ABSTRACT

The current trend to downsize automotive engines calls for engine performance recovery via torque enhancement techniques, such as hybrid drive or supercharging. Traditional supercharging techniques using mechanical chargers or turbochargers are costly and have a limited time response [1]. This study investigates a fluid-dynamic supercharger (FDS) operated by compressed air which provides momentary supercharging during increased torque demand. FDS offers precise, computer-controlled boost of up to 11 psi (75 kPa) with instant response to demand. Compressed air for FDS operation can be generated using recovery of kinetic energy during vehicle deceleration. FDS is compact, simple, economical, and it does not require an intercooler. Simulations show that even with a downsized engine, an FDS-equipped automotive vehicle would be “fun-to-drive.”

INTRODUCTION

The fluid-dynamic (FD) pump has been previously considered for momentary supercharging of internal combustion engines during periods of high torque [2]. In such a charging system, FD pump uses high-pressure air from an engine-operated compressor to entrain and pump intake air into engine cylinders. However, limited understanding of FD pump aerodynamics at the time led to low compression, poor pumping efficiency, ineffective controls and excessively large hardware.

The prospect for the use of fluid-dynamic supercharger (FDS) for boosting the output of automotive engines is now receiving new attention. FDS offers an effective reduction in response time lag in turbocharged engines and an acceleration performance recovery in downsized naturally aspirated (NA) engines. This study shows that a modern version of the FDS can be a very compact supersonic device delivering precise, computer-controlled boost levels of up to about 11 psi (75 kPa) with fast response to demand. FDS is compatible with electronic engine controls and torque management techniques, and its function can be coordinated with the operation of a conventional or automatic transmission. Compressed air tank can be in-part recharged using recovery of kinetic energy during vehicle deceleration and braking. Except for a compressor, FDS has no moving parts and does not require an intercooler. These attributes allow for easier integration into many automotive systems at a competitive cost. FDS offers an attractive alternative to mechanical chargers and turbochargers in spark ignition engines, and turbo lag reduction in turbocharged compression ignition engines.

This paper briefly reviews existing power boosting technologies, notes the history of FDS development, and focuses on the FDS physics, engineering, and control techniques. Design challenges, design options, trades, engineering solutions, and performance models are discussed. Comparisons to mechanical superchargers, exhaust-gas turbochargers and electric turbochargers are presented. Also included are comparative performance simulations of an FDS-equipped automotive vehicle.

TORQUE ENHANCEMENTS FOR PERFORMANCE AND FUEL ECONOMY

The current emphasis on fuel economy and reduced emissions drives the effort to downsize internal combustion engine power plants for automotive applications. However, downsized engines having smaller displacement often exhibit insufficient power and torque when operating with natural aspiration (NA). Such a power deficit is most noticeable under high-torque conditions, namely acceleration and grade ascent. It is well known that the performance of automotive vehicles powered by small displacement engines can be improved with torque enhancement techniques such as a hybrid drive and supercharging.

HYBRID DRIVE - An automotive vehicle equipped with a hybrid drive has dual propulsion means; one driven directly by an internal combustion engine and a second driven by a battery-operated electric motor. During low-torque conditions (such as constant speed travel on a level road), the engine has a spare power capacity that can be used to operate an electric generator and store the produced electric energy in a battery. During high-torque conditions, electric energy is extracted from the battery to power the electric motor, which assists the engine in propelling the vehicle. The battery may be also charged using energy recovered from vehicle deceleration. While being very effective, hybrid drive is rather complex and costly.
SUPERCHARGING - Supercharging (SC) is a method of introducing combustion air into engine cylinders at a density in excess of that obtainable by NA. Supercharging is accomplished with a supercharger, which is an air-delivery system, air pump, blower, or a compressor in the intake system of an engine. Increased weight of air charge allows generating increased power from an engine of a given displacement or generating a given power output from an engine of a smaller size, weight, cost, and emissions [3]. It has been suggested that engine downsizing in combination with performance recovery via supercharging could improve fuel economy by 7-9% [4].

MECHANICAL SUPERCHARGER - One commonly used supercharger is the engine-driven supercharger also known as a mechanical supercharger (MS). The MS is a positive displacement pump such as a vane pump, Roots blower, Lysholm (screw) compressor, or scroll compressor mechanically coupled to the engine. To operate, the MS requires significant engine power, which must be often supplied at the least desirable moment, i.e., during high demand on the engine output. Another limitation of MS is the low-volumetric output at low engine speeds. Finally, the transition from NA to SC mode must be carefully controlled to avoid a sudden surge in output torque. Solutions to these problems increase the MS complexity and cost. Note that a single-stage MS can provide a boost pressure ratio in excess of 2, see Figure 1.

TURBOCHARGER - Another commonly used supercharger is the turbocharger (TC). In a TC, a flow of engine exhaust gas drives a turbine, which spins a turbo-compressor feeding compressed air flow to the engine. The TC provides the advantage of relatively smooth transitions from NA to SC mode while utilizing residual energy of hot exhaust gas, which would otherwise be largely wasted. However, TC must run at very high rotational speeds (typically 20,000 to 100,000 rpm) and use sophisticated engineered materials to withstand the high temperatures of engine exhaust, both of which lead to rather costly construction. A turbocharged engine is susceptible to a slow response time known as the “turbo lag,” which is caused by the low pressure and low quantity of exhaust gases available to operate the turbine at low engine speeds. This translates to insufficient quantity of intake air delivered to the engine and insufficient torque. The turbo-lag is reduced in a variable geometry turbine (VGT), which alters the cross-sectional area of the exhaust gas flow in accordance with engine speed. However, this approach adds complexity and cost while reducing reliability. Note that the TC adds a back pressure to the engine exhaust, which increases the intrinsic pumping loss. A single-stage TC typically provides a boost pressure ratio of up to about 1.8.

E-CHARGER - A recently-introduced electric turbocharger (“e-turbo” or “e-charger”) uses a battery-operated electric motor to drive a turbo-compressor [6]. The e-turbo is primarily intended for momentary boost during periods of high-torque demand. Because the e-turbo operates independently of engine speed, it is less susceptible to response time lag. To be effective, the e-turbo must accelerate to about 50,000 rpm in less than one second. Meeting this challenge requires a combination of ultra-high power electronics, a powerful battery, and sophisticated controls [7]. So far, the e-turbo boost ratio has been modest, around 1.3. The full potential of the e-turbo may be realized with the future 36/42V automotive electric systems.

COMPRESSED AIR DELIVERY SYSTEMS - Momentary demand for increased power can be also satisfied by feeding the engine with compressed air from an air tank. Advantages of this approach include instant response and the capacity to provide high boost at a predetermined level. Known concepts [8,9], however, waste the potential energy of high-pressure air by venting the air into the engine without performing any useful work. This means that the air tank must have a large storage capacity, which translates to a large volume or high pressure, neither of which is desirable in an automotive vehicle. This limitation can be overcome with an FD pump using the potential energy of compressed air from an air tank to pump combustion air into the engine.

FLUID-DYNAMIC SUPERCHARGER (FDS) - Numerous attempts have been made in the past to adapt the FD pump for supercharging of automotive engines but no concept known to the author has ever entered production. The FD pump (also known as an ejector) has a deceivingly simple appearance, which often leads to misconceptions. In reality, ejectors, just as many other “simple” aerodynamic components (such as aircraft wings and rocket nozzles), must be properly engineered and operated to obtain desirable performance. For example, some earlier FDS used pumps with subsonic or sonic motive nozzle, and straight diffusers. Such simplified devices led to unacceptably low compression, poor utilization of compressed air, and narrow control range.

One of the more successful FDS attempts was made by General Motors (GM) in the late 1950s, Figure 2. GM’s FDS used a supersonic nozzle with a venturi-shaped
diffuser, and it was reported to produce a pressure boost of 4-8” Hg (about 14-27 kPa) [2]. In the GM design, all intake air flowed through the FDS diffuser even when the engine was naturally aspirated. To avoid excessive pressure drop in the NA mode, the diffuser throat area was made larger than appropriate for best FD pump operation. This approach limited the FDS compression to marginal values and resulted in a rather long diffuser, which complicated integration with the engine.

Figure 2: GM’s FDS concept from the late 1950s [2]

BASIC PRINCIPLES OF FD PUMPS

The FD pumps, also known as jet pumps, jet compressors, or ejectors are widely used in chemical and process industries for pumping gases [10]. One key advantage of an FD pump is that it is mechanically simple because it has no pistons, rotors, or other moving components. On the other hand, an FD pump requires a supply of compressed air to operate. This adds complexity in applications that do not have an existing compressed air supply. Figure 3 shows a general configuration of an air-operated FD pump for pumping air consisting of an air-operated motive nozzle, suction chamber and a diffuser duct. The motive nozzle, which operates in a supersonic regime, is fed “motive” air at high pressure and converts its potential energy into a kinetic energy of a high-velocity jet discharged into the suction chamber. Pumping action occurs when the air downstream of the nozzle is entrained by the jet, acquires some of its velocity, and is carried into the diffuser where it undergoes mixing. Kinetic energy of the mixture is converted into a potential (pressure) energy in the straight and diverging parts of the diffuser.

Figure 3: Schematic diagram of an ejector

Note that the operation of an FD pump is quasi-isothermal. This can be explained as follows: motive air flow is cooled by isentropic expansion in the nozzle. Subsequently, the mixed motive and suction flows are slightly heated due to compression in the diffuser. Overall, if the motive and suction flows are at ambient temperature, the discharge flow will be at about ambient temperature [11].

An FD pump is termed “non-critical” if the absolute pressure at the discharge is less than 1.8 times the absolute pressure at the suction [12]. A non-critical ejector is suitable for FDS because its operation can be controlled by varying the motive flow rate. This can be accomplished, for example, by changing the motive pressure or by changing the throat area of the motive nozzle. A nozzle with a variable throat size is known as the variable area nozzle (VAN). One commonly used VAN is a pintle nozzle [13]. Since FD pumps with fixed nozzles are constructed to operate at a design single point, VAN makes it possible for the pump to operate efficiently over a broader range of pressure and flow conditions. In particular, VAN allows optimizing the FD pump throat size over the range of required motive air flow rates and the range of supply pressures during a storage tank drawdown.

In an FD pump, the motive mass flow $d m_{m}/d t$ and suction mass flow $d m_{s}/d t$ sum up to yield the discharge mass flow $d m_{d}/d t$ as

$$d m_{d}/d t = d m_{m}/d t + d m_{s}/d t$$

One key parameter in FD pump design is the entrainment ratio $R_w$ which is the ratio of suction (entrained) mass flow and motive mass flow:

$$R_w = (d m_{s}/d t) / (d m_{m}/d t)$$

In view of eq. (2), one can rewrite eq. (1) as

$$d m_{d}/d t = d m_{m}/d t (1 + R_w)$$

It is easily seen that in FDS, achieving high $R_w$ values is important to economizing consumption of motive air.

There are no first principle models available for accurate prediction of FD pump performance. Computational fluid-dynamic (CFD) models are also unreliable for FD pump analysis because of the complex supersonic shock structures inside the diffuser. In practice, FD pumps are designed according to empirical models and experience [11].

Figure 4 shows a plot which relates entrainment ratio $R_w$ and back pressure (difference between discharge pressure $p_d$ and suction pressure $p_s$) for motive pressures from 50 to 150 psig (440 to 1120 kPa) in an air-operated FD pump drawing air at 14.3 psia (98.6 kPa) and 80°F (27°C). The regime of interest for engine supercharging is denoted with a broken line boundary. Note that the economical operation (high entrainment
As shown in Figure 4, Rw is also adversely affected by temperature and decreasing suction flow temperature. Generally, it increases with increasing motive flow temperature and decreasing suction flow temperature. As shown in Figure 4, Rw is also adversely affected when the suction pressure is reduced from 14.3 psia to 12.18 psia (84 kPa). The motive nozzle design controls the expansion of the motive flow and its momentum while the diffuser design controls mixing and recovery of static pressure.

One challenge to adapting an FD pump for engine supercharging is its limited control range. As the motive air flow is reduced from its design point value, entrainment ratio Rw drops off significantly, making the air flow is reduced from its design point value, which is a result of poor mixing due to under-filling the diffuser throat at less than nominal motive flow rates. Hence, an FD pump with a fixed diffuser throat area has a rather narrow range for efficient operation. This limitation makes it challenging to adapt ejectors for supercharging of automotive engines where a wide range of intake air flow rates must be accommodated. It is important to recognize that entrainment ratio of a “throttled down” pump could be restored to optimum value by reducing the area of its diffuser throat to match the reduced motive flow rate.

![Figure 4: Performance regime of an air-operated FD pump (back pressure = p_d – p_f) [14]](image)

**FDS DESIGN APPROACH**

A systems engineering approach was followed to adapt FD pump for supercharging of automotive engines. At the outset, supercharging needs were defined in view of FDS physics and operational constraints, Figure 5. This made it possible to evolve a set of design options and establish traceability of selected technologies.

As shown in Figure 4, an FDS operating in a non-critical regime can generate a compression ratio of up to 1.8, which compares well with traditional supercharging technologies (see Figure 1). Consumption of compressed air must be sufficiently low so that the size of the air storage tank required to support several momentary boosts can be easily fitted into an automotive vehicle. In addition, the air tank pressure should be as low as practically possible for safety and to keep the “cost” of compressed air within acceptable limits. As already discussed in the previous section, obtaining low consumption of compressed air necessitates high entrainment ratios Rw. Downstream of the throat, the diffuser walls must be gently sloped at small angles (typically less than 4 deg) to avoid flow detachment and consequential loss of compression. However, this leads to a rather long diffuser. Furthermore, FDS requires effective control of discharge flow (= engine intake air flow) over a broad range of conditions while operating at varying air tank pressure. Finally, FDS should avoid impeding the intake air flow when the engine operates in NA mode. These are difficult requirements to meet for an FD pump, which is traditionally optimized to operate at fixed pressure and flow conditions. Another challenge is mitigating acoustic noise.
Overcoming these challenges calls for innovative solutions. First, FDS should be equipped with a motive flow VAN. By changing the nozzle throat size, VAN can deliver a broad range of mass flow rates over a range of air tank pressures. Second, the FDS diffuser can be provided with a variable throat area, Figure 6. Such a variable area diffuser (VAD) can be adjusted in accordance with the motive flow rate to provide optimum entrainment ratio Rw, and thus, the best motive air economy. The required throat adjustment range should correspond to a desired engine speed range (typically about 1:4). The range of the diffuser throat may be also further enlarged to allow passage of intake air flow during NA mode. In particular, during NA mode, the diffuser throat should be widely open to offer very low impedance to engine intake air flow. During supercharging, the throat should be adjusted to the size corresponding to the best entrainment ratio for a given motive flow. This approach requires the range of throat variation to increase at least to 10:1. Suitable engineering concepts for VAD construction is described by this author in reference [15].

Figure 6: Variable area diffuser (VAD) concept

Although VAD provides a very elegant solution, its engineering implementation is non-trivial, especially if throat variation over a wide range is desired. An excessively large throat variation range can be avoided with a bypass duct as shown in Figure 7, which allows engine intake air to “bypass” the pump during natural aspiration.

Figure 7: FDS with a bypass duct

An engagement control valve installed in the duct is closed during supercharging to prevent a backflow of boosted air. The engagement control valve actuation is computer controlled and coordinated with FD pump operation to assure smooth redirection of intake flow as required for a smooth transition between NA and SC modes.

An alternative to VAD is using several FD pumps with fixed diffusers operating in parallel, Figure 8. While each pump has only a limited control range (typically less than 1:1.5), the pumps can be arranged to provide a piecewise continuous control over a desired broader range. In particular, the pumps can be designed to activate in sequence to accommodate increasing air flow requirements as may be necessary at higher engine speeds and/or higher boost. Preliminary analysis shows that four pumps should be sufficient to provide positive control over a flow range corresponding to engine speeds from 1200 to 4800 rpm. The disadvantages of this approach include complexity and the risk of uneven filling efficiency.

Figure 8: FDS with multiple parallel pumps

The compressed air system generally follows a layout used in air brake systems on heavy trucks, and it includes an engine-driven compressor, aftercooler, dryer and an air tank. The compressor preferably has two stages and it can be driven from the engine crankshaft via a clutch or an unloader (governor) valve for engaging the compressor on an as-needed basis. The compressor can be preferentially engaged during vehicle deceleration allowing recovery of a large portion of the vehicle kinetic energy otherwise lost to braking. To enable energy recovery in vehicles with automatic transmissions, independent lockup management may be used. Undesirable load on the engine can be avoided by...
disengaging the compressor during periods of high-torque demand. Later in this paper, it will show that in city driving, about 50% of compressed air required to operate FDS can be generated using vehicle energy recovery. This feature makes the FDS comparable to a hybrid drive and provides an edge over traditional supercharging technologies.

The aftercooler reduces the temperature of air delivered by the compressor to near ambient level. Excess moisture from compressed air is removed as a liquid condensate in the dryer. The air tank is sized to support a range of FDS operating modes. The analysis below indicates that a vehicle with a 1.5-liter engine equipped with FDS requires at 20 to 30-liter air tank charged to 2 MPa (~300 psig). Preferably, the tank would be fitted under the vehicle body where it could be configured as several smaller tanks for an easier fit and increased safety.

MODELING OF FDS PERFORMANCE

To simulate operation of an FDS-equipped engine, a simple mathematical model was constructed to correlate the FDS motive flow and pressure boost at any given engine speed. Simplifying assumptions in this zero-order model include 100% engine volumetric efficiency and negligible engine speed effects. Consider a 4-cycle piston engine with a volumetric displacement \( V_E \). When operating at a rotational speed of \( n \) revolutions per minute (rpm), the engine draws intake air at a volumetric rate

\[
\frac{dV_E}{dt} = \frac{1}{2} n V_E
\]

The quantity of air ingested by the engine depends on the intake pressure \( p_i \), and it can be expressed as

\[
q_i = p_i / p_o \cdot \frac{dV_E}{dt}
\]

where \( p_o \) is the standard pressure (101.325 kPa). Note the \( q_i \) is in units of volume-pressure/time and it corresponds to intake air mass flow. In the analysis below, the units of \( q_i \) will be expressed relative to air density at standard pressure and temperature (15°C). In particular, units of \( q_i \) can be standard-liters/second or standard-cubic feet per minute. Boost ratio \( R_B \) is defined as the ratio of engine intake pressure under SC (boost) conditions and under NA conditions:

\[
R_B = p_i,\text{Boost} / p_i,\text{NA}
\]

As explained earlier in this paper, FDS can provide essentially isothermal operation. This means that the temperature of boosted air is about the same as the ambient air temperature. In view of this, the boost ratio \( R_B \) can be expressed as

\[
R_B = q_i,\text{Boost} / q_i,\text{NA}
\]

By expressing the mass flow in the units of pressure times volumetric flow, eq. (3) can be rewritten as

\[
q_d = q_i (1 + R_w)
\]

Assuming that the FDS discharge pressure is the same as the engine intake pressure \( p_i \), eq. (8) yields

\[
q_i = q_i,\text{Boost} / (1 + R_w)
\]

\[
= p_i,\text{NA} R_B / (1 + R_w)
\]

\[
= p_i,\text{NA} R_B (dV_E/dt)/(1 + R_w)
\]

Substitution for \( dV_E/dt \) from eq. (4) produces the desired expression for correlating motive flow to boost pressure:

\[
q_m = n V_E p_i,\text{NA}/120 \cdot R_B/(1 + R_w)
\]

The units of \( q_m \) in the eq. 10 are standard-liters/second or standard-cubic feet/second. In FDS performance simulations, it is useful to express \( R_B \) in an analytical form. As shown in Figure 4, the entrainment ratio \( R_w \) is dependent on the motive pressure \( p_m \), the discharge pressure \( p_d \), and the suction pressure \( p_s \). In FDS, the suction \( p_s \) may be assumed constant and essentially equal to the ambient pressure \( p_s0 \). Thus, \( R_w \) can be approximated as

\[
R_w(p_s, p_m, p_d) = R_{w1}(p_s0, p_m) \cdot R_{w2}(p_s0, p_d)
\]

where \( R_{w1} \) is only a function of the motive pressure \( p_m \) and \( R_{w2} \) is only a function of the discharge pressure \( p_d \). An approximate functional dependence for each \( R_{w1} \) and \( R_{w2} \) was developed based on curve fits to FD pump test data, such as shown in Figure 4. Using these approximations, one can generate an FDS performance chart such as shown in Figure 10. This chart illustrates a three-way relationship between engine speed, boost ratio and compressed air consumption.

INTEGRATED CONTROLS

Unlike traditional SC technologies, FDS is completely programmable and can deliver a predetermined level of boost at any engine speed. Mathematical models such as described above can be used to control FDS performance. Figure 11 shows an architecture suitable for integrating model-based FDS controls with a torque management system.
In a torque-based engine control system, engine torque is used as the major interface between the engine control system and other subsystems of the vehicle control system [16]. FDS control sequence begins by sensing acceleration demand. Accelerator position data is converted to torque demand in accordance with a predetermined response function and modulated by external inputs such as vehicle speed. Torque demand is interpreted by the subsystem coordinator and appropriate signals are sent to ignition, fuel, and air charge controllers. The air charge controller coordinates the throttle position and boost demand. The latter is interpreted in the FDS controller and synthesized with engine speed, air tank pressure, and ambient pressure data. Requests for motive flow and engagement valve position are generated according to the FDS control model and sent to FDS actuators.

Integration with the torque management system allows reaping full benefit from the unique controllability of the FDS. In particular, integrated controls enable precise realization of the driver torque demand and improve the drivability of an FDS-equipped vehicle.

**MODELING OF VEHICLE PERFORMANCE**

A model was developed to simulate the performance of an automotive vehicle equipped with FDS. The model, which simulates engine and vehicle dynamics, is based on a road equation in the form:

\[
F = A \frac{d^2x}{dt^2} + B \frac{dx}{dt}^2 + C
\]

where \(F\) is the propulsion force, \(A\) is the vehicle inertial mass, \(B\) is the air resistance coefficient, \(C\) is the rolling resistance coefficient, and \(x\) is the distance traveled.

An exemplary vehicle with a 4-cylinder gasoline engine and a 5-speed manual transmission was chosen as a basis for the simulations. Pertinent vehicle parameters are shown in Figure 12. It was assumed that the engine has been adapted for supercharging and it is capable of efficient operation from NA to a boost ratio of \(R_B = 1.7\). Engine output torque variation as a function of engine speed was defined in accordance with the torque parameters shown in Figure 12. Losses in the drive train downstream of the engine were ignored.

### Parameter | Value | Units
--- | --- | ---
Engine displacement | 1.49 liter |
Engine torque with NA | 70 @ 1100 rpm | N-m
| 82 @ 2200 rpm | N-m
| 67 @ 4000 rpm | N-m
Transmission | 5 speed |
Gear ratios | 3.737, 1.963, 1.364, 1.000, and 0.776 |
Final drive ratio | 4.55 |
Vehicle equivalent inertia | 1250 kg |

For the sake of simplicity, it was assumed that boosted torque \(T_B\) was a direct product of the NA torque \(T_{NA}\) and the boost ratio \(R_B\), namely

\[
T_B = T_{NA} \cdot R_B
\]

The model allowed for comparative studies of several different types of engine aspirations: naturally aspirated - NA, large naturally aspirated - LNA, turbocharged - TC and FDS-equipped. In all instances, it was assumed the engines were the same and that they responded to intake air boost in the same way. An exception was the LNA engine, which represented an NA engine with a 50% larger displacement and having a correspondingly greater output. Except as noted, each simulation started...
with the engine at 1200 rpm and the vehicle firmly in first gear.

Each FDS and TC were assumed to have a maximum pressure boost ratio \( R_{B,\text{max}} = 1.7 \). The TC, which was assumed not to have a VGT, was modeled to start boosting at 1600 rpm and to reach a 90% of its maximum pressure boost at 2300 rpm. The TC was assumed to have an intercooler delivering an output air with a temperature 30°C above ambient. The model automatically corrected the density of delivered air according to temperature.

Figures 13 and 14, respectively, show the comparative evolution of boost pressure ratio for each FDS and TC engines as a function of time and engine speed during a typical acceleration of the vehicle in the first gear. Note the TC boost ratio is not temperature de-rated. The FDS provides instant response to demand (starting at 0 seconds and 1200 rpm). The roll-off near full boost is pre-programmed to allow for a smooth transition. The TC engine shows a characteristic time lag as it reaches its full pressure boost about 0.5 second after the FDS. Figure 15 shows the evolution of air density boost ratio with time and illustrated the effects of air temperature.

Figure 16 shows the comparative accelerations of NA, LNA, TC and FDS-equipped vehicles during initial acceleration in first gear at maximum performance. FDS shows an overall superior performance with quick response and highest level of acceleration. The acceleration roll-off after initial peak at about 0.25 second for FDS and about 0.55 second for TC is due to torque roll-off at higher engine speeds. Limited performance of the TC engine is due to two factors: 1) lower delivered air density; and 2) initial time lag, which delayed the engine from reaching higher rpm and, therefore, from developing more power quickly.

The vehicle performance model was set to shift gears whenever engine speed approached 4300 rpm. In all engine options, the throttle was 100% open for maximum performance; however, during gear shifts, it was relaxed to 30% power. In addition, the FDS boost was momentarily cut during gears shifts for smoother shift and to conserve compressed air.
torque enhancement capability, the FDS-equipped vehicle is the only one going through four gear shifts within this simulation window. The TC torque is about 11% lower than FDS, which can be directly attributed to the disparity in delivered air temperature. LNA torque is 12% lower than FDS, which is due to both the LNA definition and the disparity in delivered air density.

While the above simulations predict superior performance for FDS, excessive consumption of compressed air could make FDS impractical. As shown earlier in this paper, FDS air consumption increases with reduction in suction pressure and with increased boost ratio \( R_B \). These effects must be included in evaluating the FDS compressed air supply. Figure 19 shows each instantaneous and cumulative consumption of compressed air tracked by the FDS model during vehicle acceleration. Air consumption is shown for \( R_B = 1.7 \) (maximum FDS boost ratio), which corresponds to the simulations in the preceding section. Note the suction pressure is 98.6 kPa (14.3 psia). Figure 20 shows the instantaneous and cumulative consumptions of compressed air for \( R_B = 1.5 \), which represents a more economically operating FDS. Vehicle acceleration to 60 km/hr (~37 mph) represents typical city driving, whereas acceleration to 90 km/hr (~55 mph) represents a typical expressway entry. Each mode of driving (city or expressway) places different requirements on FDS-compressed air system.

CONSUMPTION OF COMPRESSED AIR

An exemplary “state of charge” diagram for an FDS air tank is shown in Figure 21. The tank is considered fully...
charged at 2 MPa (294 psig). The charge is tripartitioned into a minimum 25% charge (to maintain 0.5 MPa (74 psig) “floor pressure” for FDS operation), a 15% reserve, and an operating charge of 60%. Figure 21 shows (in a notional sense) a typical city driving cycle with the air tank discharging during vehicle acceleration followed by a recharging process during cruise and deceleration.

A reasonable target for air tank recharge in city driving is about 60 seconds. This would allow for a rather aggressive driving with one 5-second FDS boost for acceleration from a stop to 60 km/hr about every 65 seconds. A reasonable target for air tank recharge in expressway driving is 120 seconds. This would also allow for aggressive driving with one on-ramp entry acceleration to 90 km/hr every 130 seconds and one vehicle passing acceleration from 85 to 120 km/hr every 65 seconds.

As stated above, the FDS air tank can be in-part recharged with energy recovered from the kinetic energy of the vehicle during deceleration. One challenge in this area is the rather short typical deceleration time (10-15 seconds in city driving), which necessitates high charging rates. Sizing the compressor to recharge the air tank in such a short time leads to unacceptable large hardware. A more practical approach is to recharge a portion of the tank during vehicle cruise and to complete the charge during deceleration, such as shown in Figure 21. Partitioning of the recharge time can be dynamically allocated according to vehicle speed. For example, at higher cruising speeds, it may be expected that a need for further acceleration is small while the potential for energy recovery from deceleration is high. For this case, a large portion of recharging may be allocated to deceleration. The compressor charge rates can be actively controlled with unloader valves to optimize the charging rates. It is important to recognize that the later stage of the charging process is more energy intensive because the compressor has to discharge into an air tank at a higher pressure.

In the operational scenario shown in Figure 21, the compressor draws engine power during cruise. At the end of cruise, the tank should be only partially charged to allow completion of the charging process during deceleration. Using the exemplary vehicle and FDS parameters shown above, the power required to operate the compressor during a 50-second cruise at 3.5 standard liters/sec delivery flow rate is about 1.2 kW.

Since FDS operation handles a large amount of air, it is important to review the effects of atmospheric moisture. Consider FDS operation at 30°C (86°F) with 80% relative humidity. Under these conditions, 1 m³ of atmospheric air contains about 31 grams of water vapor. After compressing humid air to 2 MPa pressure and cooling it back to ambient temperature, about 30 grams of air moisture content will be converted into liquid condensate. For a typical city driving with acceleration-deceleration cycle repeating every 2 minutes, this

![Figure 21: Exemplary air tank “state of charge”](image)

Figure 22 summarizes the air consumption for power boost events in city and expressway driving conditions for 1.5 and 1.8 boost ratios and 84 and 98.6 kPa suction pressures. This figure also shows the implications for air tank size and compressor output. These results indicate that a 20-liter tank would suffice for city driving while expressway driving may, depending on boost conditions, require a 30-liter or larger air tank. One approach to reducing the air tank size requirements for expressway driving is to reduce the boost level with increasing speed and, thus, conserve compressed air. Because during deceleration the speed of an engine-driven compressor declines, the compressor must be sized to deliver more than the average output rates. Figure 22 also shows that low resistance (pressure drop) in the FDS suction flow path allows economizing on compressed air use. One favorable condition for low-pressure drop is that during FDS operation, only about one half of the intake air flows through the FDS pump suction port (with the remaining flow going through the motive nozzle). According to Figure 4, FDS performance (as measured by Rw) is more sensitive to a reduction in suction pressure than to an increase in discharge pressure. This information suggests placing an FDS pump upstream in the engine intake flow path.

![Figure 22: Impact of driving environment, boost ratio, and suction pressure on tank and compressor parameters for the exemplary vehicle](image)
translates to condensate production rates of 90 to 180 grams per hour. This condensate is removed from the compressed air system by an automatic water separator in the same manner as practiced in truck air brake systems.

Finally, the air tank charge must be well managed to achieve efficient FDS operation over a wide range of driving conditions. For example, one would like the air tank to be at the same time full for acceleration and empty to take advantage of energy recovery from deceleration. Such conflicting requirements can be resolved by a computer, which may use historical and current data on vehicle motion to optimize the tank charge based on predictive analysis.

PERFORMANCE OF COMBINED TC AND FDS

FDS can be also used in TC engines (especially compression ignition) to overcome the turbo-lag. The control strategy in a combined TC-FDS is to initiate the boost with an FDS and control the FDS output according to evolving TC speed to achieve a desirable level of boost over time. Figure 23 shows the results of comparative simulation of pressure boost conducted using the above model for a combined TC–FDS system. The figure also shows individual boost profiles for each TC and FDS system. Air temperature effects were ignored in this simulation. The simulation starts at engine speed of 1000 rpm. Being independent of engine speed, the combined TC–FDS delivers an instant response to boost demand. In practice, TC and FDS can be installed in series, preferably with the FDS downstream of the TC. Compressed air consumption in a combined TC-FDS is very low because the FDS is used only for a very short time, typically less than 2 seconds. If the turbocharged engine already has a compressed air system (e.g., for vehicle brakes), such a system can be adapted for FDS operation.

CONCLUSION

A fluid-dynamic supercharger (FDS) operated by compressed air was investigated for a possible use for momentary supercharging of automotive engines during periods of increased torque demand. Existing boost technologies were described, FDS physics was outlined, a design process for adapting the FDS was described, and a simple, zero-order model for simulating the performance of an FDS-equipped vehicle was developed. Analysis and model simulations show that FDS can deliver precise, computer-controlled boost ratio of up to 1.8 (about 11 psi or 75 kPa at a sea level) with instant response to demand.

The FDS requirements for compressed air were shown to be modest and that much of the compressed air can be generated using recovery of kinetic energy during vehicle deceleration. Results indicate that for economical operation FDS boost ratio should not exceed 1.5 and the pump suction pressure should be kept as close to ambient as possible. FDS can be compact, simple, economical, and it does not require intake air intercooler. Simulations predict that even with a downsized engine, an FDS-equipped automobile could be “fun-to-drive.” As shown in Figure 24, FDS attributes compare favorably with traditional supercharging technologies. Planned future work will compare the overall efficiency of FDS-equipped vehicle to traditional technologies.

The figure also shows individual boost profiles for each TC and FDS system. Air temperature effects were ignored in this simulation. The simulation starts at engine speed of 1000 rpm. Being independent of engine speed, the combined TC–FDS delivers an instant response to boost demand. In practice, TC and FDS can be installed in series, preferably with the FDS downstream of the TC. Compressed air consumption in a combined TC-FDS is very low because the FDS is used only for a very short time, typically less than 2 seconds. If the turbocharged engine already has a compressed air system (e.g., for vehicle brakes), such a system can be adapted for FDS operation.

Figure 23: Pressure boost ratio versus engine speed for a combined TC-FDS system showing the contributions of each FDS and TC

Figure 24: Comparative performances of several supercharging techniques

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

CFD - Computational fluid dynamic
FD - Fluid-dynamic
FDS - Fluid-dynamic supercharger
GM - General Motors
LNA - Large naturally aspirated
NA - Natural aspiration / naturally aspirated
MS - Mechanical supercharger
rpm - revolutions per minute
SC - Supercharging / supercharged
TC - Turbocharger / turbocharged
VAN - Variable area nozzle
VAD - Variable area diffuser
VGT - Variable geometry turbine