz	48 & 87Hz
06M	88Hz	73Hz
06TS/TT/TU/TV	296Hz	247Hz
06TA/TR 033cfm	440Hz	366Hz
06TA/TR 039cfm	525Hz	438Hz
06TA/TR 044cfm	589Hz	491Hz
06TA/TR 048cfm	638Hz	532Hz
06TA/TR 054cfm	725Hz	604Hz
06TA/TR 065cfm	892Hz	743Hz
06TA/TR 078cfm	1065Hz	888Hz
06TA/TR 088cfm	1194Hz	995Hz
06TA/TR 108cfm		1226Hz

Table 1 – Excitation for Semi-hermetic Fixed Speed Compressors

The critical length equation can be rewritten for open drive compressors this equation:

$$L_{critical} = \frac{C}{N \times \left({}^{RPM}/_{30}\right)}$$

Equation 2 – Critical Length for Open-Drive Fixed Speed Compressors

Where;

N Number of compressor cylinders

RPM is the shaft speed

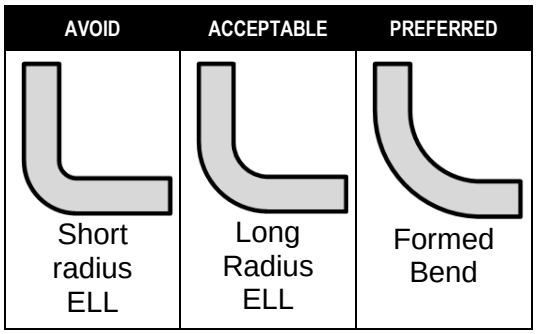
Piping Recommendations

Through the utilization of good design practices forced vibration, and acoustical and structural resonance problems can be kept at a minimum. There is no design method that will eliminate all possible vibration problems 100% of the time except through experimental testing of each system. From the previous material, it is shown that predicting vibration problems can be very difficult and complex. It should be noted that when a system is designed and found to be free of vibration problems, that design should be replicated in its entirety.

NOTE: Vibration problems may be created by any changes to the design i.e., different dimensions, additional fittings, different mufflers. In conclusion if you find a design that works, keep it.

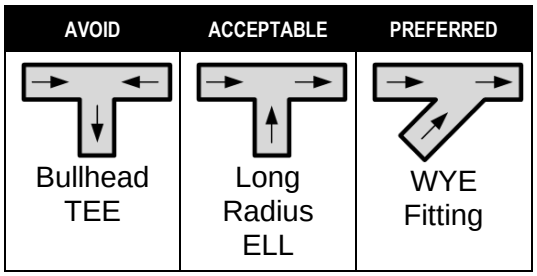
The following is a list of good design practices:

1. Keep line as stiff and light as possible. Keep in mind that the line may require inherent flexibility to handle any system motion (start and stop kicks ...).
2. Keep lines as straight as possible. Use formed discharge lines in place of elbows wherever possible. If elbows must be used, use long radius in place of short radius or square elbows. Use 45-degree elbows or bends in place of 90's wherever possible.



Formed bends with a bend radius that is greater than 10 pipe diameters has very low pressure drop and acoustically behaves like a straight section of piping.

3. Use Y's or a 45-degree bend in place of T's wherever possible. Use fittings of the proper size, don't reduce line size for fittings. Do not use bullhead tee fittings to combine two discharge gas flows.



4. Use largest practical discharge line diameters possible. Practically speaking, if a line is sized for a given load and proper oil entrainment, there is usually no significant increase in line size possible.
5. Determine whether spring or solid mounting is appropriate.
6. When using solid mount compressors with vibration isolators, clamp the isolator at both ends as shown in Figure 1. If using a soft mount then the vibration isolator should be clamped one end on the spring platform and the other end to the rigid base as shown in Figure 2.
7. If natural frequencies have been occurring in existing designs, redesign piping system.
8. System geometry should be checked by calculating standing wave resonance.
9. Carlyle compressors are supplied without valves to allow flexibility in sizing refrigerant lines. Always calculate both suction and discharge line sizes and use the appropriate service valves. Although we recommend using the most generous discharge line diameters practical, please note that oversizing lines can result in oil return problems. Refrigerant lines should always be sized to handle not only the specific design condition, but the maximum and minimum load conditions as well.
10. When using mufflers, place them as close to the service valve as possible and/ or before the first fitting. Ideal placement of the muffler is not always possible but note that a muffler is ineffective for any upstream piping.

Existing Systems

To eliminate vibration problems in existing system, the cause of the vibration should first be determined. Structural resonances are the most common problem, they can usually be detected and corrected by the addition of mass to the line. This may sound contradictory to the statement that lines should be designed as light as possible. When a system has a structural resonance it is because the piping has a natural frequency at the same range as the discharge gas pulsations. The easiest way to eliminate this problem is to add mass to the discharge line which can effectively lower the natural frequency of the piping. This is a very effective method of tuning the system out of the range of the gas pulsations frequency. The addition of mass can be in the form of a muffler and in some cases an existing muffler can be the culprit in the system. For example, if your system is designed so that the natural frequency of the line is greater than that of the gas pulsations, the addition of a muffler may add sufficient mass to drop the natural frequency of the discharge line to the range of the gas pulsations. The addition of mass could also be in the form of a plastic molded weight that could be slid up and down the discharge line. Moving the mass around the piping to find its most effective position, can be a very effective means of tuning a system. If a weight is used in a system, make sure it is rigidly attached to the refrigerant line. If a weight can move independent of the line it may cause wear and potentially a leak. Clamps may also be added to increase the stiffness of the line and are very effective. The use of clamps is discussed later in this section.

Forced vibration is the next most common cause and can be corrected by using mufflers, baffle plates, larger discharge lines, reduction of bends, long radius elbows, formed discharge lines, and clamps. To determine if a muffler will be effective, throttle the discharge valve for approximately 20psi increase in discharge pressure. The pressure increase does not occur until the valve is almost completely front seated

WARNING!
Do not completely front seat valve. This will cause extreme internal compressor pressure which may cause compressor damage and potentially rupture the compressor.

If the piping vibration subsides with the throttling of the discharge valve, a muffler will usually be effective in mitigating the issue.

Forced vibration is generally due to gas pulsations, but in rare instances may be due to compressor imbalance. Motor or coupling problems can also be the cause of forced vibration in open drive compressors. To determine whether the vibration is due to compressor imbalance or gas pulsations the following method can be used on reciprocating compressors.

WARNING!

This method should be used ONLY for reciprocating compressors.

Screw compressors should never be run on air.

1. Tum compressor off.
2. Shut the suction and discharge service valves.
3. Bleed all pressure from compressor.
4. Remove both service valves.
5. Secure a rag over the discharge port to separate oil mist.

WARNING!

Oil mist from the reciprocating compressor can be extremely flammable. Be sure to use the oil separation method in step #5. Make sure both service valves are removed from compressor.

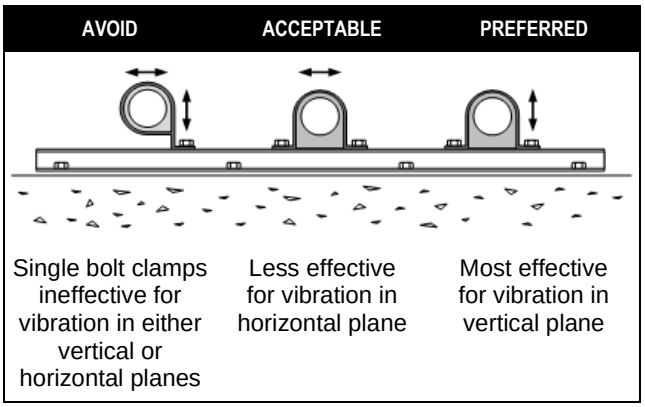
6. Run the compressor for a short period of time (10 to 15 seconds) on air.
7. During run time examine compressor for imbalance

This method quickly determines imbalance problems on semihermetic compressors, but on open-drive compressors it has only narrowed it down. On open-drive compressors, if imbalance occurs after running the compressor on air, check the coupling alignment and then run the motor by itself. When using the above methods, take into account any start and stop kicks due to the mounting method of a weak base.

Clamping

Clamping lines can be very effective and is usually the easiest method to correct vibration problems. Muffler and baffle plates are another means for dealing with pulsation driven vibration. The pros and cons of muffler versus baffle plates will be discussed later in this guide.

Clamps are most effective if placed in the direction of maximum motion. If a line is moving horizontally and you clamp it vertically the clamp will be less effective. When clamping lines, they must be clamped to something that is more rigid than the line itself.



Clamping a discharge line to a light piece of channel may cause the channel to resonate. If the channel begins to resonate it may cause noise and possibly transmit vibration to other system components.

The discharge piping system geometry should be checked for any acoustical resonances using the calculations shown previously.

Mufflers

Mufflers can reduce discharge gas pulsations and effectively eliminate vibration problems downstream of them. In most cases, mufflers are ineffective at low frequencies, this primarily depends on the muffler itself. Carlyle offers mufflers to reduce discharge gas pulsations. Muffler placement is critical to its effectiveness,

it should be placed as close to the compressor as possible, but the previously described critical pipe length must be avoided.

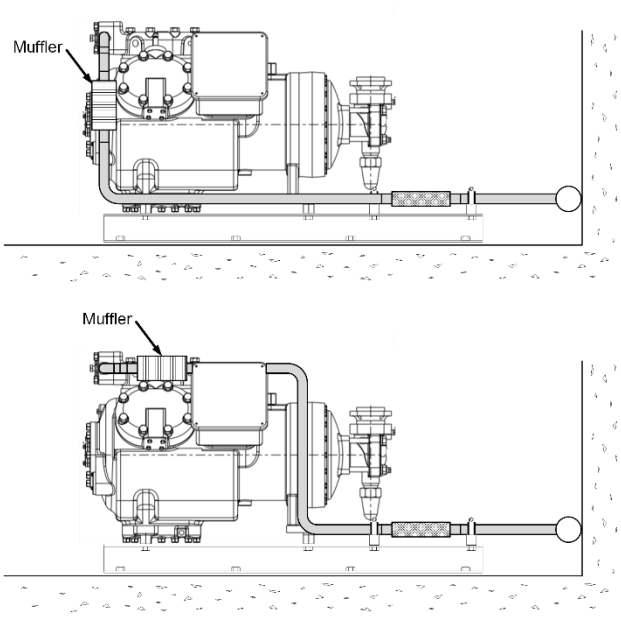


Figure 3 – Recommended Muffler Locations close to Discharge Service Valve

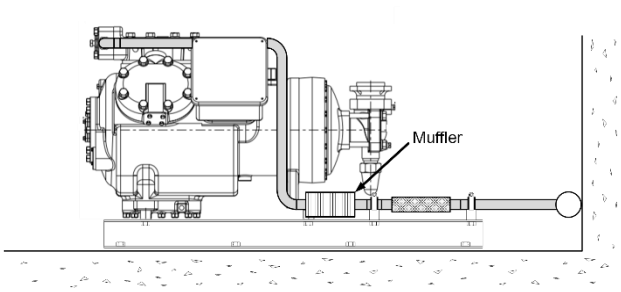


Figure 4 – Less Effective Muffler Location Not Recommended

Mufflers are not typically needed in system designs. However, positive displacement compressors do emit gas pulsations and sometimes these pulsations can interact with specific system designs to cause vibration. In these systems where pulsation driven vibration has proven to be a problem, a properly selected muffler can mitigate the issue.

In addition to vibration reduction, mufflers can be used to effectively reduce piping system noise. Lengthy piping runs can radiate audible noise. The use of mufflers in systems with remote condensers has proven very effective at reducing piping noise levels.

Mufflers should be used on all of the following applications:

1. All 06E compressors with capacity control.
2. All non-unloading 06EM and 06ER 99cfm compressors.
3. All 06TA/06TR screw compressors.
4. All 5H46, 66, 86 and 126 compressors. Carlyle also recommends their use on the 5H40, 41, 60, 61, 80, 81, 120 and 121 models.

Baffle Plates

Baffle plates are basically an orifice used to create a pressure drop. This pressure drop will attenuate discharge gas pulsations. Baffle plates can be very effective at low frequencies, where mufflers offer limited attenuation. The use of baffle plates will increase the discharge temperature, but for most applications the amount is insignificant. Baffle plates are also very easily installed in a system. The orifice on a baffle plate is designed to reduce discharge gas pulsation amplitude at full load, but when unloaders are used the baffle plates effectiveness becomes very limited. In effect a baffle plate can eliminate vibration problems at full load, but at part load vibration problems may appear because of the lower mass flows.

NOTE
Baffle plates are designed to create a 6-10 psi pressure drop.

Appendix A – Speed of Sound Figures

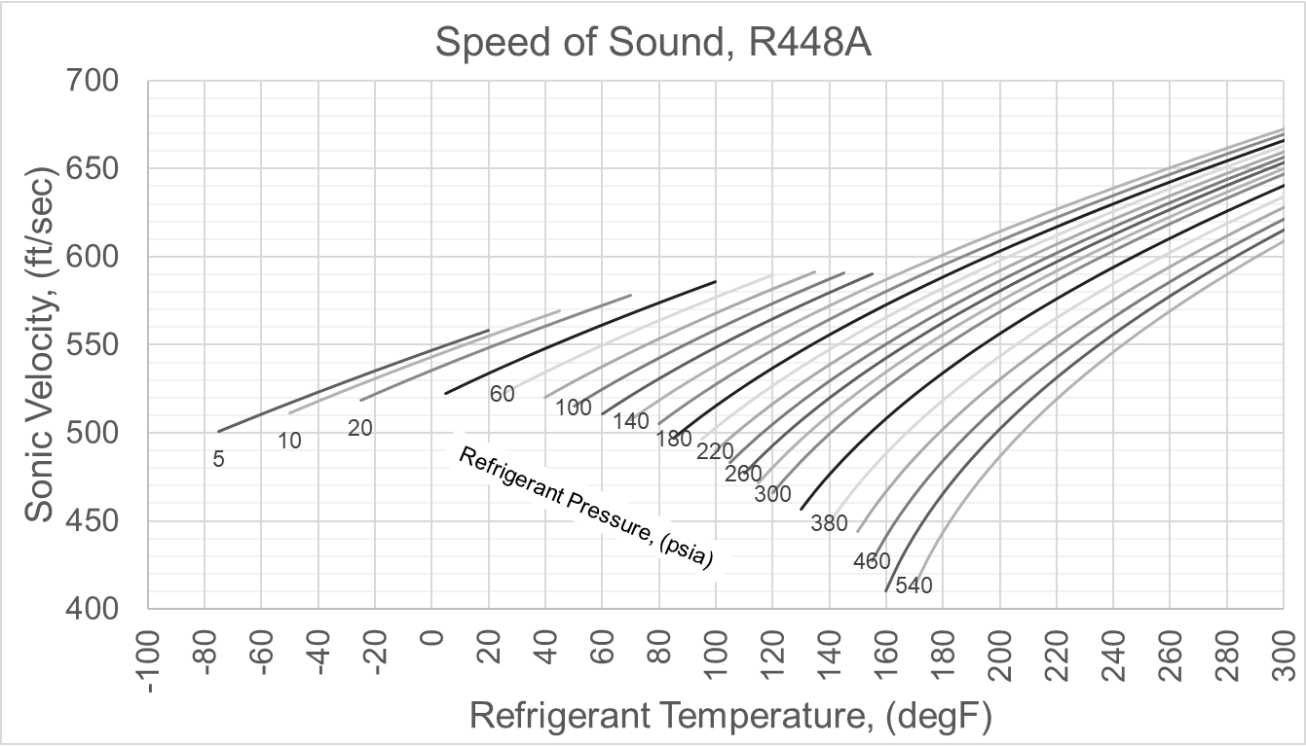


Figure A1 – Speed of Sound R448A

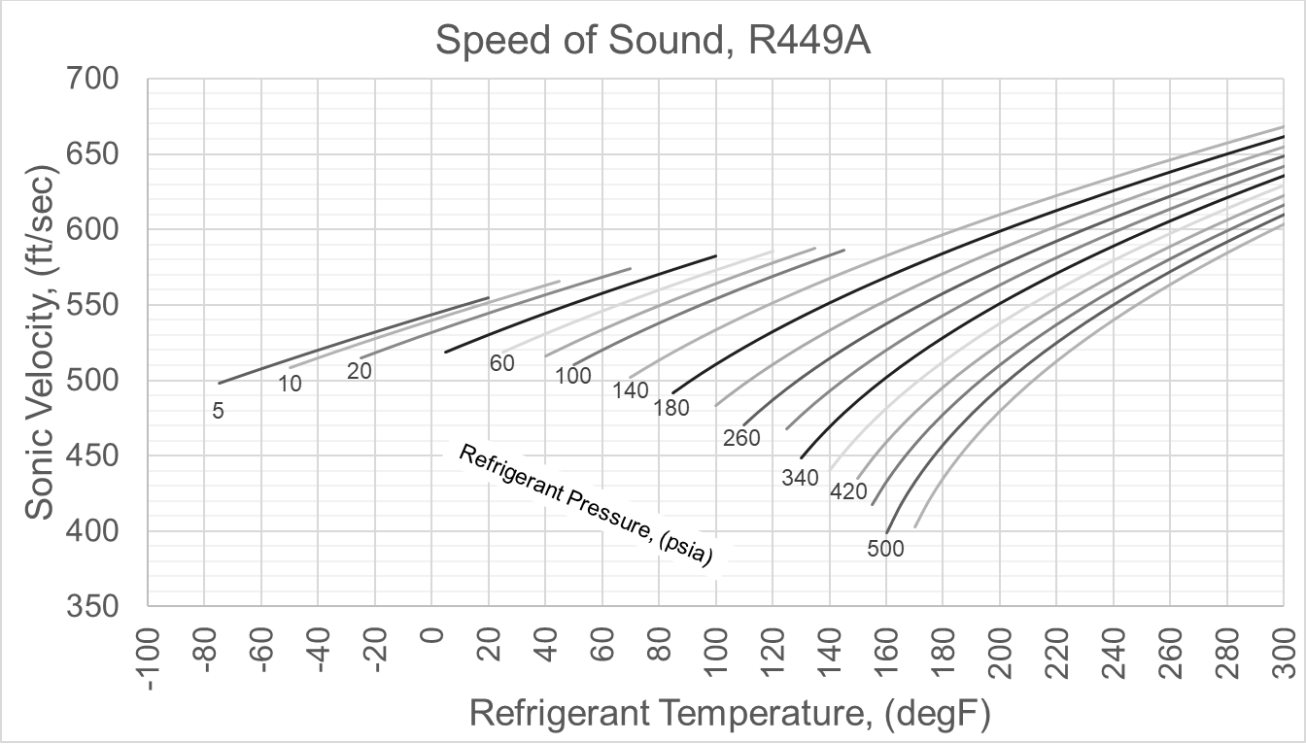


Figure A2 – Speed of Sound R449A

Appendix A – Speed of Sound Figures

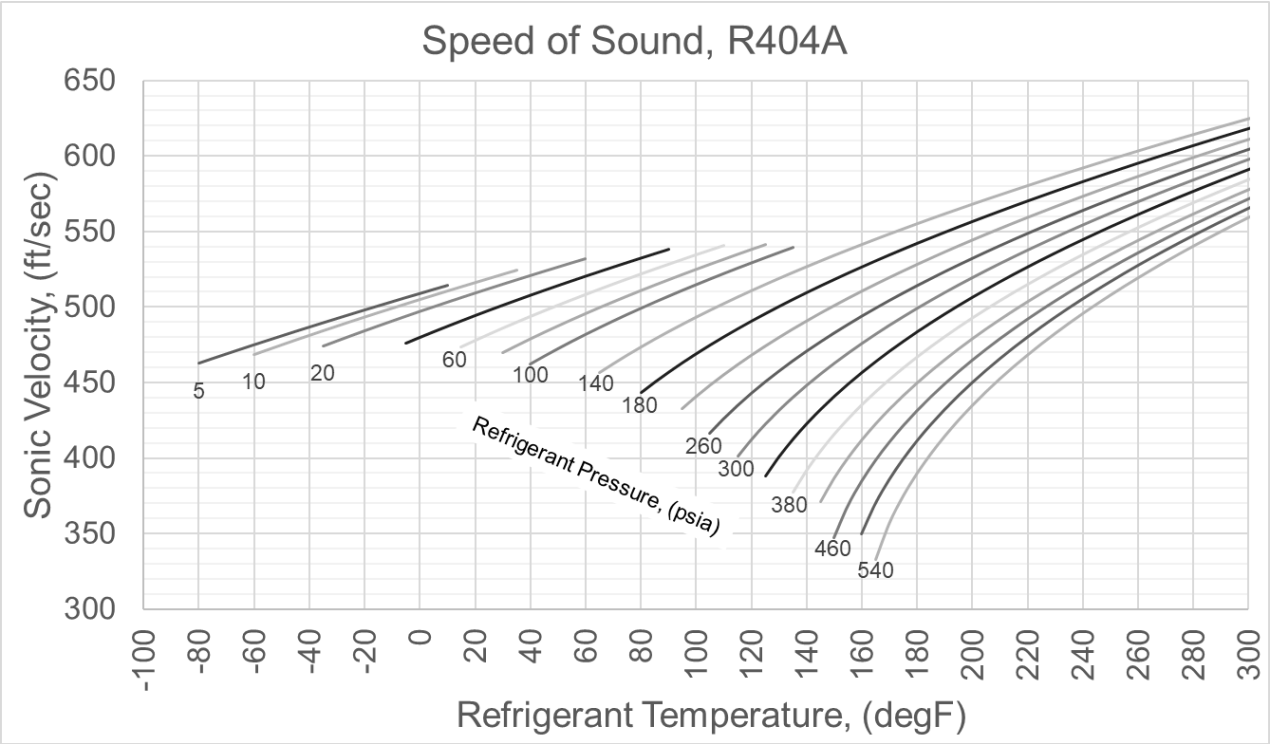


Figure A3 – Speed of Sound R404A

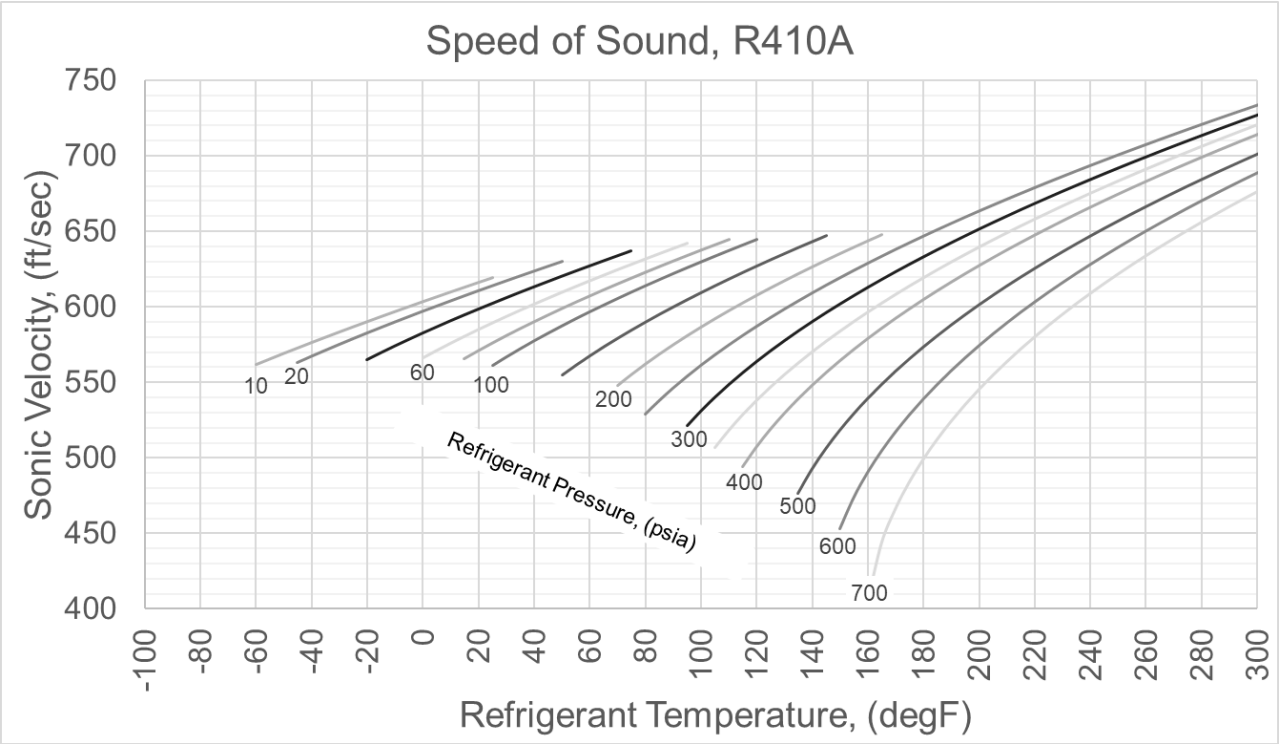


Figure A4 – Speed of Sound R410A

Appendix A – Speed of Sound Figures

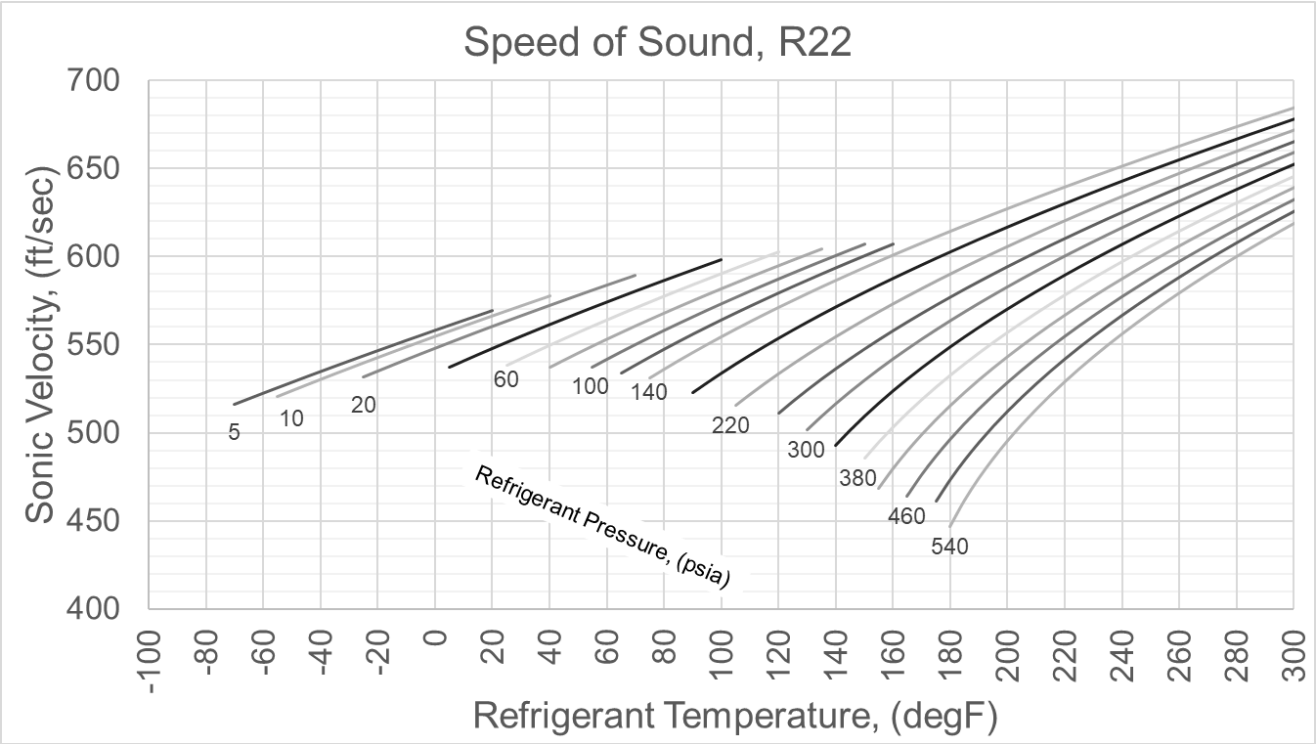


Figure A5 – Speed of Sound R22

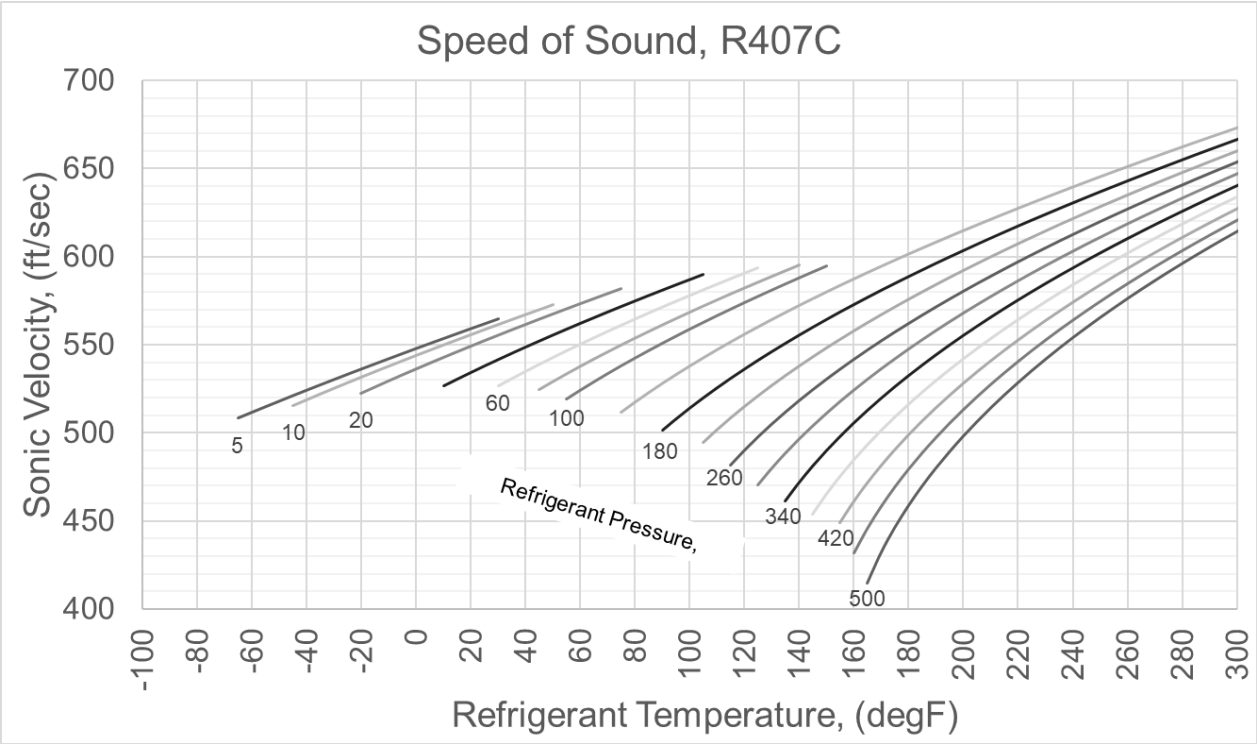


Figure A6 – Speed of Sound R407C

Appendix A – Speed of Sound Figures

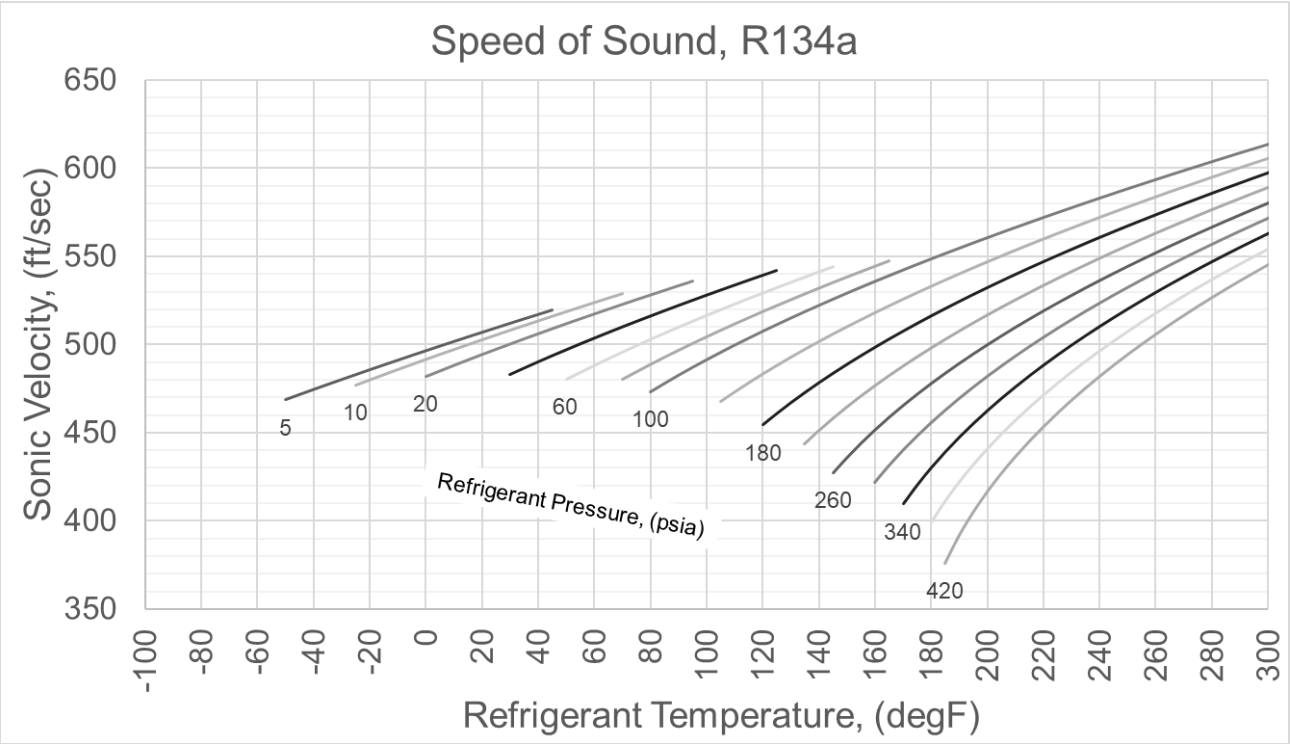


Figure A7 – Speed of Sound R134a

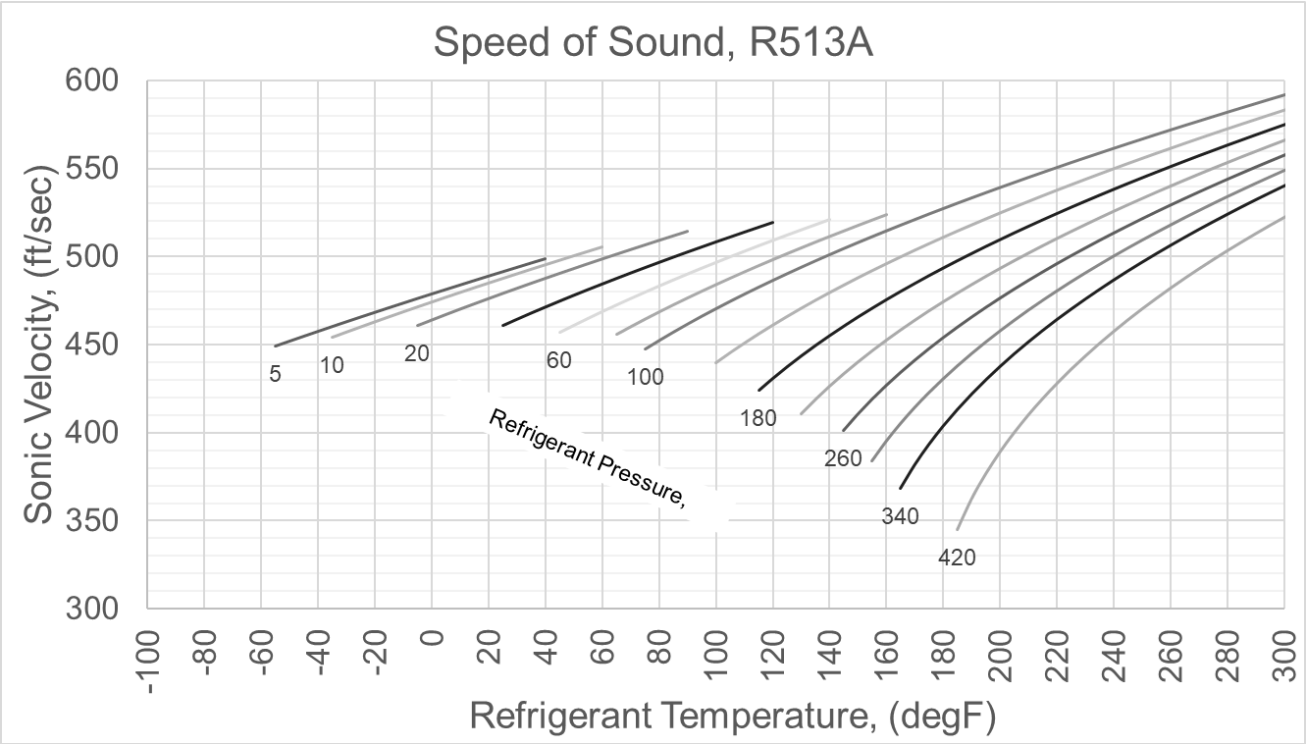


Figure A8 – Speed of Sound R513A

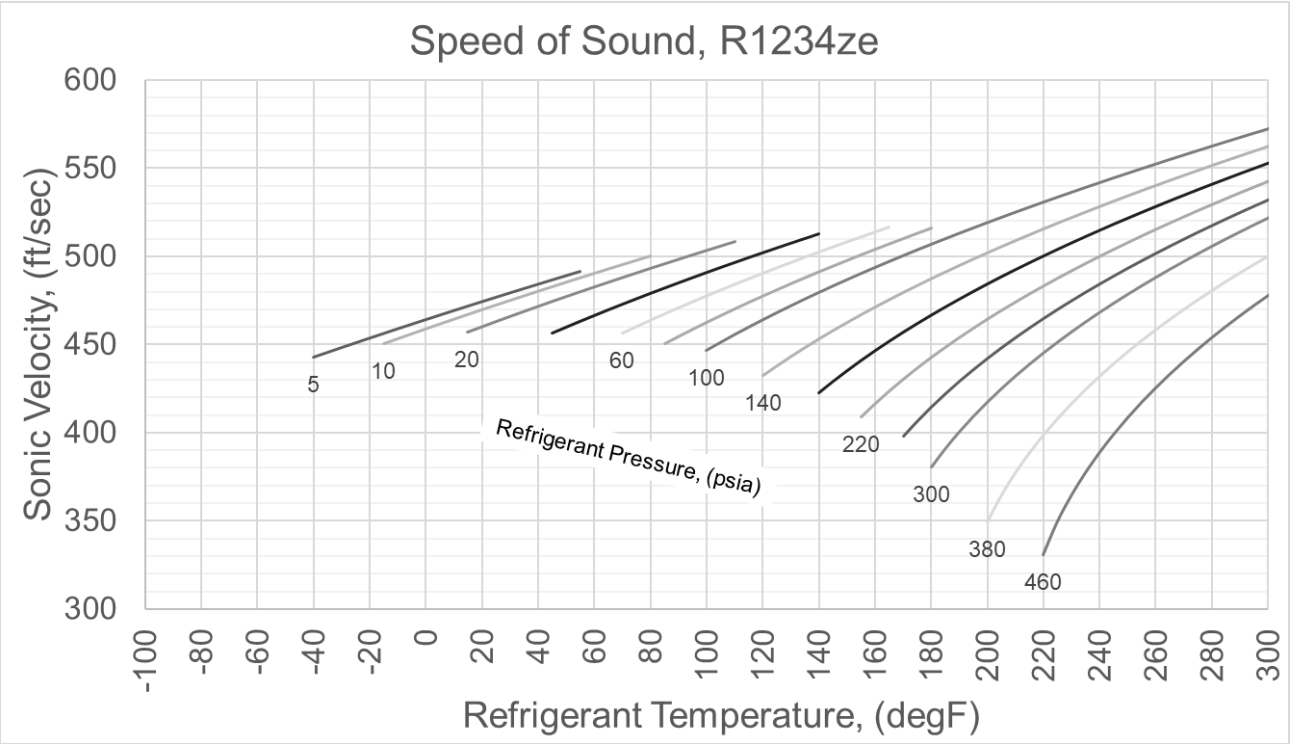


Figure A9 – Speed of Sound R1234ze

Appendix A – Speed of Sound Figures

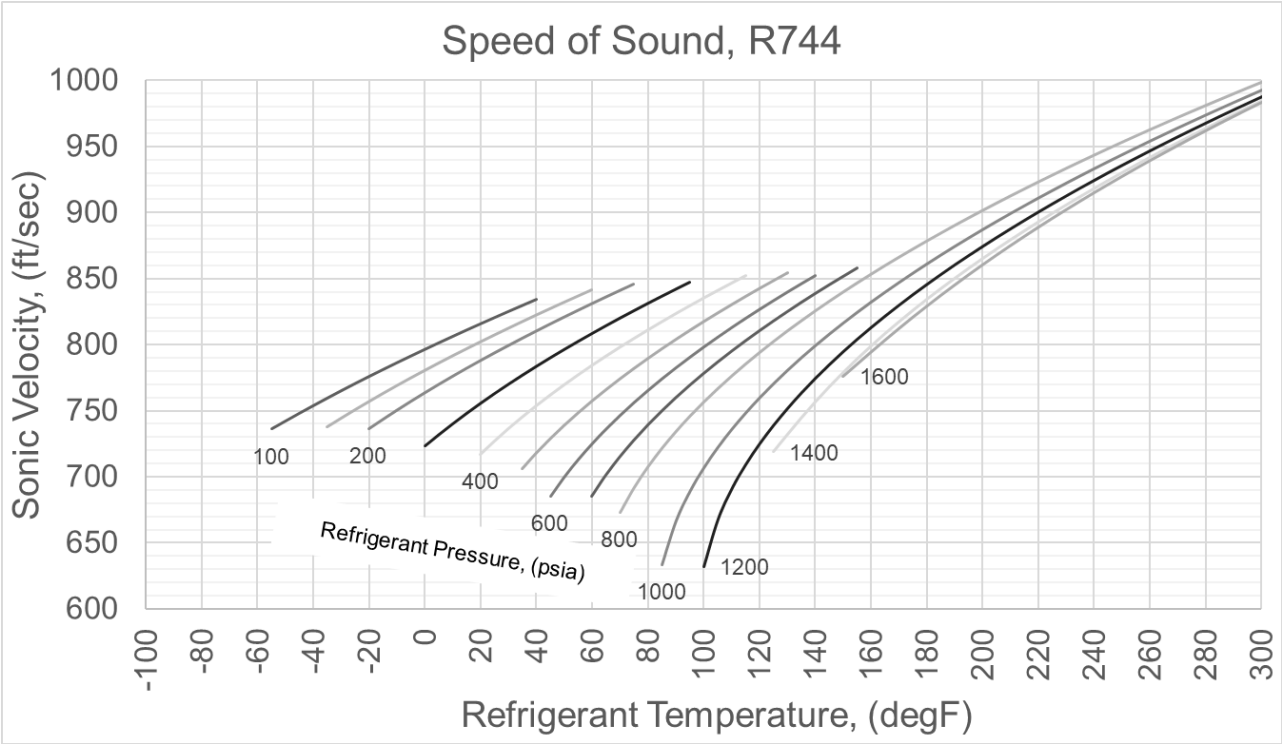


Figure A10 – Speed of Sound R744, CO_2

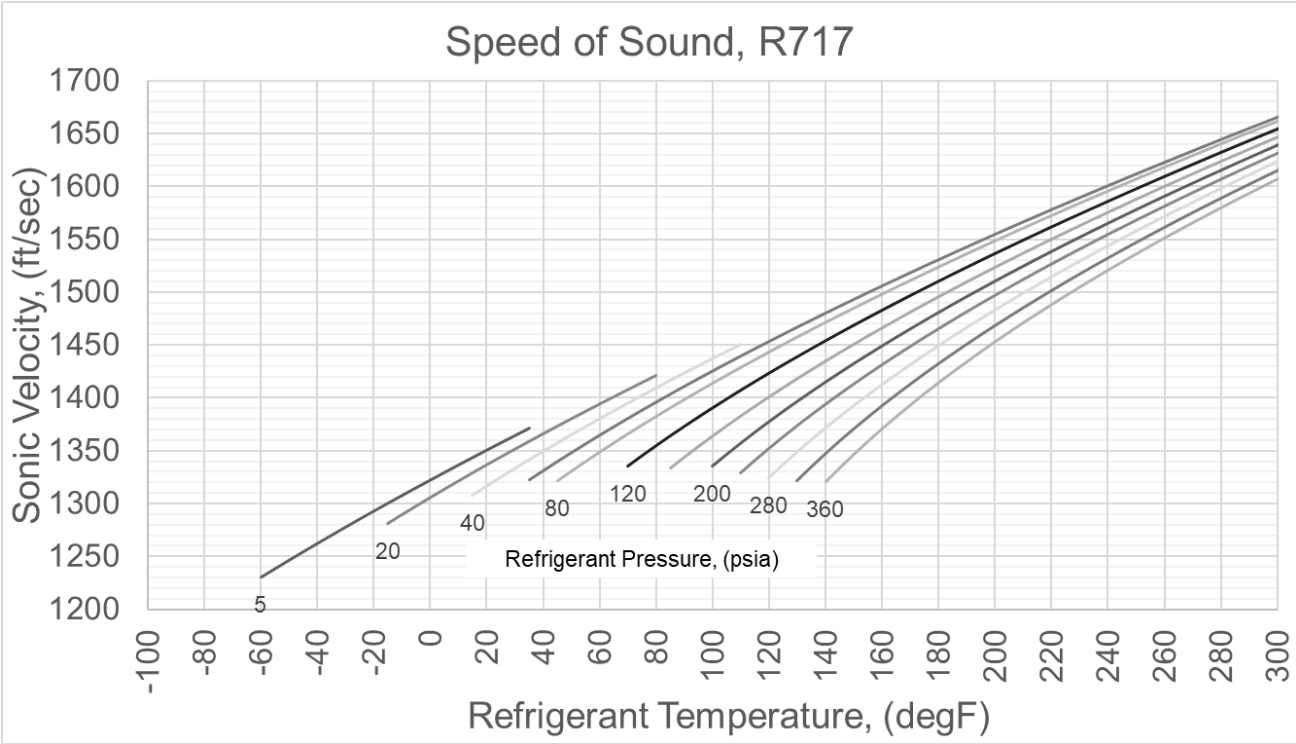


Figure A11 – Speed of Sound R717, Ammonia