SURFACE VEHICLE INFORMATION REPORT	J2914™	NOV2022
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Exhaust Gas Recirculation (EGR) Cooler Nomenclature and Application

RATIONALE

Five-Year Review.

1. SCOPE

This document provides an overview on how and why EGR coolers are utilized, defines commonly used nomenclature, discusses design issues and trade-offs, and identifies common failure modes. The reintroduction of selectively cooled exhaust gas into the combustion chamber is just one component of the emission control strategy for internal combustion (IC) engines, both diesel and gasoline, and is useful in reducing exhaust port emission of nitrogen oxides (NOx). Other means of reducing NOx exhaust port emissions are briefly mentioned, but beyond the scope of this document.

2. REFERENCES

2.1 Related Publications

The following publications are provided for information purposes only and are not a required part of this SAE Information Report.

2.1.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or +1 724-776-4970 (outside USA), www.sae.org.

SAE J922 Turbocharger Nomenclature and Terminology

SAE J1726 Charge Air Cooler Internal Cleanliness, Leakage, and Nomenclature

SAE J1994 Laboratory Testing of Vehicle and Industrial Heat Exchangers for Heat Transfer and Pressure Drop Performance

3. SYSTEM ARCHITECTURES FOR EXHAUST GAS RECIRCULATION (EGR) COOLER APPLICATION

3.1 EGR in Internal Combustion Engines

EGR is a combustion strategy of scavenging some of the exhaust gas, cooling it, and then mixing it with fresh charge air, before entering the cylinder to achieve a target inlet manifold air temperature (IMT). The two effects are an increase in specific heat of the charge air mixture entering the cylinder, and a decrease in maximum flame temperature due to lower oxygen content. For a given fuel energy released by the burned fuel, the peak combustion gas temperature in the cylinder is reduced with the desired effect of reducing NOx output. The higher the percentage mass of EGR, the lower the peak combustion gas temperature, and the lower the NOx produced. Since the heat removed from the EGR flow is transferred into the engine coolant, the downside is a corresponding increase in jacket water (JW) heat rejection and required external cooling system capacity.

3.1.1 Implications on Engine Design

The addition of EGR into combustion intake flow requires a larger pressure differential across the cylinder to force that mass flow. This differential is not significant in engine air system design in IC engines at lower power density or brake mean effective pressure (BMEP). But for higher BMEP ratings, especially heavy duty (HD) diesels with turbochargers, higher pressure ratio turbo charging is required than without EGR flow. Increased EGR heat rejection often requires an increase in JW pump capacity to maintain the same desired ΔT across the radiator at design point heat load.

All the following applications of EGR coolers are illustrated with turbocharged EGR air system architectures, and are shown with air to air coolers (CACs) in the external cooling system. Only the charge air cooling (CAC) related system components are shown. For simplified illustration, other heat exchangers in the coolant circuits (JW or low temperature circuits) are not shown, nor are components of the external cooling system (fans, radiators, etc.).

3.2 EGR System Architecture Types

3.2.1 Low Pressure Loop EGR

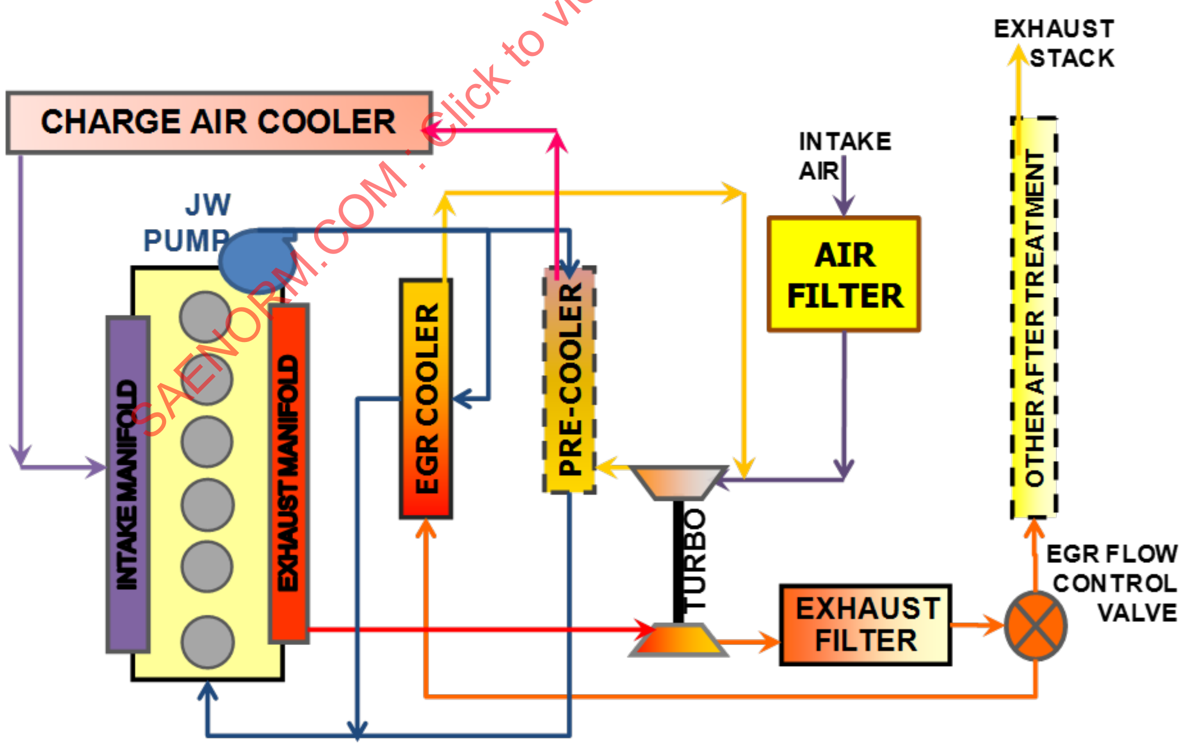

Figure 1 - Low pressure loop EGR schematic

Low pressure loop (LPL) EGR is differentiated by its source of exhaust gas taken from the downstream low pressure side of the turbocharger. Less thermal energy removal (cooling system heat load) is required to reach the desired IMT, since the exhaust gas has already lost some energy after exiting the turbine. Although not required, the gas may also be drawn downstream of the exhaust particulate filter (as shown in Figure 1), resulting in less particulate matter entering the cylinder before combustion. Both have the added benefit of even lower temperature exhaust to be cooled. The pre-cooler shown in both the LPL and high pressure loop (HPL) figures is optional, and only used if high enough pressure ratio turbo charging requires cooling of the charge air ahead of the CAC to stay below material temperature fatigue limits of the CAC core.

Advantages of low pressure loop EGR architecture:

1. The biggest advantage of LPL EGR after the exhaust filter (particulate matter (PM) trap), is that cleaned exhaust is reintroduced to the cylinder for combustion, with reduced vulnerability to piston, ring, and liner wear related to abrasive exhaust particles.

2. The risk of abrasive wear on the thin walled tubes of the EGR cooler and fouling of the wall surface are also reduced. If the exhaust gas is taken upstream of the PM trap then these two advantages disappear.

3. Because the exhaust gas enters the EGR cooler at a temperature lower than the following high pressure loop (HPL) configuration, the risk of boiling failure modes is decreased. Sufficient JW flow entering the cooler is still required to prevent boiling on the tubes and tube-header joints at the inlet, but to a lesser degree. Thermal cycle fatigue is still a major issue, and not considered a major reduction in risk with LPL architecture.

4. Turbo charger speed and efficiency is less affected by EGR rate than the HPL configuration.

5. Mixing of fresh air and EGR flow is very complete.

Disadvantages of low pressure loop EGR architecture:

1. A major disadvantage of the LPL configuration is that corrosive exhaust gas is exposed to various components downstream of the EGR cooler, e.g., mixer valve, compressor, and CAC. This creates more expensive material choices and/or coatings to prevent corrosion.

2. The higher humidity levels of the mixed gases (exhaust and fresh air) entering the CAC will create condensation. Under loaded conditions this frequently results in a corrosive liquid entering the cylinders which can have various detrimental effects. Under extended no load (idle) conditions, condensation can collect in the CAC since air velocities are not high enough to purge it. This can create a slugging (hydraulic lock) risk when the engine speed is subsequently increased.

3. Since the exhaust temperature enters the EGR cooler at a lower temperature than the HPL system, the EGR heat transfer surface area required is increased due to the lower entering temperature difference, all other variables being equal. This results in either very closely spaced tubes (adding coolant side restriction and the risk of localized boiling), or larger space required for the core matrix.

4. The addition of exhaust piping to feed the EGR cooler from a point downstream of the after treatment adds additional cost and space claim. Furthermore, the integrity of this piping must be maintained over the life of the vehicle since this is an emission related system. In some systems, the operating points may create a vacuum in this line which can draw contaminants into the combustion air system if a leak occurs. Furthermore, the additional piping may increase engine or vehicle enclosure temperatures depending on the configuration design.

3.2.2 High Pressure Loop EGR

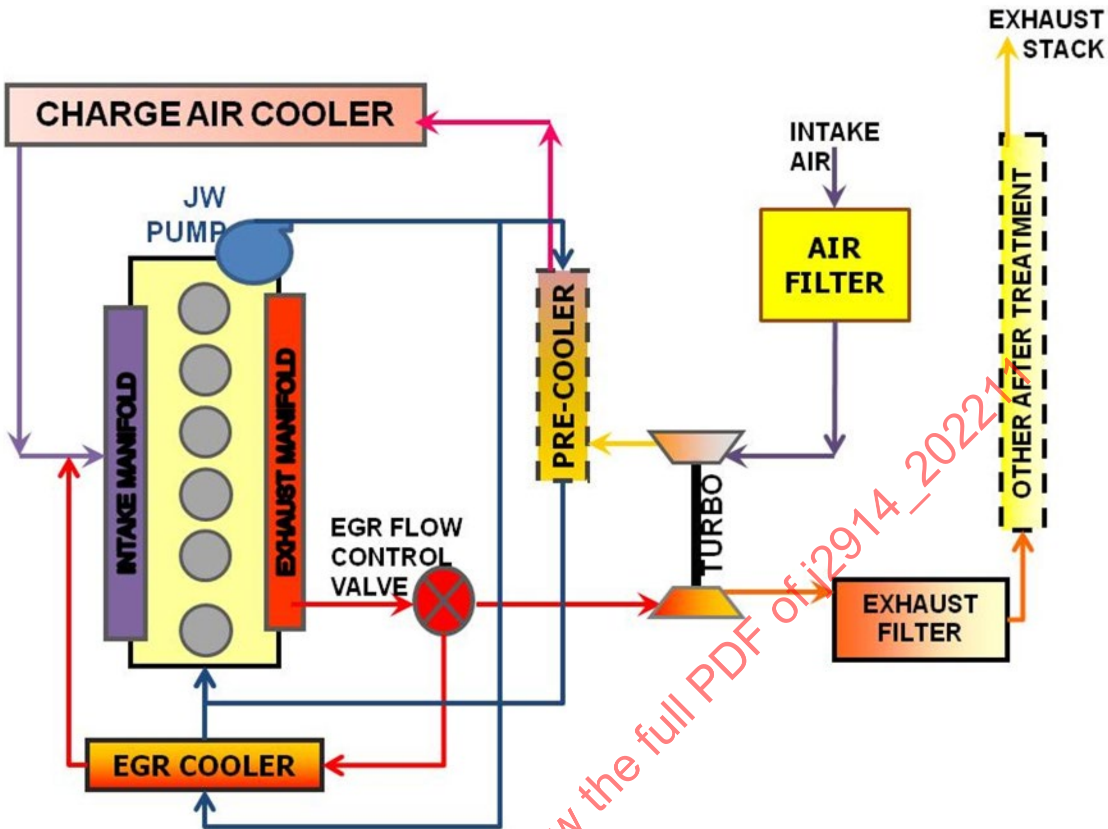

Figure 2 - High pressure loop ERG schematic

High pressure loop (HPL) EGR is differentiated by its source of exhaust gas taken from the exhaust manifold ahead of the turbocharger. More thermal energy removal is required to reach the desired IMT than LPL architecture which extracts exhaust gas further downstream at a cooler temperature. The cooled gas also contains all corrosive and abrasive particles before any after treatment is applied. The pre-cooler is again optional, depending on the charge air temperature from the compressor and the temperature limitations of the CAC material.

Advantages of high pressure loop EGR architecture:

1. The exhaust piping is simpler, with fewer connections, and associated reliability.

2. Since the exhaust temperature source is at its higher temperature, the heat transfer surface area is reduced to meet the cooler effectiveness requirement, everything else affecting EGR cooler size being equal.

Disadvantages of high pressure loop EGR architecture:

1. Because the exhaust gas enters the EGR cooler at higher temperature, the risk of boiling failure modes is increased. More JW flow is required to prevent boiling on the tubes and tube-header joint at the inlet. This drives increased flow-capacity required from the JW pump.

2. Thermal cycle fatigue within the EGR core matrix is also a greater risk with the higher differential between hot and cold fluid temperatures.

3. Because the exhaust gas contains abrasive particulate matter, risk of abrasive wear is increased inside the core matrix in areas of high velocity, which may drive material selection of the thin walled EGR cooler tubes or plates.

4. The presence of unfiltered particles in the cylinder intake stream occurs as the exhaust flow enters the cylinder, increasing piston, ring, and liner wear.

5. Fouling of the exhaust side passages will occur to a greater degree than the LPL filtered circuit. Fouling factor, or degradation from new, must be factored in to the design related to heat transfer performance as well as gas side pressure drop. Provisions for cleaning a fouled cooler are also a serviceability issue.

3.2.3 Two-Stage EGR Coolant Circuits

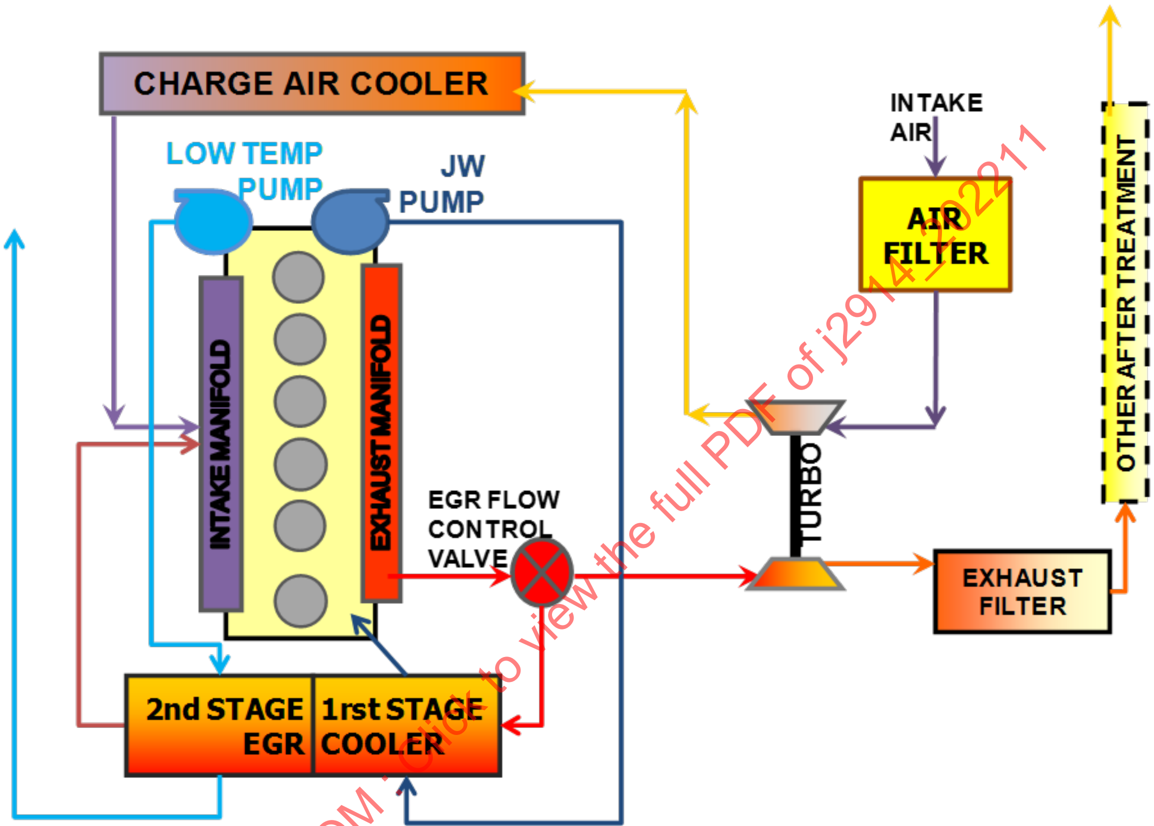

Figure 3 - Two-stage EGR schematic

This third variation illustrates a difference in the coolant side of the EGR architecture. It is applicable to both LPL and HPL configurations, and is independent of the presence of a charge air pre-cooler upstream of the CAC. The differentiating feature is that the EGR cooler is a single pass gas side design, but with two separate coolant circuits passing through the liquid side. The first stage cools the gas in the JW circuit, as in earlier diagrams. But a second, lower temperature coolant circuit, with temperature available below JW thermostat controlled temperature, is used to further lower the exhaust gas temperature, thereby increasing its density and decreasing exhaust port NOx.

Advantages of two-stage EGR cooling architecture:

1. The lower temperature of the exhaust at the intake manifold lowers the mixed IMT, and the increased density increases the ratio of EGR flow. Both reduce the NOx content at the exhaust port and reduce the need for further downstream after-treatment.

2. If the IMT goal is equal to the single stage configuration, the lower temperature coolant in the second stage increases the overall entering temperature difference ΔT available to the cooler, allowing less surface area and space required for a given heat load removal.

Disadvantages of two-stage EGR cooling architecture:

1. If the overall cooling system already includes a low temperature circuit with other auxiliary coolers, then there is a small penalty for increased flow capacity for the addition of the EGR cooler second stage. But if the EGR cooler is the only heat exchanger in the low temperature circuit, then the penalties for the addition of a second coolant pump, low temperature radiator, and associated water lines add cost, space requirements, weight, and reliability risk.

2. The design of the cooler itself is structurally more challenging since thermal gradients between high and low temperature sections of the cooler are higher than a single stage cooler.

Another form of two-stage EGR cooling utilizes a gas-liquid heat exchanger for the first stage, follow by a gas-air cooler for the second stage. Advantages and disadvantages are similar to those already mentioned. One additional disadvantage of this type of system would be the added cost and space for piping of exhaust gas between the two coolers. Pressure drop limitations on the exhaust gas side would drive up both pipe sizes as well as heat exchanger sizes. The alternative to large line sizes would be higher pressure ratio turbocharging.

3.2.4 Donor Cylinder EGR

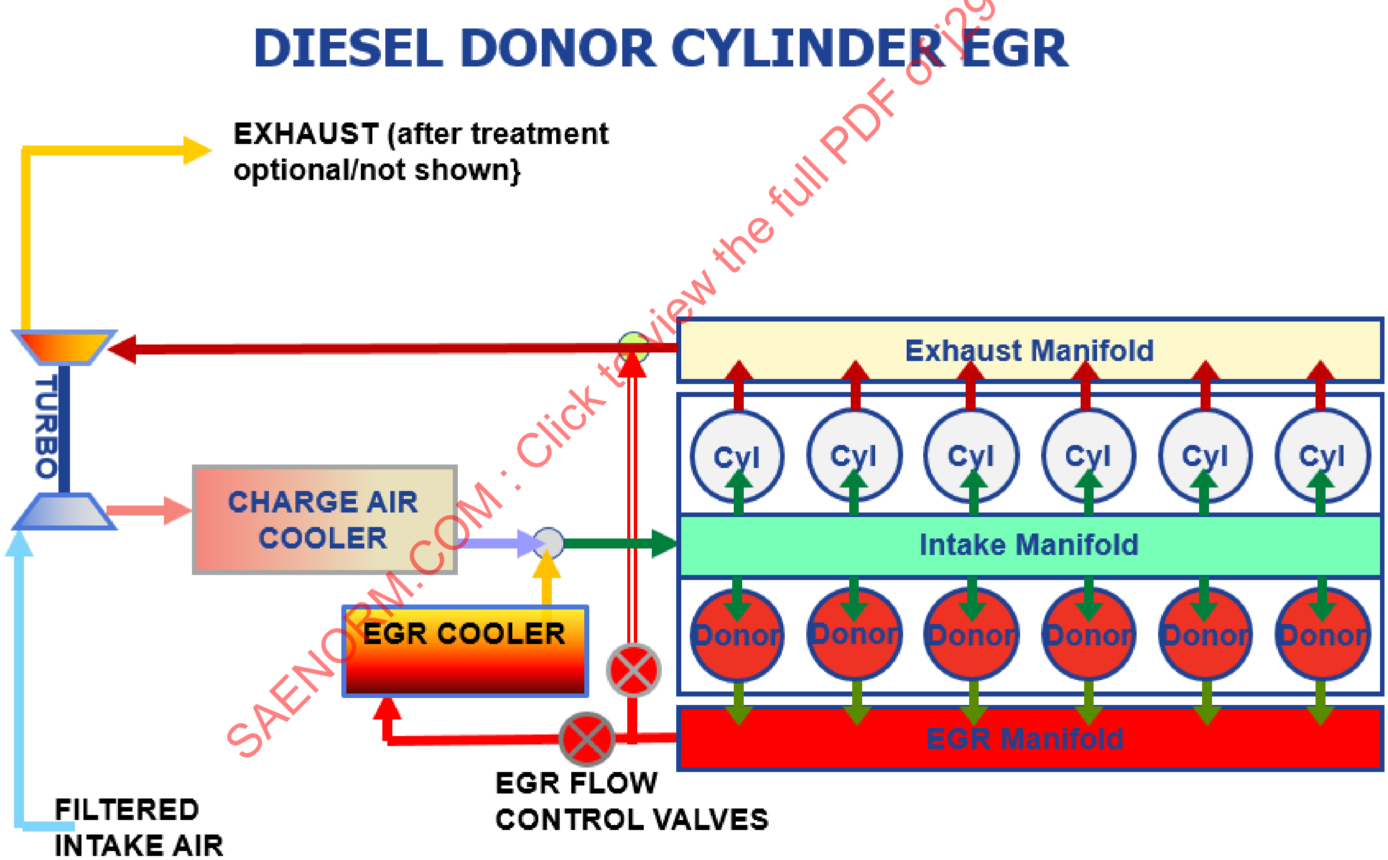

Figure 4 - Donor cylinder EGR schematic

The fourth variation of EGR architecture scavenges exhaust gas from only some of the cylinders. It is most advantageous in V engine configurations where there are already two separate exhaust manifolds. One exhaust manifold goes directly to the turbine side of the turbocharger, while the other manifold is the source of EGR flow. Two valves are required to control how much of the donor cylinder flow returns directly to the turbocharger, and how much flow is directed to the EGR cooler. This system does not require changes to the water system schematic described in the earlier systems.

Advantages of donor cylinder EGR architecture:

1. Fuel consumption can be reduced due to lower pumping losses on non-donor cylinders.

Disadvantages of donor cylinder EGR architecture:

1. The system involves an additional EGR valve, and the control system for both is more complex.
2. The exhaust piping arrangement is slightly more complex.
3. Cylinder breathing and loading will differ slightly between donor and non-donor cylinders.

4. COOLER ARCHITECTURES

4.1 Coolant Cooled EGR Coolers

4.1.1 Shell and Tube EGR Cooler

Gas to coolant shell and tube coolers construction is well documented in other standards. This type includes both round and rectangular tubes, and is probably the most common choice for EGR coolers, with the gas passing inside the tubes and coolant around the tubes. Several illustrations are provided in Figures 5 and 6.

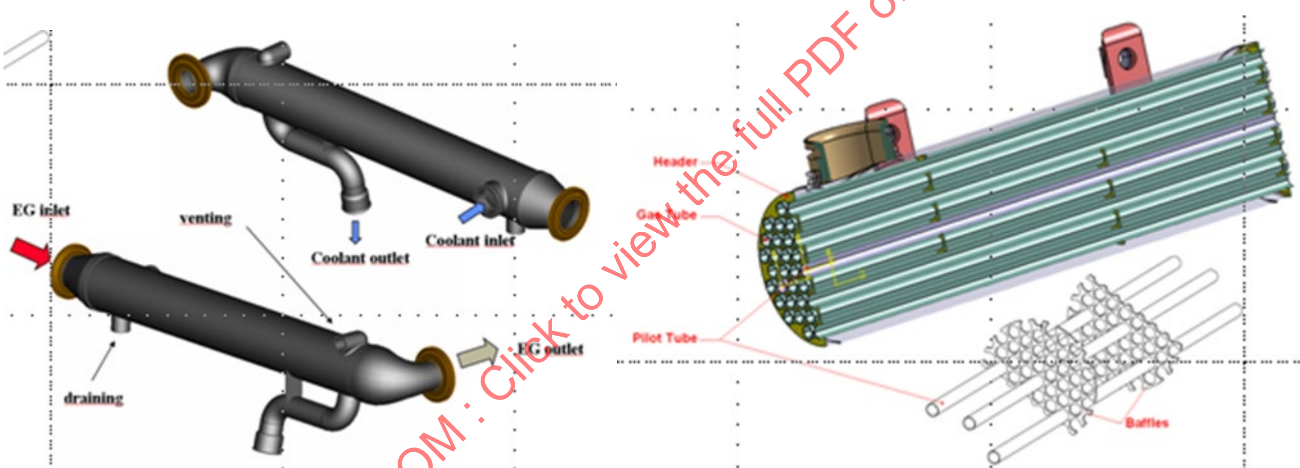

Figure 5 - Round tube with coolant side baffle design

The round tube design in Figure 5 with baffles gives a better distribution of coolant flow over the tubes at a macro level. But depending on baffle design, eddy currents and low velocity coolant zones within the shell do present a risk of boiling on small areas of the tubes (discussed further in failure modes). This risk can be mitigated in the design using computational fluid dynamics (CFD).

The rectangular tube-shell design in Figure 6, although it is usually designed without baffles, has the same inherent risk of non-uniform coolant flow and localized boiling anywhere in the cooler. A major factor in the distribution of flow over the tubes will be determined by the coolant inlet piping and manifold geometry. But from inlet to outlet, each coolant streamline will distribute itself to flow the path of least resistance until each streamline has the same pressure difference from inlet to outlet. Again, CFD is a useful tool to design away from tube surface with low coolant velocity and mitigate the risk of boiling.

Figure 6 - Rectangular tubes without coolant baffles

The tubes may be a stacked design of multiple tubes running the width of the cooler, or a tube matrix of multiple tubes within the height and width of the shell providing more heat transfer surface density. Illustration of both are shown in Figure 7. In both cases the tubes are closely spaced with limited distance for coolant flow between tubes. The driver for this is limited space for mounted components outside the engine block, so high surface area density helps meet this constraint. This further exacerbates the concern of having sufficient velocity over the entire core's surface area to avoid boiling. An extreme example of temperature difference may be a coolant temperature of 90 °C and 500 °C exhaust gas. Not only does this present a potential for boiling, but the temperature gradient across these tubes, usually less than 1 mm thick, results in a very high thermal gradient across the wall thickness, and resulting thermal stress.

Figure 7 - Rectangular tubes in stacked or tube matrix internal configurations

Special design considerations apply to both designs.

1. The flow orientation is preferably in parallel flow; e.g., both gas and coolant inlets are at the same end of the cooler. This is not the most efficient configuration for heat transfer, but the entering temperature difference is so large that high coolant velocity at the hot gas inlet for boiling mitigation, and avoidance of tube-header joint expansion and thermal fatigue, make parallel flow the preferred choice.

2. The mounting and coolant plumbing of the cooler must have the coolant entering the bottom side of the shell, and exiting the top side. This helps ensure that any vapor bubbles that do form on the tubes, which will rise upward due to buoyancy, will be carried to the exit on the high side of the cooler. The same rationale applies to filling a dry system and venting air.

3. Further provisions for venting air during filling are critical to ensure that no air is entrapped within the cooler after the system is filled and the engine is run with hot EGR flow inside of a tube surface with no coolant coverage. This can be accomplished with vent lines, as illustrated in Figure 5, or by proper orientation of the cooler mounting relative to the coolant lines.

4.1.2 Bar and Plate EGR Cooler

Bar plate cooler construction is also well documented in other standards, with an EGR illustration shown in Figure 8. While there are many design options for hot and cold side inlet and outlet orientations, the same system and component design considerations mentioned above still apply. Attention to flow distribution providing sufficient coolant velocity over the entire internal heat transfer area is still a key design consideration.

Figure 8 - Bar plate EGR cooler

4.1.3 Layered Core EGR Cooler

Brazed layered core construction is not as well documented in other standards. The cooler functions with alternating layers of hot gas and liquid coolant between each layered plate. Surface area density is much higher than tube-shell coolers, so this design is space efficient.

Again, there are many design options for hot and cold side inlet and outlet orientations, the same system and component design considerations mentioned above still apply. Attention to flow distribution providing sufficient coolant velocity over the entire internal heat transfer area is still a key design consideration. Attention to inlet and outlet header design on both hot and cold sides, and its affect on internal flow distribution is a key design consideration.

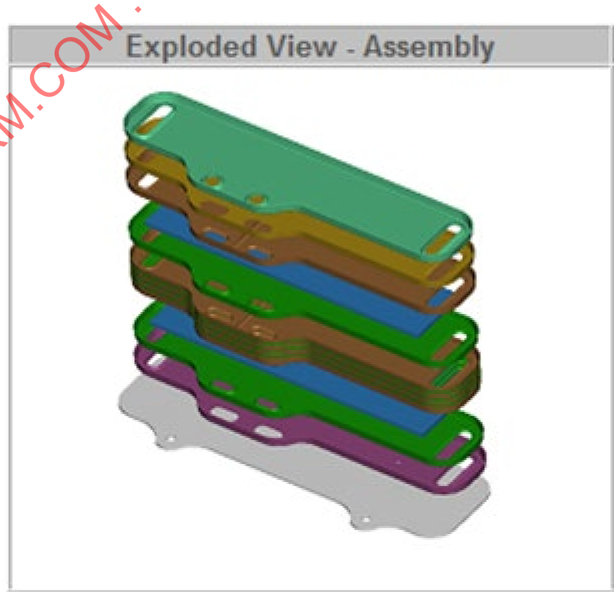

Figure 9 - Layered core EGR cooler

4.1.4 Tube and Fin EGR Cooler

The tube and fin or round tube and fin design has been known in many industrial engine charge air cooler applications for many years. Manufacturers have been using the basic design as a starting point for further development of the tube and fin technology to meet the requirements of EGR cooling especially in the industrial engine market where rather large thermal performance demand exceeds the capabilities of typical tube and shell automotive EGR coolers.

The exhaust gas is flowing on the fin side and the coolant on the tube inside. Very high operation temperatures and corrosion potential from acid exhaust gas condensate require high grade stainless steel materials. Depending on available coolant flow rates, the coolant typically has several coolant passes through the cooler matrix in a serpentine pattern to maintain the required coolant velocity in the tubes and best possible pressure drop curve over the entire cooler. A parallel-cross flow arrangement of the coolant side is suggested to start with the lowest coolant temperature and highest coolant pressure possible at the hot end of the cooler to reduce the risk of coolant boiling and overheating components.

CFD analysis is highly suggested for homogeneous flow distribution on water and gas side due to high heat flux especially at the hot gas inlet which leads to common failure modes like boiling, too high component stress from thermal growth, and thermal fatigue.

The gas side CFD should include upstream geometry (gas inlet diffuser and piping) which might result in non-uniform inlet conditions and potential high velocity plumes which drive up localized heat flux. Water manifolding CFD is important to ensure near uniform velocity distribution between tubes to avoid local water shortage increasing the risk of boiling and potential dry out followed by significant thermal expansion of the tube.

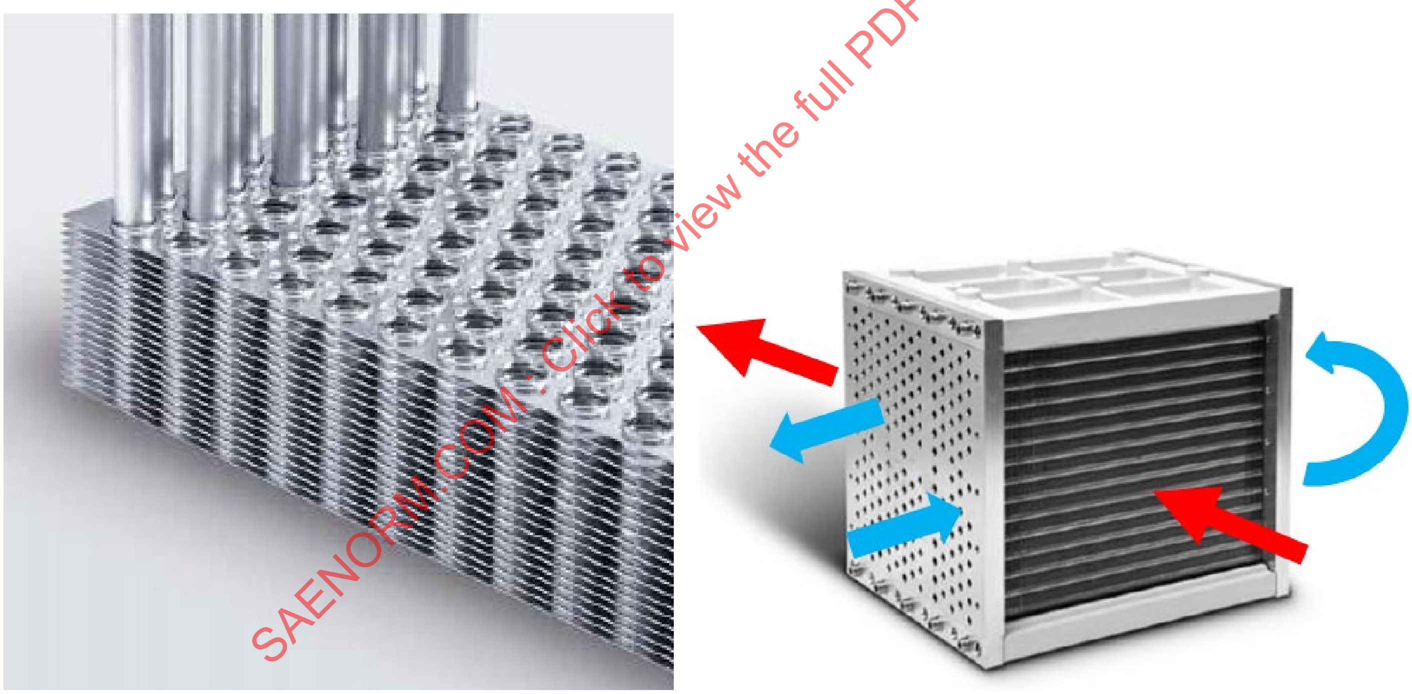

Figure 10 - Round tube and fin EGR cooler

5. EGR COOLER DEFINITIONS

5.1 Boiling

Boiling can occur anywhere on the exhaust tube coolant surface where the wall temperature exceeds the bulk coolant flow temperature (outside the boundary layer), the temperature difference designated as superheat. The value of superheat required for nucleate boiling to begin to occur on the tube depends on other variables:

- Heat flux at the tube surface
- Coolant absolute pressure
- Coolant velocity flow field surrounding the wall surface
- Coolant flow incidence angle relative to tubes
- Wall material and surface finish parameters related to surface finish: crater angle, crater depth, and crater density

A typical superheat value to initiate nucleate boiling on a smooth EGR cooler tube could be as low as 2 °C, but may range from 5 to 20 °C, depending on the other variables. Boiling may not necessarily occur at the hottest end of the tube. Consider that although the gas side temperature is dropping from inlet to outlet:

- Flow is not evenly distributed over the tubes
- Turbulence over rows of tubes changes incidence angles
- The coolant is absorbing energy as it moves downstream, thereby raising the bulk coolant temperature
- As coolant flow goes from inlet to outlet it is losing static pressure, thereby lowering the boiling point of the bulk fluid

In terms of coolant flow direction relating to bubble buoyancy and surface tension, upward is the preferred choice with flow velocity and buoyancy acting to help purge bubbles from the water passage. Horizontal water flow passages are a second choice and proper vent lines are a must. A downward flow direction is less preferred due to the flow force needing to overcome both surface tension and buoyancy to purge the passage of vapor bubbles and a potential for premature EGR cooler failure. High heat flux and superheat can be tolerated, but require higher tube velocity than the other configurations.

5.2 Condensation

Water is introduced to EGR systems as a result of the combustion process itself and due to the fact that engine induction systems ingest atmospheric moisture along with dry air. The moisture in the exhaust flow is subject to condensation if the localized temperature drops below the pressure dew point. The pressure dew point is the temperature at which the prevailing humidity ratio equals the saturated humidity ratio. This is the point at which the air stream maintains a maximum amount of water in a vapor state. Indeed, at the pressure dew point the water vapor partial pressure is equal to the saturation pressure. Moisture content beyond this in the air stream, will condense. Condensed water can chemically react with fuel, sulfur and other elements resulting in an acidic fluid which can be harmful to the internal surfaces of EGR coolers.

5.3 Coolant Inlet

The line connection where coolant enters the cooler. The flow direction may be parallel or perpendicular to the gas side inlet tubes/plates, depending on the cooler construction type, and is likely not evenly distributed. Flow direction relative to each tube depends on the intake manifold design and outside wall geometry and determines incidence angle relative to the tube, which is a key variable determining boiling.

5.4 Coolant Inlet Pressure

Bulk coolant pressure entering the inlet manifold of the cooler. Boiling point is a function of absolute pressure, so given that this pressure drops as the coolant passes downstream due to viscous losses, entering pressure is a critical design parameter. Areas of even lower absolute pressure can occur in coolant eddy currents passing sharp corners inside the tube matrix.

5.5 Coolant Inlet Temperature

Bulk coolant temperature entering the inlet manifold of the cooler.

5.6 Coolant Outlet Temperature

Bulk coolant temperature at the exit manifold of the cooler.

5.7 Cooled EGR

Exhaust gas drawn downstream of the exhaust port and cooled by a cooler fluid in the cooling system inside a heat exchanger.

5.8 Cooled Gas Injection

Cooled exhaust gas mixed with the ambient filtered intake charge air either at the intake manifold or upstream of a turbo charger. The amount injected, or mixed with clean air, is determined by an EGR valve with the flow rate determined by the engine's electronic combustion control algorithm as a function of speed, torque, and other potential inputs. The mass fraction of cooled gas injection used in combustion is part of a recipe of other variables controlled to achieve desired exhaust port emissions including IMT, fuel injection timing, multiple injections and rates, injector nozzle hole diameters-number-spray angles, injection pressure, turbocharger boost pressure, intake and exhaust valve intake and closing crank angles, piston bowl geometry, and potentially others.

5.9 Cooler Exhaust Gas Inlet Manifold

Structural component which receives the bulk exhaust gas flow from an intake pipe and distributes it to multiple smaller passages (in tubes or between plates) leading into the heat exchanger portion of the EGR cooler.

5.10 Diesel Particulate Filter (DPF)

Part of a complete exhaust after-treatment system designed to remove particulate matter and unburned hydrocarbons, which result from incomplete combustion. DPF's help meet Environmental Protection Agency (EPA) regulated tailpipe emissions.

5.11 Exhaust Gas Outlet Manifold

Structural component which receives the exhaust gas flow exiting multiple smaller passages (in tubes or between plates) inside the heat exchanger portion of the EGR cooler, and collects the bulk flow for connection to the cooler outlet pipe.

5.12 Fouling

A term used to describe the effect of exhaust deposits accumulating on the gas side of the heat exchanger tube and header. Fouling results in a reduction in heat transfer coefficient at this surface, as well as increasing the pressure drop of the gas from the inlet to outlet of the cooler. These in turn result in higher cooled gas outlet temperatures and lower flow rates versus when the cooler is first produced. Some degree of fouling, or fouling factor, is built into the design specification for sizing the cooler. Fouled versus clean tubes are illustrated in Figure 11.

Figure 11 - Visual illustration of fouled EGR cooler inlet and tubes

5.13 Gas Inlet Temperature

Bulk exhaust gas temperature entering the inlet manifold of the cooler.

5.14 Gas Outlet Temperature

Bulk exhaust gas temperature exiting the outlet manifold of the cooler.

5.15 Nitrogen Oxide

NOx is a natural chemical byproduct of combustion. The nitrogen and oxygen are both elements naturally occurring in the atmospheric air entering the cylinder. It is formed non-uniformly in the cylinder, produced at an exponentially faster rate in areas of higher temperature during combustion. These high temperature areas are generally found at the piston bowl rim and the outside diameter of the piston near the crevice volume during the beginning of the power stroke. NOx is an EPA regulated tail pipe emission.

5.16 Series EGR Cooler

In this heat exchanger application, the exhaust gas makes a single pass through a single cooler. The series term implies that the tubes first pass through a higher temperature coolant section (normally JW), separated from a lower temperature coolant circuit section for further cooling of the gas in the same tubes.

5.17 Thermophoresis

There are two fouling transport mechanisms: physical and thermal. Thermophoresis is a subcategory of thermal. Thermophoresis describes a phenomenon in which particulates are forced from a region of high temperature to a region of low temperature. Physically, the gas molecules in the high temperature region have more energy and push harder against the particles than the region of low temperature, thus driving the particles to the low-temperature region toward the cooler walls. Thermophoresis is important for particles up to about 10 μm, and is considered by some to be the primary transport mechanism in fouling.