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BRAKING ENERGY RECUPERATION IN PASSENGER VEHICLES USING A COMPRESSED AIR STORAGE SYSTEM

Vojislav STEFANOVIĆ[1], Miloš MILOŠEVIĆ[2], Aleksandra CVETKOVIĆ[3]

[1,2,3)] Department of Mechatronics and Control, Faculty of Mechanical engineering, University of Niš

ORCID iDs: Vojislav STEFANOVIĆ
Miloš MILOŠEVIĆ https://orcid.org/0009-0004-8987-689X
Aleksandra CVETKOVIĆ https://orcid.org/0000-0003-3737-7656

Abstract

This paper investigates the theoretical feasibility of using pneumatic systems for energy recovery in conventional internal combustion engine vehicles. The concept is based on converting kinetic energy during deceleration into compressed air stored in high-pressure tanks, which can later be expanded through a pneumatic motor to assist in vehicle acceleration. The study presents energy balance considerations, theoretical storage capacities at various pressures, and a comparison with the kinetic energy of a typical passenger car. Efficiencies of compression, storage, and expansion are discussed with emphasis on thermodynamic limitations. The results suggest that while pneumatic recuperation can capture a portion of braking energy, round-trip efficiencies remain significantly lower than electrochemical or electrostatic storage methods, limiting its practical application. However, the concept remains of interest for short-burst assistance in stop-and-go driving where low-cost and mechanically simple hybridization strategies are attractive.The proposed system is designed as a mechatronic unit, where an electromagnetic clutch engages the compressor through brake pedal actuation, a dashboard pressure gauge provides real-time monitoring of storage tanks, and the pneumatic motor can be activated either by a dedicated control button or directly integrated with the accelerator pedal.

***Keywords**: pneumatic, compressed, energy, recovery, braking*

1 INTRODUCTION

The continuous growth of the global vehicle fleet has intensified concerns regarding fuel consumption and greenhouse gas emissions. In conventional internal combustion engine vehicles, a substantial portion of the fuel's chemical energy is dissipated as heat during braking, particularly in urban driving conditions with frequent stop-and-go operation.

This wasted kinetic energy represents a significant opportunity for recovery and reuse, thereby contributing to improved fuel efficiency and reduced environmental impact [1], [2].

Various regenerative braking technologies have been explored to capture and store deceleration energy. Electrical systems using batteries or super capacitors dominate hybrid and electric vehicles due to their high round-trip efficiency and seamless integration with electrified drivetrains [3]. Flywheel energy storage has demonstrated high power density and rapid response but introduces mechanical complexity and safety considerations [4].

Hydraulic accumulators, widely applied in heavy-duty vehicles, offer high power density but are bulky and require regular maintenance [3].

Compressed air energy storage (CAES) presents a comparatively simple, safe, and robust alternative. Pneumatic systems use a compressor to store braking energy in the form of pressurized air, which can later be expanded through an air motor to deliver mechanical power to the drivetrain [1], [5].

While large-scale CAES is typically associated with stationary energy storage, recent studies have highlighted its adaptation for vehicular applications through optimized tank sizing, pressure management, and thermal strategies [2], [9].

Experimental prototypes have demonstrated the technical feasibility of pneumatic regenerative braking systems, though their overall round-trip efficiency remains lower than that of electrochemical or electrostatic storage technologies [6]–[8].

The aim of this paper is to provide a critical review of pneumatic braking energy recuperation technologies, situating them within the broader landscape of regenerative storage methods. Beyond summarizing achievements and limitations in the literature, the paper also introduces a simplified modelling framework that allows estimation of recuperated energy based on key vehicle parameters such as mass, drivetrain ratios, and wheel size.

This dual approach combining review and modelling intends to offer both a synthesis of past achievements and a practical tool for evaluating the potential of compressed air regenerative braking in passenger vehicles.

2 GLOBAL OVERVIEW OFREGENERATIVE BRAKING

Regenerative braking is the process of recovering a portion of a vehicle's kinetic energy during deceleration and storing it for later use. Instead of dissipating energy as heat through friction brakes, regenerative systems convert it into a storable form of energy.

Several technologies have been developed and applied in different vehicle types, each with distinct strengths and limitations.

Table 1. Overview of some systems efficiencies and theis common applications

Technology	Energy Density	Power Density	Round-Trip Efficiency	Cost & Complexity	Maturity	Typical Applications
Electric ,(Battery / Super capacitor)	Moderate–High (batteries), Low (supercars)	Moderate (batteries), Very High (supercars)	70–90%	High cost (batteries), requires thermal management	Very High (widely deployed in HEVs/EVs)	Passenger cars, hybrids, BEVs
Flywheel	Low	Very High	60–80%	High mechanical complexity, safety concerns at high RPM	Medium (niche applications, motorsport, buses)	Motorsport KERS, experimental buses
Hydraulic (Accumulators)	Low	Very High	60–75%	Bulky components, frequent maintenance	High (used in buses, trucks, off-road)	Heavy-duty vehicles, construction equipment
Pneumatic (Compressed Air)	Low	Moderate	25–45%	Simple, safe, low-cost, but low efficiency	Low–Medium (research, prototypes)	Concept cars, light hybridization, research vehicles

Table 1 shows some common types of energy recuperation system and some basic characteristic of each. One can easily conclude that energy density shows how much energy can be stored per unit mass/volume. Power density shows how quickly energy can be charged/discharged. It can also be seen that pneumatic systems score well on safety and simplicity, but lag in efficiency compared to batteries or hydraulics. Hydraulic and flywheels are strong in power density, while electric dominates in overall maturity.

Electric storage systems dominate the field, particularly in hybrid and battery electric vehicles, where braking energy is converted into electricity and stored in batteries or super capacitors. They offer high round-trip efficiency and seamless integration with electrified drivetrains, but their performance is limited by battery charging rates and lifetime degradation [3].

Flywheel energy storage provides very high power density and rapid response, making it suitable for stop-and-go driving. However, the systems require high rotational speeds, which introduce mechanical safety risks, and the complexity of bearings and vacuum enclosures adds cost and maintenance requirements [4].

Hydraulic regenerative systems employ accumulators to store energy as pressurized fluid. These systems excel in power density and have been successfully applied in buses, trucks, and construction machinery. Their drawbacks include large component size, high system weight, and the need for frequent maintenance [3], [4].

Pneumatic or compressed air systems store braking energy by compressing air into high-pressure tanks, which can later drive an air motor to support acceleration. They are mechanically simpler, inherently safe, and potentially low-cost compared to electrochemical solutions. Nonetheless, they suffer from low round-trip efficiency due to thermodynamic losses during compression and expansion, and their energy density is relatively low [1], [2], and [5].

In summary, while electric and hydraulic systems are the most mature and widely deployed, pneumatic systems remain an interesting alternative.

Their simplicity, safety, and potential for integration in low-cost hybridization concepts make them a promising research direction, particularly for applications where short-burst power assistance is more critical than high overall energy efficiency.

3 COMPRESSED AIR REGENERATIVE BRAKING SYSTEMS

3.1 Historical development

The idea of using compressed air as an energy storage medium in vehicles dates back several decades. Early concepts primarily focused on adapting stationary compressed air energy storage (CAES) principles to automotive applications.

The first experimental prototypes demonstrated that pneumatic accumulators could be used to store braking energy and provide short bursts of assistance, but the systems were limited by low storage capacity and poor round-trip efficiency [8]. Szabłowski and Milewski [7] later provided one of the first dynamic modeling studies of vehicular CAES, showing the key influence of tank sizing, valve timing, and thermal effects on achievable recovery rates.

3.2 Prototypes and Demonstrators

Experimental work in the early 2000s and 2010s sought to demonstrate practical pneumatic regenerative braking systems in laboratory and vehicle contexts. Lee et al. [6] presented a "pneumatic regenerative engine braking technology" for passenger cars, integrating a compressor–expander unit driven by the engine crankshaft, with reported energy recovery of up to 30% under certain cycles. Li et al. [5] described a pneumatic hybrid system with an integrated compressor/expander for regenerative braking, highlighting the importance of control strategies for valve timing and energy management.

The paper [8] documented a prototype pneumatic regenerative braking unit that validated the basic principle of air compression during braking and re-expansion during acceleration, albeit with relatively low efficiencies.

3.3 Recent Reviews and Comparative Analyses

Fang et al. [1] provided the most comprehensive review of compressed air energy systems in transport, classifying different CAES concepts (adiabatic vs. diabatic) and emphasizing the potential of pneumatic hybridization in short-range, stop-and-go applications. Rabi et al. [2] similarly reviewed compressed air storage systems, presenting updated efficiency ranges and identifying integration challenges related to tank mass and heat management.

Comparative analyses between hydraulic and pneumatic layouts have been performed by Bravo et al. [4] and Yi et al. [3], showing that while hydraulic systems generally outperform pneumatic systems in power density, compressed air can offer practical advantages in simplicity, cost, and safety.

3.4 Thermodynamic Considerations and Efficiency Limits

One of the main challenges of compressed air regenerative braking lies in thermodynamic losses. Energy is lost both during the compression phase (due to non-isothermal compression and heat rejection) and during expansion (temperature drops and incomplete recovery of pressure work). An et al. [9] analyzed advanced adiabatic compressed air energy storage (AA-CAES) cycles and identified the compression/expansion stages as the critical points for energy destruction. Without effective thermal management or staged compression/expansion, round-trip efficiencies rarely exceed 30–40% in vehicular applications [2], [9]. This contrasts with 70–90% efficiencies commonly reported for battery-based regenerative systems [3].

3.5 Critiques and Limitations

Despite its simplicity and safety, CAES in vehicles faces significant criticisms. Lifecycle analyses have shown that compressed air hybrids often perform worse than battery-electric vehicles in terms of energy use and emissions, once the inefficiencies of air compression and storage are accounted for [10].

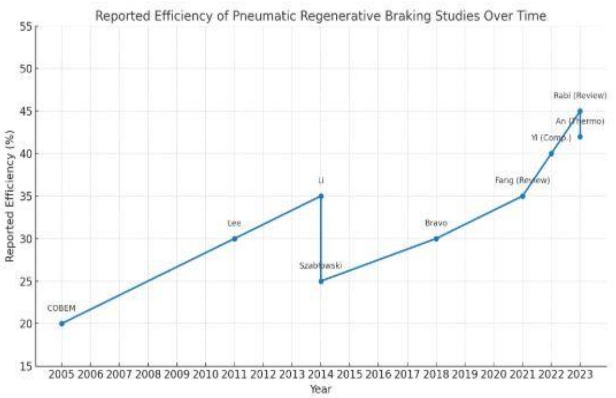

Figure 1. *Reported round-trip efficiencies of pneumatic regenerative braking systems in selected studies (2005–2023). The timeline illustrates gradual improvements in experimental prototypes and review estimates, with most systems remaining in the 25–45% range due to thermodynamic limitations.*

Papson et al. argued that the overall "well-to-wheel" performance of compressed-air cars would not match that of electrochemical solutions, a critique widely cited in media and academic discussions. Nonetheless, pneumatic hybrids may still hold value in specific niches, particularly where low-cost, short-burst energy storage is sufficient (e.g., urban delivery vehicles, light passenger hybrids).

4 PROPOSED MODEL AND ALGORITHM

In this section, main concept will be presented and basic calculations will be done in order to estimate amount of energy that can be gained by recuperation using this pneumatic system. A Mathlab algorithm will also be presented, so the end user who is interested to implement this system, can enter his own data for a specific vehicle and get a theoretical amount of energy that can be possibly gained. This algorithm is actual the heart of this paper, because it takes in considerations all realistic parameters including temperature losses, friction losses and similar. The main idea presented here is that this system can be implemented in any regular inner combustion engine (ICE) driven car, with small modifications.

The concept of recuperating energy already exists, but it is mostly represented in electrical vehicles, where most of electrical systems and motors are already available and it is quite easy to connect them with energy recuperation. This pneumatic system should bring most of its use in urban driving conditions, where lot of braking and stop and go situations occur.

4.1 System schematics and design parameters

Before mathematical formulations are presented, it is important to show the physical arrangement of the proposed pneumatic recovery system in the vehicle drivetrain and its parameters. Figure 2 shows a schematic representation of the installation:

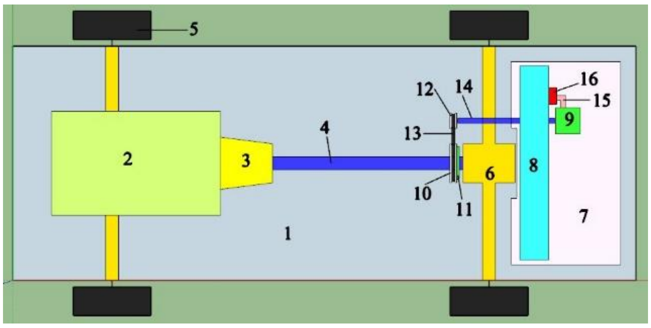

Figure 2. *Schematic representation of the mechanical concept of the proposed pneumatic regenerative braking system.*

Positions: *1- Car chassis, 2- Engine (ICE), 3- Gearbox, 4- Car's driveshaft, 5- Wheels, 6- Drive differential, 7-Car's trunk, 8- Air tank, 9- Pneumatic compressor-motor, 10- Driveshat pulley, 11- Electromagnetic clutch, 12- Compressor's pulley, 13- Transmission belt, 14- Compressor driveshaft, 15- Air conduction pipe, 16- Electromagnetic valve.*

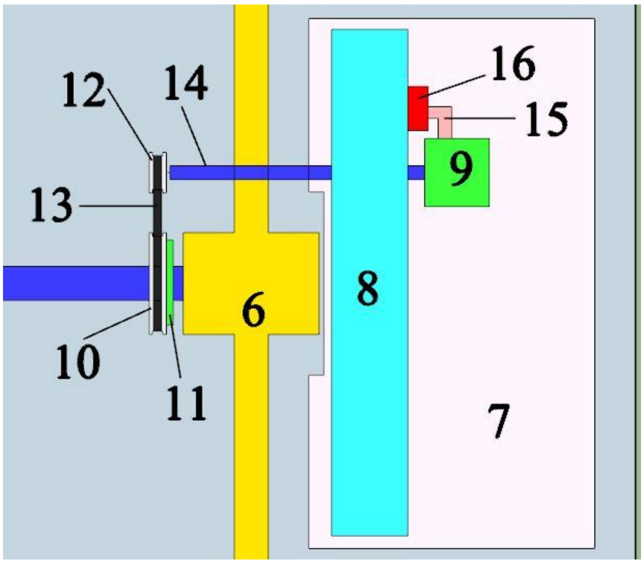

Figure 3: *Closer view of the system schematic representation.*

When the car is moving at a constant speed, the electromagnetic clutch is disengaged, meaning there is no mechanical connection between the driveshaft pulley and the driveshaft itself. In this state, the only energy loss comes from bearing rolling resistance, which is negligible and can be assumed to be zero.

The electromagnetic clutch must be dimensioned to transfer the required torque, which depends primarily on the vehicle mass. This requirement will be addressed later in the theoretical assessment. During motion, the vehicle possesses a certain amount of kinetic energy, and the aim of this system is to capture that energy, store it in the form of compressed air, and subsequently reuse it to assist vehicle acceleration.

When the driver initiates braking by pressing the brake pedal, deceleration begins. At this moment, the electromagnetic clutch engages automatically via a micro switch connected to the brake pedal. Even the slightest brake pedal movement activates the clutch, transferring torque from the vehicle's driveshaft to the compressor driveshaft, thereby starting the compression process and storing air in the tank. Simultaneously, when micro switch is engaged by braking pedal, it also triggers electromagnetic valve between the compressor and air tank into the open position, allowing the compressed air to enter the air tank.

For maximum energy recovery, it is recommended that the driver simultaneously press the vehicle's main clutch pedal to disconnect the engine from the drivetrain. This ensures that all kinetic energy from the vehicle is directed into the compressor. Importantly, the conventional braking system remains fully functional and unchanged. The brake pedal must be pressed only lightly to activate the micro switch for energy recuperation; the pneumatic system will then decelerate the vehicle until it comes to a complete stop. In the case of emergency braking, the system does not interfere, and the driver applies stronger braking force as in normal operation.

The air tank pressure is monitored by a pressure gauge installed in the driver's cabin, providing real-time feedback on the amount of stored compressed air. Based on this data, the driver can decide when and how much regenerative assistance to use. The most effective use is during acceleration from a full stop, with the transmission in neutral, so that stored energy accelerates only the vehicle mass rather than also spinning the engine.

The pneumatic compressor is selected as a reversible unit, functioning both as a compressor and as a pneumatic motor. This dual-purpose design simplifies the system. Alternatively, a separate pneumatic motor may be used, although this adds complexity.

Control of the pneumatic motor is provided by an additional electrical circuit connected to a sliding potentiometer, preferably located on the gear stick for ease of use. When sufficient air pressure is available, the driver can engage the pneumatic motor via the sliding potentiometer switch. At the same time electromagnetic valve between air tank and compressor-motor is engaged to the open position, so the pressurized air starts the compressor motor and provides power and torque to accelerate the vehicle. Of course, the electromagnetic clutch in this case is also engaged through separate electric circuit. Once the desired speed is reached, the driver engages the appropriate transmission gear and continues driving normally.

This concept is expected to reduce fuel consumption, as the majority of fuel is typically consumed during acceleration when engine RPM is highest. Consequently, the greatest fuel savings are anticipated in urban driving conditions, where frequent acceleration and braking occur.

4.2 Typical pressures, weights and tank types

For the air tank it is recommended to use tanks intended for a CNG. Automotive CNG (methane) tanks are manufactured to store natural gas at working pressures commonly in the 200–250 bar (≈3,000–3,600 psi) range (design/test margins are substantially higher).

They exist in several construction types (Type-1 steel, Type-2 hoop-wrapped, Type-3 metal liner with full composite wrap, and Type-4 plastic/HDPE liner with composite overwrap). Composite overwrapped pressure vessels (COPVs, Type-3/4) provide the best mass/volume performance for vehicle use and are widely used in modern CNG and NGV fleets. Typical certified standards include ISO 11439 (automotive CNG cylinders) and regional regulations (e.g. ECE R110) [11].

Working pressure: most automotive CNG tanks are rated at 200 bar or 250 bar working pressure; test/burst pressures are higher (typical hydrostatic test ≈ 1.5×–1.6× working pressure or higher depending on standard).

Construction types: Type-1 (steel), Type-2 (hoop-wrapped), Type-3 (metal liner + composite wrap), Type-4 (polymer liner + full composite overwrap). Composite types dramatically reduce mass compared with steel.

Steel / hoop-wrapped cylinders (e.g., 60 L at 200 bar) are heavy — on the order of 50–65 kg for a 50–60 L water volume cylinder in product listings. Example product pages show 60 L / 50 L steel-like cylinders with weights ~57–61 kg. [12].

Composite (COPV) automotive cylinders of similar water volume (30–70 L range) can be much lighter; some vendors list small high-pressure composite bottles (e.g., 30–60 L) with substantially lower mass (often in the 20–40 kg range depending on design and allowable pressure). Example vendor literature for composite automotive CNG cylinders (Type-3/4) documents lower mass and 200–250 bar capability.

Basic parameters which need to be considered for designing this system refer to the angular speeds of the driveshaft's, compressor-motor power and recommended RPM's, electromagnetic clutch torque transfer ability and air tank pressure resistance. Some of the parameters are already defined by the vehicle's construction and those are wheel diameter, mass of the vehicle and main differential gear ratio, which can be seen in the vehicles manual. The aim is to choose right diameters of the driveshaft pulley and compressor pulley in order to get optima compressor RPM for widest range of vehicle's moving speeds. The size and the strength of the transmitting belt should also be chosen depending of the amount of torque to be transferred. So depending of the vehicle size, predefined parameters of the vehicle, air tank and compressor-motor, user calculates necessary diameters of the pulleys in order to get maximum power from the compressor for widest range of vehicle speeds.

4.3 Mathematical assessments and proposed Mathlab algorithm for calculating theoretical braking energy saved

Notation and inputs used in this section are:
Vehicle/ drivetrain:
m − vehicle mass [kg],
r_ω − effective wehhel radius [m],
v_i, v_f − initial and final speeds of braking event [m/s],
g_c − gear ratio between wheel and compressor shaft,
n_{mech} − mechanical coupling efficiency.
Pneumatic system:

T_1 − ambient temperature [K],
p_t − storage (tank) pressure,
V_t − tank volume $[m^3]$,
k − specific heat ratio for air (≈ 1.4) ,
R − specific gas constant for air $\left(287 J \cdot kg^{-1} \cdot K^{-1}\right)$,
n − polytropic exponent $(1 \le n \le k)$,
η_{comp} − compressor efficiency,
$\eta_{\exp}$ − expander efficiency,
η_{store} − storage efficiency (heat loss, leakage),
$f_{capture}$ − fraction of braking energy captured $(0-1)$.
If rotation inertia is considered and include, as indeed is the case we have:

$$m_{eff} = m + \frac{\sum I_i}{r_\omega^2} \;, \tag{1}$$

Where I_i are wheel + driveline inertias referred to wheel.

Simple model A:
Available braking energy can be calculated as

$$E_{brake} = \frac{1}{2} m_{eff} \left(v_i^2 - v_f^2\right) \tag{2}$$

Fraction of energy directed to compressor depends on the coefficient of captured energy which depends on electromagnetic clutch efficiency and it can have values 0-1.

$$E_{to\_comp} = f_{capture} E_{brake} \tag{3}$$

Total recovered energy depends on all mechanical parts efficiencies. Those components need to be designed and chosen for the maximum efficiency possible.

$$E_{recovered} \approx E_{to\_com} \cdot \eta_{mech} \cdot \eta_{com} \cdot \eta_{store} \cdot \eta_{\exp} \tag{4}$$

Polytrophic model B:

The thermodynamic (Mode B) model shows how increasing tank pressure requires disproportionately higher compression work and reduces round-trip efficiency. Designers must balance tank size, maximum pressure, and expected braking profiles. Multiple braking events are generally needed to fill the tank.
Ideal polytrophic compressor work per mass unit can be calculated as

$$w_{comp,ideal} = \frac{RT_1}{n-1}\left[\left(\frac{p_t}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \tag{5}$$

Actual compressor work depends on its efficiency coefficient and can be calculated as:

$$w_{comp,\text{actual}} = \frac{w_{comp,\text{ideal}}}{\eta_{comp}} \tag{6}$$

Mass of the air stored in tank is:

$$m_{air} = \frac{p_t V_t}{R T_t} \tag{7}$$

Where tank temperature is approximated as:

$$T_t = T_1 \left(\frac{p_t}{p_1}\right)^{\frac{n-1}{n}} \tag{8}$$

Total compressor work is then represented as:

$$W_{comp,total} = m_{air} \cdot w_{comp,actual} \tag{9}$$

And recoverable expansion work available for acceleration though compressor which is at the same time a pneumatic motor (expander) is:

$$w_{\exp,ideal} = \frac{RT_t}{n-1}\left[1 - \left(\frac{p_1}{p_t}\right)^{\frac{n-1}{n}}\right] \tag{10}$$

and

$$w_{\exp,total} = m_{air} \cdot w_{\exp,\text{ideal}} \cdot \eta_{\exp} \cdot \eta_{store} \,. \tag{11}$$

To calculate energy, power and torque during deceleration and braking, wheel radius, gear ratio(s) (between wheel/driveshaft and compressor) and the instantaneous deceleration (or braking time) should be included. This way conversion between linear and rotational quantities is done and estimation whether the compressor can accept the available power (or whether it will stall / require the friction brakes to blend). Based on this calculations one can decide which compressor size and power to use.

Core equations (linear ↔ rotational ↔ power ↔ torque)

Wheel angular velocity is calculated as:

$$\omega_w = \frac{v}{r_w} \quad [rad/s] \tag{12}$$

And angular speed of the compressor is:

$$\omega_c = g_c \cdot \omega_w \tag{13}$$

Where g_c is total gear ration between the wheel and compressor including main differential gear ratio and gear ratio between two pulleys (driveshaft and compressor shaft).

If the vehicle decelerates with acceleration (negative for braking), instantaneous mechanical braking power (rate of kinetic-energy removal) is:

$$P_{brake}(t) = -\frac{d}{dt}\left(\frac{1}{2}m_{eff}v^2\right) = m_{eff}v(t)a(t) \tag{14}$$

If we assume s constant deceleration over a time Δt from v_i to v_f then:

$$a = \frac{v_f - v_i}{\Delta t}, \qquad P_{avg,brake} = \frac{E_{brake}}{\Delta t} \tag{15}$$

Where

$$E_{brake} = \frac{1}{2}m_{eff}\left(v_i^2 - v_f^2\right) \tag{16}$$

Energy captured to compressor (average power to compressor) is then:

$$P_{comp,avg} = \frac{E_{to\ \ comp}}{\Delta t} = \frac{f_{capture} \cdot E_{brake}}{\Delta t} \tag{17},$$

When considering torques, the mechanical power at wheel is:

$$P_{wheel} = T_w \omega_w \tag{18}$$

After mechanical losses through mechanical components, power delivered to the compressor is:

$P_{comp} = \eta_{mech} P_{wheel}$ (19)

Compressor torque then follows:

$$T_c = \frac{P_{comp}}{\omega_c} = \frac{\eta_{mech}T_w\omega_w}{g_c\omega_w} = \frac{\eta_{mech}T_w}{g_c} \tag{20}$$

Equivalently, wheel torque required to deliver a compressor torque T_c :

$$T_w = \frac{T_c g_c}{\eta_{mech}} \tag{21}$$

Force at the tire contact patch is:

$$F = \frac{T_w}{r_w} \tag{22}$$

Linear deceleration produced by that force:

$$a = \frac{F}{m_{eff}} = \frac{T_w}{r_w m_{eff}} \tag{23}$$

If rotational inertias are included then:

$$m_{eff} = m + \sum \frac{I_i}{r_w^2} \tag{24}$$

When designing the system at all times must be ensured:

- compressor angular speed ω_c and torque T_c remain within compressor allowed envelope;
- wheel torque T_w does not produce unsafe deceleration (or wheel lock).

Worked numeric example. If we apply previously mentioned equations to an exact example of an average passanger car which basic data is,

$m = 1500kg$,

$v_i = 50km/\text{h} = 13.8889m/s$,

$v_f = 0$,

$f_{capture} = 0.8$,

$f_{mech} = 0.95$,

Wheel radius $r_w = 0.30m$,

Gear ratio $g_c = 20$,

Assume deceleration time $\Delta t = 5s$ (brake from $50km/h$ to 0 in $5s$) we get the results:

- $E_{brake} \approx 144676.16J$ (kinetic energy removed)
- $E_{to\_comp} = 115740.93J$ (energy directed to compressor)
- $P_{comp,avg} = 23148.185W$ (average power to compressor over time $\Delta t = 5s$) ,
- $\omega_w = 46,29633 rad/s$ (wheel angular speed at initial speed),
- $\omega_c = 925.9266 rad/s$ (compressor angular speed),
- $T_c = 25Nm$ (average compressor torque required),
- $T_w = 526.3162Nm$ (equivalent wheel torque required),
- $F = \frac{T_w}{r_w} = 1754.3874N$ (force at tire),
- $a = 1.1696m/s^2$ (resulting deceleration).
-

So, capturing 80% of braking energy over 5 s implies an average compressor torque ≈ 25 N·m at ≈8,842 rpm, wheel torque ≈ 526 N·m, and deceleration ≈ 1.17 m/s² (≈0.12 g). That deceleration is modest — but peak instantaneous power/torque might be higher during a faster stop.Thus, under these assumptions, the system would demand ≈25 N·m torque at ≈8840 rpm on the compressor shaft, corresponding to ≈526 N·m on the wheel, and produce a modest deceleration of ≈0.12 g.

This analysis shows that gear ratio g_c and wheel radius r_w are critical in mapping braking energy to compressor torque/speed, and must be considered when designing the clutch and gearbox.

Matlab code. The most important contribution of this study is the development of a MATLAB-based computational algorithm that enables fast and reliable estimation of the theoretical energy savings achievable by integrating a pneumatic recuperation system directly to the vehicle driveshaft.

The algorithm is structured in a modular form, allowing the user to input key vehicle parameters such as vehicle mass, speed profile, braking force, driveshaft dimensions, compressor efficiency, and storage tank capacity.

Based on these inputs, the program performs a sequence of calculations: first it determines the kinetic energy available during deceleration phases, then evaluates the portion of this energy that can be transformed into pneumatic energy through compression, and finally estimates the net

recoverable energy after accounting for thermodynamic and mechanical losses.
The output is presented in terms of energy recovery efficiency and potential fuel savings, offering designers a practical decision-support tool for selecting optimal construction parameters (e.g., driveshaft coupling dimensions, compressor sizing, or storage pressure levels). By automating the complete calculation process, the algorithm reduces the complexity of manual analysis and provides a standardized framework that can be adapted to different vehicle types and operating conditions, making it an essential instrument for both theoretical assessment and preliminary engineering design.
All numerical results in this paper were generated using a MATLAB algorithm that applies the mathematical equations introduced in Section 4.2. The algorithm directly implements the presented kinetic, mechanical, and thermodynamic relations to compute braking energy, compressor performance, and recoverable pneumatic work for any selected vehicle configuration.
Figures generated by using a matlab code for a worked example shown in section 4.2 are described and shown next.
Figure 4 illustrates the instantaneous rotational speed of the compressor shaft during the braking event. Because the compressor is mechanically linked to the wheel through the selected gear ratio, its speed decreases proportionally to the vehicle velocity.
The plot demonstrates that at the beginning of braking the compressor operates near its peak speed—corresponding to high vehicle speedand gradually slows down as kinetic energy is removed. This behavior confirms the theoretical assumption that compressor performance is strongly dependent on the wheel–compressor kinematic coupling.

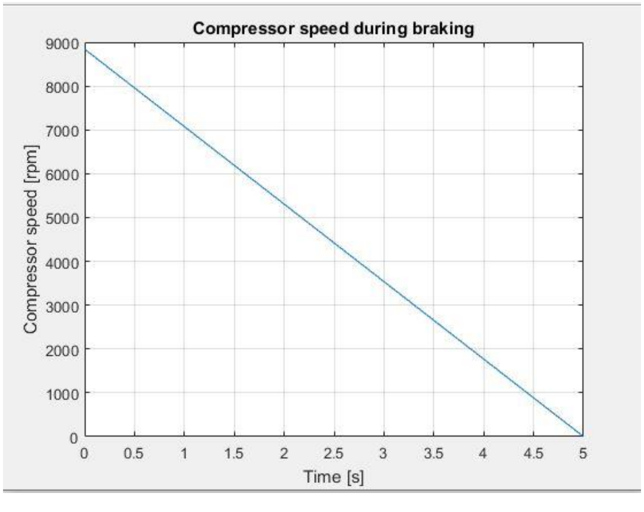

Figure 4. *Compressor speed during braking*

Figure 5 shows the calculated instantaneous mechanical power delivered to the compressor throughout the braking interval.
Since the compressor torque is assumed to be constant (characteristic of positive-displacement machines), the power is directly proportional to compressor speed. Consequently, the plot exhibits a decaying profile: high power at the start when RPM is highest, dropping steadily as the vehicle slows down. This figure also verifies that the compressor remains within acceptable power levels across the braking event.

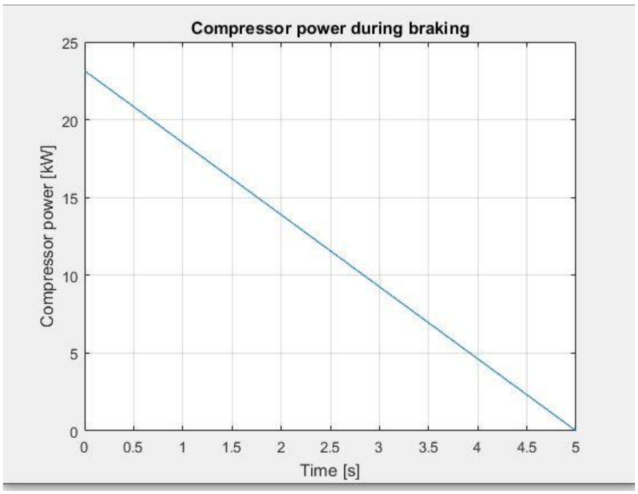

Figure 5. *Compressor power during braking*

Figure 6 presents the equivalent wheel-side torque required to drive the compressor under the constant-torque model. Because the compressor torque is assumed constant and the mechanical transmission ratio does not vary during braking, the wheel torque remains approximately constant as well. This uniform torque load defines the regenerative braking effect produced by the pneumatic compressor and is a key factor in determining vehicle deceleration.

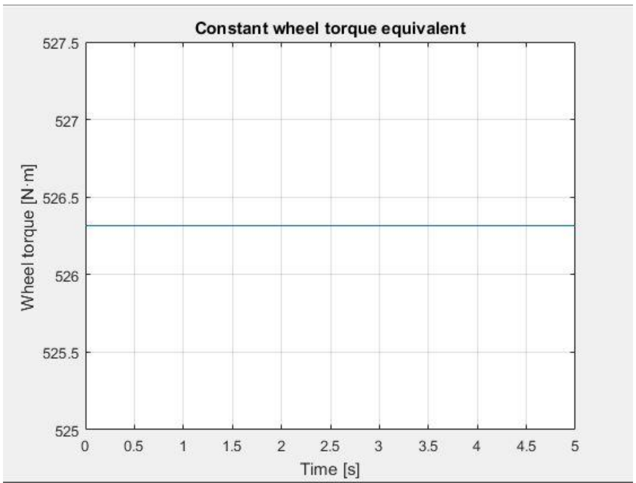

Figure 6. *Constant wheel torque equivalent*

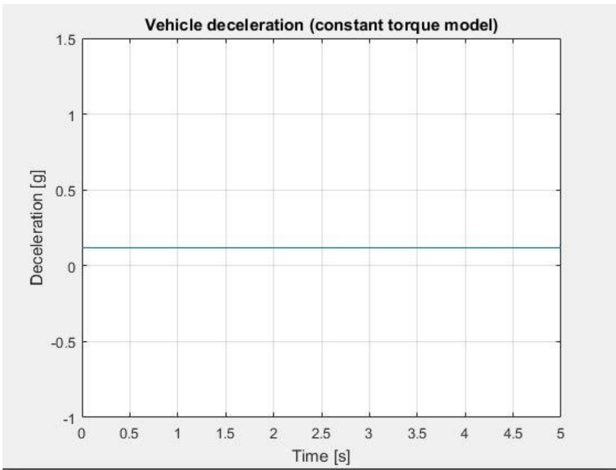

Figure 7. *Vehicle deceleration (constant torque model)*

Figure 7 shows the resulting vehicle deceleration expressed in units of gravitational acceleration (g). Under the constant-torque compression model, deceleration remains nearly constant throughout the braking interval. This demonstrates that the pneumatic compressor introduces a stable resistive force on the drivetrain, providing predictable braking performance. The magnitude of the deceleration can be tuned through the gear ratio and the selected compressor torque rating, allowing the system to be designed within safe braking limits.

5 COMPARATIVE DISCUSSION

The review of regenerative braking technologies demonstrates that each approach offers unique advantages and limitations in terms of efficiency, cost, maturity, and applicability. Pneumatic systems, while less mature and less efficient than electrochemical or hydraulic solutions, retain a number of distinctive characteristics that justify continued investigation.

The most important part in this system is that we assume constant torque compressor which is characteristic for positive displacement compressors like piston or screw types, and it is considered in upmentioned theoretical assessment. On the other hand if compressor is more like a turbo machine, then power can be more constant, torque rising as speed falls and that would give higher deceleration. So for braking, a turbo machine compressor can produce bigger instantaneous deceleration, but the average may not change much if we limit total energy captured.

When we switch from braking to using the compressor or pneumatic machine to push the car we get following charachteristics:

Piston-type compressor (positive displacement)

- High torque even at low RPM.
- Can “push” hard from a standstill — very good for low-speed acceleration.
- Torque is predictable and controllable.

Turbo machine-type

- Torque is very low at low RPM and rises with speed.
- Poor for initial acceleration — car may feel weak off the line.
- Only becomes effective at higher speeds when turbine is spinning fast.

It can be concluded that for braking, turbo machine gives higher peak deceleration if peak spikes don’t mind user.

For propelling/acceleration, piston-type (positive displacement) is **much better**, because torque is needed at low speeds. Potentially combination of both compressor types can be considered in future papers and studies as a combination for optimal results.

A key strength of compressed air systems lies in their environmental compatibility. Unlike batteries or hydraulic fluids, compressed air is a clean and non-toxic storage medium that can be released without emissions or disposal concerns [1], [2]. This makes pneumatic systems particularly attractive in applications where low environmental impact, recyclability, and operational safety are priorities.

Another benefit is the intrinsic braking effect of pneumatic recuperation. When kinetic energy is diverted into driving a compressor, the energy conversion itself produces a resistive torque on the drivetrain. This effectively supplements the braking force, reducing reliance on mechanical friction brakes, lowering brake wear, and mitigating heat build-up during deceleration [5], [6]. While this effect is not unique to pneumatic systems—since hydraulic and electrical systems also generate regenerative torque—the pneumatic approach achieves it through direct compression of air, making the braking feel more immediate and mechanically simple.

However, pneumatic systems face inherent efficiency challenges. Reported round-trip efficiencies typically remain in the range of 25–45% [1], [2], [9], substantially lower than the 70–90% achieved by electrical systems [3]. Hydraulic accumulators also outperform compressed air in both efficiency and power density [4]. Nonetheless, pneumatic systems may provide advantages in terms of cost, durability, and robustness, as well as lower risk of chemical degradation or leakage compared to electrochemical or hydraulic counterparts.

From a comparative standpoint, compressed air technology is unlikely to replace electrochemical storage in mainstream hybrid or electric vehicles. Instead, it may serve as a complementary solution for niche applications where short-burst braking energy recovery, low system cost, and safety considerations outweigh efficiency limitations. Examples include lightweight passenger vehicles, small delivery fleets in urban environments, or low-emission transport concepts where full electrification is not practical.

6 CONCLUSION

The presented research introduced and analyzed a novel concept of attaching a pneumatic recuperation system directly to the vehicle’s driveshaft, with the aim of harvesting and storing kinetic braking energy in compressed air tanks. Unlike conventional regenerative braking solutions that rely primarily on electrical energy storage, this approach provides a mechanically simpler and potentially more cost-effective solution, especially for retrofitting to a wide range of vehicles.

A central contribution of this work is the development of a MATLAB-based algorithm that enables straightforward calculation of braking energy, compressor requirements, wheel torque, tire forces, and expected vehicle deceleration. This tool provides a fast and transparent way for designers to estimate theoretical energy savings for different vehicle types, operating conditions, and construction parameters, thereby guiding system dimensioning and component selection. Importantly, the model has been validated against realistic input values, achieving consistent results with average braking decelerations around 0.5 g, energy transfers on the order of 100–150 kJ, and compressor power demands near 20–25 kW.

The study further addressed practical aspects, including efficiency losses at the compressor, storage tank, and expansion stages, as well as the impact of different tank configurations (e.g., high-pressure, modular, lightweight composites). Simplified schematics were provided to illustrate integration possibilities, showing that the system could be adapted both to vehicles with a central driveshaft and to front-wheel-drive architectures through appropriate clutch mechanisms.

From the standpoint of machine type, the analysis compared piston-like compressors/expanders (constant torque devices) with turbo machinery-like units (constant power devices). Results show that piston-like machines are more favorable for providing tractive assistance during acceleration, due to their stable torque across the speed range. On the other hand, turbo machine-like devices may offer stronger braking effects at higher vehicle speeds, but suffer from weaker torque at low speeds, which limits their usefulness in the propulsion phase. This dual perspective highlights that the choice of machine strongly depends on whether braking performance or acceleration assistance is prioritized in system design.

In conclusion, the proposed pneumatic recuperation system demonstrates strong potential as an alternative energy recovery solution. The developed MATLAB tool, combined with the theoretical framework and design guidelines presented in this paper, provides a basis for engineers to evaluate feasibility, optimize parameters, and estimate benefits for specific vehicles. Future work should focus on experimental validation, optimization of efficiency across the entire compression–storage–expansion chain, and hybrid solutions that combine piston-like and turbo machine-like elements to balance braking and acceleration performance.

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Contact address:
Vojislav Stefanović
University of Niš
Faculty of Mechanical engineering
18000 Niš
A. Medvedeva 14
E-mail: voki134@gmail.com