The New 2.7L Turbo Engine from General Motors

Abstract
An all new gasoline engine has been developed for use in the General Motors’ Full-Size Truck, light duty line up. There are six core efficiency building blocks which have been employed to deliver segment leading CO₂-emissions and drivability performance. The core CO₂ enabling technologies are:

• Advanced high charge motion combustion system enabling 22 bar BMEP on 91 RON fuel
• Integrated Exhaust Manifold (IEM) and Dual-Volute turbo charger producing peak torque starting at 1,500 RPM
• Significant light weighting resulting in leading edge mass vs engine displacement
• Low friction enabling technology including a fully variable oil pump, offset bores and other “Efficient Fundamentals”
• Advanced active thermal management with split targeted cooling
• Tripower multi-discrete-variable valve lift system enabling Early Intake Valve Closure (EIVC), cylinder deactivation and high lift states

These technologies complement each other, delivering both on cycle and real-world CO₂-emissions reductions, as well as customer-focused torque response needed in this demanding application. Moreover, much attention was placed on increasing customers’ expectations of drivability and transient response.

In this paper, the engine’s architecture, mechanical design, and integration of these core CO₂ enabling technologies will be described. The control of each subsystem including actuators, and sensors will be explained to show how these systems are implemented. Finally, an evaluation of customer benefits will be explained.

Abstrakt

In diesem Beitrag werden die Architektur des Motors, das mechanische Design und die Integration dieser Kerntechnologien für CO₂ beschrieben. Die Steuerung jedes Subsystems einschließlich Aktuatoren und Sensoren wird erläutert, um zu zeigen, wie diese Systeme implementiert werden. Abschließend wird eine Bewertung des Kundennutzens erläutert.
Introduction
The all new General Motors 2.7L Turbo engine was designed from the ground up to service the demanding light duty truck customer. Expectations of a light duty truck and its engine are unique, as many customers use their trucks in extreme conditions in a wide variety of applications, often requiring high output and demanding durability. The 2.7L Turbo is optimized for light duty full-size trucks, as it delivers up to a 15% reduction in CO₂ emissions, and provides 14% greater torque output, is over 38 kg lighter and results in over 1 second faster 0-100 kph time over the 4.3L V6 engine it replaces. This engine option, an inline turbo charged 4-cylinder engine using GM’s “Efficient Fundamentals” strategy, which places focus on mass savings, friction reduction, combustion efficiency and thermal management, all combine to provide customers with excellent value and performance. This paper will describe how the attributes of this engine including architectural decisions, application of 6 core technologies, and use of “Efficient Fundamentals” merge to achieve challenging customer-focused demands and capability.

Engine Architecture
Engine architecture started with displacement and cylinder count studies based on 91 RON fuel, with power density targets of 85 kW and 175 Nm per liter (22 bar BMEP), for a balance of performance and efficiency. Determining the correct level of power for this engine was done based on vehicle level performance attributes. The primary attributes used to determine the engine’s torque curve were a 0-100 kph performance of 7.0 secs, assuming a 2,275 kg test weight class, as well as towing requirements. Towing and cargo capacities were set at over 3,000 kg and 1,000 kg respectfully. The results of these studies indicated the need for a power curve with a peak power of 230 kW and 475 Nm starting from 1,500 engine RPM. Based on these criteria, the ideal engine displacement was determined to be 2.7 L.

Various engine configurations were studied including “Vee” and “inline” engines of both 4- and 6- cylinder arrangements. Key attributes were low RPM torque capacity and response, as well as high power density with stoichiometric operation. These two attributes are very important in the light duty truck application for providing low fuel consumption while delivering high levels of towing and hauling capability. In order to meet or exceed customers’ expectations in these areas, a 90% of peak torque response at 1,500 RPM of 2.0 seconds and maximizing stoichiometric operation near full load were targeted.

Throughout architecture studies, the inline 4 engine arrangement proved to be the most capable of producing both low-end torque, and fast torque response. The inline 4-cylinder architecture also provides inherent benefits for cost, mass, and friction reduction along with combustion efficiency and engine packaging advantages. From a packaging perspective the turbo charger was the most important component for response and cooling, then catalyst packaging with high levels of face utilization. The inline 4-cylinder arrangement allowed for turbo packaging at the midpoint of the engine. This provides the turbo with near equal length runners from each cylinder for efficient blowdown, and an optimized Integrated Exhaust Manifold (IEM) capable of rejecting heat and cooling the exhaust before the turbine at high power conditions. The IEM, turbo integration and matching will be described in further detail in upcoming sections. The IEM, turbo and catalyst flow domain are shown in Figure 1.
Engine efficiency core technology selection was determined using the GM Balanced Architecture (BA) process. This process explores available technologies which can be applied to a new production engine as developed through GM Advanced Engineering or co-developed with the industry supply base. Through the BA process, each potential technology is explored and developed adequately to understand its merit for the vehicle application. This is accomplished with the use of 1D and 3D engine simulations while in some instances full scale testing on surrogate engine families was performed. Attributes such as CO₂ reduction, cost, mass, performance, real-world fuel economy, packaging, technical risk, and others are weighed using value equations. Vehicle system modeling was used extensively to represent the fuel consumption maps of these technologies. Both internal and benchmark fuel consumption maps were used to determine the best vehicle level fuel consumption value of a complementary selection of engine technologies. The result yielded a broad minimum fuel consumption map in the engine operating range typical with the 8-speed automatic transmission. The engine efficiency technologies selected for the 2.7L Turbo engine are shown in Figure 2. These technologies can achieve up to a 15% reduction in CO₂ emissions.

**Figure 1. Exhaust Flow Path of Near Equal Length Integral Exhaust Manifold**

**Figure 2. GM 2.7L Turbo Engine Key Technologies**

**Base Engine Layout**

The block layout was studied to best package the crankcase and determine bore spacing, deck height, and crank offset, all of which were optimized to reduce mass. Special focus
was put on bore spacing as its impact on mass is the most significant. As part of the BA process, many design iterations were made moving the main oil gallery, with and without a dedicated piston cooling jets gallery. The final optimized design required only one main oil gallery positioned on the hot side of the engine for priority feed to the turbocharger. Activation of the cooling jets is accomplished with the fully variable oil pump by controlling pressure to overcome relief valves at the oil nozzles. There were additional benefits of this layout including allowance for a large volume, coarse PCV separator on the cold side of the block. There was much effort put in the very early stages of architecture to ensure Positive Crankcase Ventilation (PCV) passages with uncompromised flow areas and volumes. This was done to ensure high separation efficiency which is a key to minimizing oil consumption enabling extended change intervals and reducing potential for low-speed high-load abnormal combustion events known as Stochastic Pre-ignition (SPI).

The crankshaft architecture was iterated simultaneously with the block as bore spacing and deck height are interdependent on crank arm stiffness, and counterweight effectiveness. Many crankshaft architectures were designed, and compared for mass, friction, stiffness and inertia. Variables explored included main journal diameters, rod journal diameters, counterweight number, sizes and placement, as well as nodular cast iron vs forged steel material specification. The resultant is an optimized balance of both mass and crankshaft stiffness, achieved with forged steel employing four counterweights and hollow rod journal pins.

In order to define the minimum deck height, the piston, and rod package required kinematic and structural optimization. The piston, pin, and rod were iterated using Finite Element Analysis (FEA) with predicted cylinder pressures as an input while optimizing mass, and piston pin oil film thickness. Some of the architectural attributes of the lower end of the engine can be seen in figure 3.

![Figure 3. Lower End System Layout (left), Architecture Key Attributes (right)](image)

The cylinder head is integral to any engine and houses many components and subsystems. The 2.7L Turbo engine’s cylinder head design started with the combustion system including the IEM and targeted cooling water jacket. All these systems were each iterated many times with CFD analysis tools to improve their respective metrics, and to meet critical metal temperatures as well as stress within the cylinder head structure. More details will be provided on each of these system.

The cylinder head also required packaging of the tripower valvetrain, starting with packaging of the type 2 valvetrain through studies of positioning the Stationary Hydraulic Lash Adjuster (SHLA) and cam center locations. Upon the completion of these packaging studies it was
decided to move forward with a “pull / pull” SHLA and RFF arrangement due to the inherent friction and wear benefits this arrangement provides. A picture of the arrangement can be seen in Figure 4. The “pull / pull” arrangement placed the SHLA on the inboard side on the exhaust camshaft and outboard side on the intake camshaft. The inboard SHLA position of the exhaust valvetrain provided an additional benefit of allowing the IEM center runners to rising over the outer ports enabling a near equal length arrangement between inner and outer runners. The cylinder head system architecture shown in Figure 4 is cut through a section along the valve centerlines.

Figure 4. Cylinder Head and Valvetrain Layout Section

An additional noteworthy system requiring packaging into the cylinder head was the PCV system, more specifically, the second stage fine separator and associated porting to the intake ports. The system is dual path, with two stages of separation, and uses a Pressure Regulating Valve (PRV) along with a fresh air make up. PCV gases are drawn to the path of lowest pressure, either the intake runners or the turbocharger compressor depending upon boosted or non-boosted operation. There was special attention to packaging this system such that it would perform well at sub-freezing temperatures and provide benchmark separation efficiency reducing oil consumption, which is a contributor to abnormal combustion. As stated previously, a coarse separator is packaged on the side of the block. The PCV gases are then fed from the block into the cylinder head, through the fine separator. Under non-boosted operation the gases travel through the PRV, and a check valve through the head and into the intake manifold where the flow is distributed into each intake runner. This system provides optimal cylinder-to-cylinder distribution without concern of frost accumulation. Under boosted conditions, the flow passes through the PRV and bypass check valve than diverts through a series of secondary passages in the head and cam cover casting to an external PCV hose which connects to the compressor inlet. A picture of the PCV components and routing is shown in Figure 5.
Combustion System

Once engine architecture was selected as a 2.7L Turbo inline 4-cylinder engine, a new combustion system was designed for the 675 cc cylinder displacement. The combustion system development and optimization considered selected engine technologies and their interactions. The combustion system for the 2.7L Turbo required a special focus on low speed high BMEP knock performance as well as reducing abnormal combustion events or SPI. The basis for the new combustion system was a 4-valve homogenous charge side Direct Injection (DI) layout with central spark plug, and high tumble. The 2.7L Turbo combustion system builds on General Motors’ fundamental knowledge of combustion systems of this arrangement.

Extensive CFD and 1D modeling was undertaken to evaluate valve size, valve angle, valve location, port geometry, chamber shape and piston topology. Over 80 variants were analyzed to progressively generate the final combustion system, with several variants procured at key stages for single cylinder testing confirming improvements and model correlation. Testing of single cylinder engine hardware placed emphasis on knock and SPI reduction, combustion stability, combustion efficiency, emissions reduction among other combustion system performance metrics. The best performing combustion system was selected for implementation in prototype multi-cylinder engine build phases. Injector patterns were iterated with over 100 patterns being evaluated through CFD with focus on improving homogeneity and minimizing surface impingement. The chosen combustion system design along with its key attributes is shown in Figure 6.

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<tr>
<th>Key Attributes</th>
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<tr>
<td>Stroke</td>
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<tr>
<td>Compression Ratio</td>
<td>10.0:1</td>
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<tr>
<td>Intake Valve Ø</td>
<td>37.5 mm</td>
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<tr>
<td>Exhaust Valve Ø</td>
<td>32.0 mm</td>
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<tr>
<td>Intake Valve Angle</td>
<td>15.5°</td>
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<tr>
<td>Exhaust Valve Angle</td>
<td>18°</td>
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<tr>
<td>Exhaust Lift</td>
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<td>Intake Duration - High/Low</td>
<td>29/18°</td>
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<tr>
<td>Exhaust Duration</td>
<td>76/7°</td>
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<td>150° / 301° ATDC</td>
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Figure 5. 2 Stage, Dual-Path (PCV) System Components and Routing

Figure 6. Combustion System Design (left) and Key Attributes (right)
The resultant combustion system development process yielded a design with an efficient intake port which balanced both filling efficiency, high tumble motion and high efficiency of Turbulent Kinetic Energy (TKE) generation before and during the combustion event. The chosen tradeoff of mixture motion (Tumble Index) and filling efficiency (Bore Area normalized Discharge Coefficient, CdBA) is shown in Figure 7.

![Figure 7. Tumble Index vs Filling Efficiency (CdBA)](image)

High levels of tumble at a macro level are well understood to promote mixture homogeneity and help reduce particulate generation. However, too high of tumble levels can also have unwanted consequences, such as hampering filling efficiency requiring higher levels of boost. As shown in Figure 7, flow capacity and tumble generation are often inversely proportional and must be properly balanced. Additionally, high levels of tumble can result in undesired combustion propagation due to residual in-cylinder flows during the combustion process. This phenomenon can result in increased knock at high speeds and loads. Much of the combustion system early development for the new 2.7L Turbo focused on conversion of tumble into TKE, not only from the perspective of maximizing TKE levels, but also proper positioning of the TKE core relative to the spark plug. The resultant improvements in mixture homogeneity and TKE distributions can be seen in Figure 8 which shows the progression from the early stages of combustion system development to the final production design. Note the base level of flow vs tumble was unchanged throughout the program but as the combustion system progressed TKE conversion was greatly improved as seen in Figure 8.

![Figure 8. Mixture Equivalence Ratio (EQR) Homogeneity (left) and Turbulent Kinetic Energy Contours (right) of Early and Final Combustion System Designs](image)
Integrated Exhaust Manifold and Dual-Volute Turbocharger

The 2.7L Turbo was designed as a dedicated single displacement turbocharged engine, thus allowing full optimization of turbocharger integration, and associated interfaces for this engine. An optimal turbo charging system was important not only from a performance perspective but also to ensure truck durability and drive quality. As stated previously, an equal length exhaust arrangement from each cylinder to the turbine was the goal. Equal length and volume from each cylinder enable balanced blow-down energy and temperature to the turbine. This improves turbine efficiency as well as minimizes temperature gradients in the turbine and housing. Thus, positioning of the turbine in the middle of the engine between cylinders two and three was a natural choice. The IEM plays a critical role for optimum turbine efficiency and temperature; therefore, many design iterations were performed using 1D simulation tools to balance overall length, effective diameter, Mach number and heat rejection. Once the ideal configuration was determined, the passages were fully optimized using 3D CFD simulation tools to represent the flow non-uniformity caused by practical packaging constraints within the cylinder head. GM racing engine exhaust designs were used to inspire the 2.7L Turbo IEM such that the inner cylinder runners rise over the outer cylinder runners much like a header. Raising the inner runners over the outer runners inherently makes the runners closer in length. Studies were initially conducted utilizing a dual port outlet at the turbine flange, as shown in Figure 9. This configuration required the outer runners to travel to the midpoint of the head prior to the outlet flange. This configuration yielded a length difference of 35% between the inner and outer exhaust runners, even with the length increase of the inner runner routing. This difference in length was substantially reduced with a triple port design, as also shown in Figure 9, where the confluence of flow for cylinders 1 and 4 is moved into the turbine housing. The triple port concept was selected for the final design, which resulted in a runner length difference of only 9%. The triple port increased turbine housing complexity while significantly reducing cylinder head water jacket complexity and increasing cooling efficiency at the turbine flange.

![Figure 9. Dual Port IEM Flange Design (bottom) and Triple Port IEM Flange Design (top)](image)

The turbine housing design was also optimized to minimize the remaining length, and volume of the exhaust runners as well as mass reduction. One of the notable details which aided in packaging is the turbo housing to IEM lower mounting studs, which are angled to 14 degrees from perpendicular. This detail allowed the turbine center housing to raise up 20 mm saving approximately 20 mm of turbine housing length, as well as improving assembly tool access.

Turbo matching was of utmost importance to meet the very broad torque curve and the aggressive torque response for the program. Design work started with a traditional twin-
scroll turbo which proved to be inadequate to meet the requirements for power, low-speed torque and torque response time. The primary issue with the traditional twin-scroll turbo is caused by excessive leakage between the scrolls through the gap between the septum ‘tongue’ and the turbine wheel, as shown in Figure 10 left. Therefore, a dual-volute turbine housing configuration was selected to minimize the leakage between the two exhaust streams. The final dual-volute turbine and compressor wheels are matched to the engine to achieve the low-speed torque, torque response and power requirements.

The dual-volute turbo provides a thermodynamic efficiency benefit by keeping the exhaust pulses separated with minimal leakage between the volutes. The volutes are nested such that each of the two exhaust streams access the full width of the turbine wheel but only half of the circumference. The two exhaust streams are introduced to the turbine wheel on opposite sectors, as shown in Figure 10 right. The advantage of the dual-volute configuration is that the septum is relatively short reducing the metal temperature such that material thermal fatigue creates no practical limitation to the clearance between the septum and the turbine wheel, thus allowing less leakage or pressure loss between exhaust streams. Conversely, a traditional twin-scroll turbine septum tongue length is practically limited by extreme temperature because of the inherently large exposure to the exhaust stream with limited area for heat rejection.

Figure 10. Traditional Twin Scroll Arrangement (left) and Dual-Volute Arrangement (right)

In order to minimize engine pumping losses, an electronically controlled waste gate was also included as part of the turbocharger system. The E-wastegate provides several advantages over traditional pneumatic actuation systems. It allows independent control of the wastegate position regardless of boost pressure, thus allowing an open position during catalyst light-off. This feature improves light-off efficiency by bypassing the turbine wheel and reducing enthalpy loss prior to the catalytic converter brick. The E-wastegate is also commanded open during non-boosted operation which reduces engine pumping losses yet provides fast response to build boost when requested. Figure 11 shows the turbo system as well as key system attributes.
As stated previously, the combination of the dual-volute turbo, near equal length IEM, and the large displacement 4-cylinder engine arrangement, proved capable of producing 22 bar BMEP starting at 1,500 RPM with leading edge response. Figure 12 shows the resulting torque response at 1,500 RPM vs the USCAR scatter band.

"Efficient Fundamentals" Leading Edge Mass and Friction

Every aspect of the engine was considered for mass reduction in order to deliver leading edge mass for a 2.7L Turbo engine. Mass reduction of the engine isn’t only an important attribute to improve CO2, but in a light duty truck, reduction of engine mass can directly benefit payload capacity, snow plow weight capacity and improved driving dynamics.

A large part of the mass competitiveness is a result of the inline 4-cylinder engine architecture, employing a deep skirt die cast aluminum cylinder block, with cast in place iron liners. The balance shafts are uniquely integrated into a die-cast Lower Crankcase.
Extension (LCE), resulting in over 1 kg of mass savings over a traditional balance shaft module assembly, shown in Figure 13.

Furthermore, the balance shafts are geometrically designed to achieve high effective counteracting mass with low overall mass by optimizing the length and fan angle of the counterweight features. Additionally, they are made from forged steel which was chosen over cast iron due to the increased density to further improve overall size and mass effectiveness. Since the maximum length of the balance shafts was packaging constrained, and the oil pump resistance is minimized for friction, a split gear was used to prevent any audible rattle from perturbations of the reversing gear lash. The split gear spring rate and preload are optimized to avoid any customer perception of rattle or whine.

From an “Efficient Fundamentals” perspective, all components and major component release groups were tasked with achieving leading edge mass as seen in Figure 14. This resulted in use of stamped aluminum front and rear covers with bonded lip seals, as well as a composite oil pan, PCV separators and intake manifold. The “Efficient Fundamentals” strategy also created an all new catalog of lightened fasteners. The fastener mat points are shorter than prior designs and the heads are dished for further mass reduction. Many fasteners were also downsized by improving the specification from 8.8 to 9.6 grade, thus allowing for more usage of M5 fasteners for most low stress joints. This attention to detail resulted in over 2 kg of mass savings at an engine assembly level. All components were subjected to multiple rounds of FEA including automated structural optimization to balance mass and structure as well as manufacturing analytical tools to assess casting or formability of these low mass designs.
An engine with low friction was an enabler to meet engine fuel consumption targets. To address this, several components were examined and optimized – main and rod bearings, piston rings, bore hone, valvetrain spring loads, cam drive and adding a controlled continuously variable displacement oil pump. The sizing for the main and rod bearings to lower friction must be balanced against strict durability requirements. Significant analysis was performed to optimize the balance of the main, and rod bearings at 51 mm diameter each. The rod bearings further added durability robustness by using tri-metal rod bearings.

The tri-metal rod bearings used with a forge steel crankshaft improves fatigue resistance while optimizing the bearing size to keep friction minimized.

The use of low-tension rings was also employed in the 2.7L Turbo engine. This was done using an open deck block design, which results in bore distortions that only require low ring tensions for conformability. Therefore, the lower ring force contributes to a low friction interface. The top ring is steel with PVD which also reduces friction of the engine.

Significant focus was put on identifying the proper bore hone specification for a low friction interface. GM developed a test matrix of various bore surface finishes (Rpk, Rk, and Rvk) to be measured for friction on fired and non-fired engines. From this data, GM set new requirements for the bore hone specifications for the 2.7L Turbo engine. The bore interface friction loss was also reduced by offsetting the crankshaft main bearing centerline from the bore centerline, such that the major thrust load of the piston is reduced during the expansion stroke.

Another contributor to overall engine friction reduction is optimized was valve spring loads. The valve spring must always balance load losses versus having enough margin to stay closed under high turbine inlet pressure conditions on the exhaust, as well as maintain dynamic control at high speeds on the intake. If the turbine inlet pressures exceed the spring margin of the exhaust valve, the exhaust valve could reopen unintentionally. The springs were thus optimized to minimize load, friction, and have high speed dynamic stability while maintain sufficient margin to remain closed during conditions of peak turbine inlet pressure.

Special attention was put on friction in the choice of the timing drive, where 3 different systems were explored: (1) Wet belt (2) Inverted tooth chain (3) 8 mm roller. All of these systems were designed in sufficient detail to understand their advantages in mass and packaging as well as friction and timing accuracy assessed through 1D simulation. Based

Figure 14. Benchmark Comparison of GM 2.7L Turbo Engine Mass (DIN)

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on these studies the choice was made to use option (3) an 8 mm roller with cushion rings, as it provides the best wear resistance, durability and the most accurate timing over the life of the engine. All details within the system were reviewed including the guiding elements of the chain drive optimized for position, intrusion and materials to further reduce friction and ensure truck durability. Several iterations of chain path layouts and hydraulic tensioner tuning were performed using simulation and confirmed with real time chain force and deflection verification on a firing engine to minimize noise and friction. The final tensioner design which was selected incorporates a pressure relief valve along with optimized spring loads and leak down rate for optimized friction and noise.

The final friction reduction contributor is the electronically controlled continuously variable displacement oil pump. This intelligent technology allows continuous optimization of the oil pump torque by operating on a demand basis for the requirements of the engine speed and load condition. Fixed displacement pumps need to be sized for extreme hot oil idle conditions to meet the engine’s oil pressure demands. Under normal driving conditions, the oil pump displacement would be “oversized” and therefore needs additional pump torque. The variable displacement oil pump takes advantage of the times the engine does not need full oil pressure and flow by moving to a smaller displacement setting. However, when the engine demands more oil based on engine speed, engine load or piston nozzle activation, the oil pump can move to a higher displacement setting to create additional oil flow and pressure. Enabling the lower pump torque allows for reduced parasitic losses from the oil pump compared to fixed displacement oil pumps can be observed in Figure 15.

![Variable Displacement Oil Pump](image)

**Figure 15.** VDOP Torque Advantage (left), VDOP Key Attributes (right)

The combination of all the friction reduction enabling technologies within the 2.7L Turbo engine yields Friction Mean Effective Pressure (FMEP) at the industry leading edge, as shown in Figure 16 where the friction of the 2.7L Turbo engine is compared to the USCAR benchmark data as a function of engine displacement.

![Benchmark Comparison](image)

**Figure 16.** Benchmark Comparison of GM 2.7L Turbo Engine Friction (USCAR Database)
Advanced Active Thermal Management with Split Targeted Cooling

The 2.7L Turbo engine was designed to include the latest General Motors Advanced Thermal Management (ATM) system. The new ATM system delivers impressive CO2 reduction by combining three key technologies: Targeted Cooling II with exhaust waste heat recovery, an Electric Water Pump, and the Coolant Control Valve (CCV). These systems when combined complement each other creating a highly flexible cooling system capable of extracting heat more efficiently from critical metal surfaces. The heat is then optimally distributed for use in warming or cooling system components to provide rapid warm-up and on-demand cooling. The majority of the CO2 improvements are realized through four (4) areas: (1) The reduction of parasitic spin losses by rapid warming of the engine with reduced cooling; (2) Rapid warming of engine and transmission oils through exhaust heat recovery; (3) On-demand split cooling which allows higher block temperatures without compromising head temperatures and knock sensitivity and; (4) Improving knock performance by better managing hot spots in the combustion chamber. Furthermore, a customer benefit is realized through improved cabin heating due to the split cooling jacket volumes making hotter coolant available sooner. Each of the key technology components of the ATM system will be described in further detail in upcoming sections.

Targeted Cooling II

Targeted Cooling was first introduced at GM in 2007 in the Chevrolet R07 NASCAR racing engine. This is a water jacket design which intelligently directs cold water to the hottest spots of the cylinder head and block through a cross flow arrangement. This strategy gave these engines an edge in restrictor plate racing, where parasitic losses can mean the difference between winning and losing. Targeted Cooling first debuted in GM production engines in 2015 with the Gen II High Feature V6 3.6L and 3.0L engine family. Both executions proved significant downsizing in water pump flow is achievable in comparison to a traditional cooling system. Downsizing the pump resulted in lower power required to move the required volume of coolant to manage critical metal temperatures. For example, the 2.7L Turbo Electric Water pump consumes 85% less power than a mechanical pump sized for the same engine designed with a traditional cooling system. Targeted Cooling and its reduction in pressure drop is a key enabler allowing an E-Water pump to adequately cool the 2.7L Turbo engine

The 2.7L Turbo has implemented GM’s improved Targeted Cooling II design which further improves system restrictions and introduces zones into the cooling system. In addition to the split flow between head and block, the IEM cooling jacket flows to its own outlet, enabling effective exhaust gas heat recovery. Flowing all components in parallel reduces the pressure drop through the system in comparison to conventional arrangements with more system elements in series. Separation of the components into zones also allows for better design of flow splits. For example, at peak power the cooling requirements of the block requires only 16% of the total flow, whereas the cylinder head and integrated exhaust manifold requires more than 65%, with the rest utilized in oil heat exchangers and turbine housing cooling.

Significant CFD analysis work was performed to optimize flow in the water jackets and achieve the proper distribution (flow split) and the required restrictions. The die cast block employees a relatively short jacket length of approximately 70% of the stroke and a composite flow diverting insert to direct inlet flows to the hottest locations in the block. The insert also increases average velocity throughout the jacket, increasing heat transfer near the top of the bore, thus improving temperature uniformity, as shown in Figure 17.
The cylinder head jacket design consists of an upper and lower jacket, with drillings at key points providing flow from the lower to the upper jacket. The combination of upper and lower water jackets forms a clamshell around the IEM producing an effective exhaust gas heat recovery heat exchanger, shown in figure 18.

The optimized jackets of the cylinder head result in even cylinder-to-cylinder temperature balance while controlling critical metal temperatures and delivering effective exhaust gas heat recovery. Figure 21 shows the critical metal temperature contours in the head as well as cylinder-to-cylinder temperature balance.

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**Figure 17.** Block Water Jacket w/ Insert Velocity Streamlines (left), Block and Head Inlets / Outlets (right)

**Figure 18.** Water Jacket Design Including IEM Port Cores

**Figure 19.** Cylinder Head Metal Temperature Contours
The resultant metal temperatures shown in Figure 19 are achieved through the optimized cylinder head jackets. The jackets provide coolant entering the head at higher velocities to the combustion chamber and exhaust manifold flanges, while also capturing the hottest coolant from the IEM separately for accelerating warming modes. Each cylinder chamber is fed with three flow streams, parting between and around the exhaust valve seats, then to the hotter spark plug region, as seen in Figure 20 which shows the cylinder head jacket optimized velocity contours and flow directions.

![Image](image1.png)

**Figure 20.** Cylinder Head Water Jacket Optimized Velocity Contours and Flow Directions

**E-Water Pump**

ATM system flow demands are met by using the electric water pump, which allows flow control independent of engine speed. The pump is powered by a brushless DC 12V motor at a maximum output rating of 600W. The electric water pump is a smart device with all motor control electronics contained on the pump PCB and controlled by the ECM via LIN communication. The pump on-board controls also maintain a unique ability to protect the power electronics, if needed, by regulating operating speed. The pump is mounted to the block through 3 isolator fastener assemblies to help reduce vibration inputs. Figure 21 shows an image of the electric water pump on the 2.7L engine along with a summary of the pump’s key attributes.

![Image](image2.png)

**Figure 21.** 2.7L Electric Water Pump and Key Attributes Summary
The electric water pump is a key enabler in the ATM system in providing the capability for on-demand cooling. Unlike conventional mechanically linked water pumps that are proportional to engine speed only, the electric water pump allows more precise flow delivery for load variations as well. This allows for optimal flow rates at all operating conditions, thus reducing overall pump losses. Also, the electric water pump can provide secondary benefits with after-run cooling, avoiding boiling within the engine structure on hotter shutdown conditions.

Coolant Control Valve
The Coolant Control Valve (CCV) is a flow regulation and distribution module with 2 actuators and 2 flow accumulation chambers. It distributes flows to the various heat exchangers and provides proportional control of the cylinder head and block outlet flows. The CCV primary chamber collects flow from the cylinder head, IEM, and block and is responsible for controlling a majority of the engine flow and distribution to the radiator or bypass circuits. The CCV secondary chamber collects cold coolant from the pump or hot coolant from the IEM and is positioned to feed one of these inputs to the engine and transmission oil heat exchangers. Ball style valves in both chambers are driven by the same shaft from the Main Rotary Valve (MRV) actuator. Each ball has slotted openings which open or close the various port openings depending on commanded valve positions.

The Block Rotary Valve (BRV) controls only the block output flow into the primary chamber. This allows the block to operate at higher coolant temperature levels to minimize parasitic losses. Figure 22 below shows images of the described Coolant Control Valve and actuators.

Figure 22. Coolant Control Valve (CCV)

System Operation
The key elements of ATM are combined in 8 modes of operation to cover the broad range of customer usage. Figure 23 provides a schematic representation of the MRV delivery port opening schedule. The figure is broken up into two parts to represent chamber 1 (primary) and chamber 2 (secondary). The upper part of the partition line shows how the primary chamber openings would align with the 30 mm radiator outlet port or 20 mm bypass outlet port as the valve is rotated through its range. The lower part of the partition line shows how the 20 mm secondary chamber port openings align with either the input from the IEM or from the coolant pump. The secondary chamber functions as a selector with one of the inputs passing through to the oil heat exchangers. The different modes of operation may be visualized by considering the three output ports as they ‘slide’ from left to right in figure 23.
Mode 1 – Minimum Flow
After engine start up, when metal temperatures are still cold, it is ideal to have virtually no flow through the engine to help warm up critical bearing surfaces faster and reduce parasitic losses. In this mode, the pump is operated at its minimum speed of 300 RPM to produce trickle flow, allowing coolant temperature sensors to monitor the warm up process, but not enough flow to cool metal surfaces. In this mode, the CCV is used to shut off block, head, IEM and oil heat exchanger flow paths, such that only a small amount of trickle flow is realized through the heater core circuit.

Mode 2 – Warm Up Heater Mode
Similar to Mode 1, metal temperatures in this mode are too cold and significant flow is undesired through the engine thus the CCV remains in the same position as Mode 1. However, customer demand for cabin heating requires some heat transfer from the engine. The pump speed is therefore increased to pass more flow through the cabin heater.

Mode 3 – Warm Up Bypass Mode
After significant warm up has occurred, critical metal surfaces in the cylinder head will require more substantial flow for cooling. However, the cylinder block will not require flow because optimal temperatures have not yet been reached. In this mode, the advantages of split cooling begin to be realized, as the CCV can independently keep the block outlet flow shut, while opening the bypass port and allowing the cylinder head and IEM water jackets to flow. In this mode, coolant flow to the oil heat exchangers is still shut off since coolant temperatures are still lower than oil temperatures within the engine.

Mode 4 – Engine and Transmission Oil Warming Mode
As coolant temperature rises in the engine but oil temperatures are still relatively cold, it becomes beneficial to actively warm up the engine and transmission oil, to reduce parasitic losses. In this mode, the CCV is moved to a position that enables the hotter IEM outlet flow to transfer to the oil heat exchangers. This coolant temperature is often 5-10 degrees higher than the main cylinder head or block outlet flows. In this mode, the CCV may also begin to allow different ratios of block and head flow to go to the radiator or bypass circuits. Once flow to the radiator is needed, the valve position is controlled to deliver a desired coolant temperature at the radiator exit.
**Mode 5 – Normal Operation with Oil Cooling Mode**

Once normal operating temperatures are obtained, oil cooling is required and the CCV moves into a position that allows coolant from the pump to be distributed to the engine and transmission oil heat exchangers. In this mode, the strategies of on-demand cooling and split cooling water jackets are fully realized. The CCV’s Main Rotary Valve (MRV) is used to control radiator return temperatures to a target level for cooling while the pump controls overall flow requirements based on load. The CCV’s BRV is used to regulate block flows to maintain optimal block temperatures which can be higher temperatures versus the cylinder head for optimal friction benefits while controlling engine knock.

**Mode 6 – Maximum Cooling Mode**

For heavy to full load operation where oil temperatures become hotter than coolant, maximum cooling is desired and the CCV actuators move to a position that allows for full block flow and maximum cylinder head and IEM flows, with all engine flow routed to the radiator. The pump is operated at maximum speed in this mode to provide the most heat transfer in the engine and in the radiator.

**Mode 7 – Engine Off Heater Warming Mode**

When engine operation is shut-off but cabin heating is still required, such as under an auto stop condition, the electric water pump is still operated with the CCV in a similar position to initial warm up modes. Flows to the block, cylinder head and heat exchangers are shut off and only flow to the heater core through the IEM outlet is possible. The circuit is regulated to maximize heat transfer to the cabin through pump speed. This mode isn’t feasible with a mechanically driven pump.

**Mode 8 – Engine Off After Run Mode**

For extreme hot conditions where boiling is possible after engine shutdown, the ATM system is capable of maintaining some engine flow in an after run mode. The CCV allows the cylinder head and oil heat exchangers to flow at a moderate rate regulated by pump speed to eliminate boiling in the engine structure. However, the 2.7L turbocharger placement provides a natural thermal siphon of coolant after shutdown minimizing the need for after-run of the coolant pump.

**tripower Valvetrain System**

A key technology for CO₂ reduction of the 2.7L Turbo engine is GM’s new tripower valvetrain system. This newly developed variable valvetrain system is a 3-step lift system which provides three discrete lift states or modes of operation for improved fuel efficiency across the engine operating range. The three modes of operation are comprised of a high-lift mode for high power operating conditions, low-lift mode for mid-range loads, and Active Fuel Management® (GM’s cylinder deactivation technology) for light load conditions.

Structurally, the tripower architecture is comprised of a set of intake and exhaust sliding camshaft subassemblies, 6 electromechanical actuators with built in hall-effect sensors, and 4 direct barrel position feedback sensors. There were two different architectures studied to house all the tripower system components including camshaft cover arrangements. Those studies focused on mass, sealing robustness, shifting pin true position relative to the camshaft shifting grooves, overall part count, as well as other attributes. The conclusion of this study integrated the camshafts and actuators into a structural cover assembly. Benefits of this architecture are the best possible relative position control of actuator pins to sliding cam shifting grooves, part count reduction, lowest engine height and the ability to fully test the tripower valvetrain as a subassembly before installation on the engine. All of these components therefore are contained in an aluminum die cast cam carrier and ladder frame.
assembly for accurate position control between all subcomponents. Figure 24 shows an exploded view of all the tripower system subcomponents.

Figure 24. Exploded View of the tripower Valvetrain System Components

The various lift states are accomplished by a series of adjustable cam lobes that slide on splines of the base shaft, and their position is adjusted by the series of electro mechanical actuator pins, guiding the lobe to the commanded position using the shifting grooves on the lobes. The intake camshaft assembly contains two separate lobe segments or ‘packs’ to control lift modes of all intake valves of all four cylinders. Each lobe pack contains a set of cam lobes for two adjacent cylinders (cylinder 1&2 or cylinder 3&4). A set of cam lobes consist of high lift, low lift and AFM mode cam lobes. The exhaust camshaft assembly contains two separate adjustable lobe packs for two inner cylinders (cylinder 2 and 3) and fixed lobes on two outer cylinders (cylinder 1 and 4). The adjustable lobes at cylinder numbers 2 and 3 contain full-lift and no-lift lobes for all-cylinder and AFM modes respectively. The fixed lobes on the two outer cylinders provide full valve lift in all engine operating modes.

When shifting is commanded, the electromechanical actuators push shifting pins in to the barrel cam groove. The shifting pins apply the axial force required to overcome the detent position latch and to shift the lobe from one lobe position to the adjacent lobe position. The intake cam lobe uses two actuators to shift between the three modes. One actuator is used to shift the lobe from high-lift to low-lift mode and low-lift to AFM mode with successive...
actuation events. The other actuator shifts the lobe from AFM to low-lift mode and low-lift to high-lift mode, also in successive actuation events. This configuration thus allows three discrete lift modes to be available with the tripower system. Figure 25 and 26 below show the intake camshaft and actuator pins that would be used to shift the lobe packs from AFM to low-lift and low-lift to high-lift modes respectively.

As part of the diagnostic system, the actuators contain hall sensors to provide information to the engine controller on the state of the shifting pin position of being expanded or retracted. Along with the direct cam barrel position sensor for the cam lobes, the diagnostic system is able to continuously monitor cam lobe positions and actuator shifting pin positions to correctly identify current state of the hardware.

Figure 25. Shift sequence for increasing torque capacity (AFM to low-lift)

Figure 26. Shift sequence for increasing torque capacity (low-lift to high-lift)

Summary of Engine Performance
As this paper has described, GM’s new 2.7 L Turbo engine is all new and specifically designed as a turbocharged GDI engine for light-duty truck products. These customers are very demanding as they use their trucks in a wide variety of applications often with high output and durability demands. Many carry heavy payloads, tow trailers, plow snow, as well as numerous other uses including daily transportation. Trucks are routinely used in extreme climates while servicing a variety of commercial sectors. The intent of this engine was to meet or exceed the expectations of these customers as an alternative high-volume truck engine. This was accomplished by achieving broad peak torque of 473 Nm from 1,500-4,000 RPM and 310 hp at 5,600 RPM, as shown in Figure 27, while delivering leading edge mass, fuel economy, and responsiveness.
The turbocharged configuration offers customers full torque even at high altitude conditions. Customers will directly benefit from reduced fuel operating costs due to the leading-edge engine efficiency, as shown in Figure 28, which is accomplished using regular grade fuel.

Including key technologies like ATM and tripower valvetrain, the 2.7L Turbo engine provides great customer benefits in cold climates with faster cabin warm up and real-world fuel economy as well as protection from after-boil in hot climates. The attributes of the 2.7L Turbo engine are setting new industry benchmark levels while offering customers spirited acceleration, towing, payload with desired drive quality performance, such as gradeability and elasticity.
Definitions/Abbreviation
AFM- Active Fuel Management®
ATM- Advanced Thermal Management
BA- Balanced Architecture
BMEP- Brake Mean Effective Pressure
BRV – Block Rotary Valve
CBSFC – Corrected Brake Specific Fuel Consumption
CCV- Coolant Control Valve
CdBA- Bore Area normalized Discharge Coefficient
CFD – Computational Fluid Dynamics
ECM- Engine Control Module
EIVC – Early Intake Valve Closure
E-Wastegate- Electric Wastegate
FEA – Finite Element Analysis
FMEP – Friction Mean Effective Pressure
HP-DI – High Pressure Direct Injection
IAM- Intake Air Module
IEM- Integrated Exhaust Manifold
LCE- Lower Crankcase Extension
LIN- Local Interconnect Network
MRV – Main Rotary Valve
NASCAR- National Association for Stock Car Auto Racing
PCB – Printed Circuit Board
PCV- Positive Crankcase Ventilation
PVD- Physical Vapor Deposition
Rk- Surface Finish Core Roughness
RON - Research Octane Number
Rpk- Surface Finish Peak Height
Rvk- Surface Finish Valley Depth
SENT- Single Edge Nibble Transmission
SHLA- Stationary Hydraulic Lash Adjuster
SPI - Stochastic Pre-Ignition
TKE- Turbulent Kinetic Energy

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