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Dynamic Programming for Hybrid Pneumatic Vehicles

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Abstract— **Pneumatic hybridization of internal combustion engines may prove to be a viable and cost-efficient alternative to electric hybridization. This paper investigates the fuel consumption reduction that is possible with this rather new concept in combination with the well-known downsizing and supercharging method.**

Depending on the available hardware, pneumatic modes can be based on either two-stroke or four-stroke operation. Both configurations are investigated.

Similarly to electric hybrids, hybrid pneumatic engines have an energy buffer, namely the internal energy of the air pressure tank, that provides an additional degree of freedom for the propulsion system. This entails the necessity of an optimal supervisory control algorithm that chooses the mode of engine operation at every time instant of the drive cycle while guaranteeing charge sustenance. In this study, a deterministic dynamic programming algorithm is used to ensure the optimal use of the energy stored to minimize fuel consumption.

Obtained results show that the combination of engine downsizing and pneumatic hybridization yields a fuel consumption reduction of up to 34% for the MVEG-95 drive cycle. Additionally, the "turbo-lag" normally associated with heavy downsizing can be overcome with this concept by using pressurized air from the tank to supercharge the engine during the speed-up of the turbocharger.

A standard gasoline engine has been modified, and first measurements presented in this paper confirm the validity of the concept.

I. INTRODUCTION

In spite all of its well-known problems, the internal combustion engine (ICE) continues to be the most important means of automobile propulsion and is expected to remain so for at least another one or two decades. However, its fuel consumption and consequent CO_2 emissions are symptomatic of two major system-inherent problems. First, vehicle braking energy cannot be transformed back into chemical or other usable energy. Second, consumer demand for high maximum power leads to increased engine displacement. The mean efficiency of the propulsion system is consequently decreased, since internal combustion engines exhibit a low efficiency at part load. Automobile manufacturers are currently attempting to tackle these shortcomings by focusing on hybrid electric vehicles (HEVs). Some research groups [1], [2], [3], [4], however, have turned their minds to yet another alternative, namely the pneumatic hybridization of ICEs. The latter system's architecture is shown in Fig. 1.

Each cylinder of the ICE is connected via a fully variable charge valve to a shared air pressure tank. In vehicle braking phases with fuel cut-off, the engine can intake air and pump it into the pressure tank. Without the injection of fuel, the pressurized air can then be used again for starting or driving the vehicle in the so-called pneumatic motor mode. The pressurized air can also be used to boost the conventional engine combustion mode, thereby overcoming the turbo-lag in supercharged engines [5]. Shifting the operating point of the engine is also possible, using at least one cylinder in a conventional combustion manner at a high load and at least one cylinder in the pneumatic pump mode without any fuel injection.

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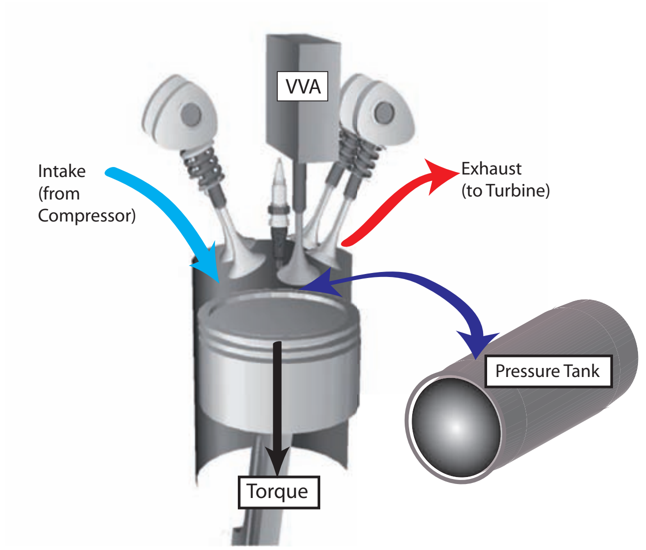

Fig. 1. Schematic hardware setup of a four-stroke hybrid pneumatic engine cylinder. Two-stroke pneumatic modes would additionally require a variable valve actuation (VVA) system for the intake and exhaust valves.

The hybrid pneumatic engine (HPE) was long thought to be a system in which *all* engine valves had to be actuated in a fully variable manner to allow for two-stroke pneumatic modes and still enable the four-stroke combustion mode. However, this setup adds complexity and cost. In [6], the idea was presented to stick to four-stroke cycles for both combustion and pneumatic operations. The intake and exhaust valves remain camshaft-driven and only the charge valve, which connects the cylinders to the pressure tank, is fully variable (see Figure 1).

The pneumatic hybridization of an internal combustion engine is possible for both spark ignition (SI) and compression ignition (CI) engines. The publications [3], [5] and [7] focus on SI engines, while [4], [8] and [9] look at CI engines.

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A. Why Pneumatic Hybridization?

To date, no one has managed to build a fully functional hybrid pneumatic vehicle (HPV). To motivate the realization of such a vehicle, fuel efficiency calculations have to be carried out. According to [4], [7], and [10], fuel gains in the range of 15% to 23% can be expected depending on the vehicles and drive cycles used. However, none of these calculations consider downsizing the engine to save even more fuel. Engine downsizing is a widely accepted method for reducing fuel consumption. To compensate for the lower rated engine power that comes with engine downsizing, the engine can be equipped with a turbocharger. Typically, a supercharged engine system responds slowly to a demanded torque step at low engine loads and speeds. This effect, known as "turbo-lag", is especially restrictive for downsized SI engines. However, an HPE can overcome this problem by injecting air from the pressure tank during the compression stroke and by injecting more fuel beforehand. Consequently, the torque increases almost instantaneously. This increases engine exhaust enthalpy, thereby propelling the turbine of the turbocharger. The accelerating compressor of the turbocharger leads to an increased intake pressure so that the additional air from the tank is needed only for a very short time. In [2] and [5], this engine mode is referred to as the "supercharged mode". Due to the presence of the turbo-lag problem, turbochargers are usually designed for achieving a good dynamic performance, which in turn compromises the engine's overall fuel efficiency. With the HPE concept, the turbocharger design can be entirely focused on efficiency, making engine downsizing an even more attractive option for fuel saving.

The part load problem of the ICE and its disability to recuperate braking energy can be overcome by both electric and pneumatic hybridization. HEVs do so by using an additional electric motor, battery packs, power electronics, and possibly an additional clutch. Unfortunately, the added weight of these components offsets part of the reduction in fuel consumption that they are designed to produce. In [11], the additional weight for an HEV with a hybridization ratio of 0.4 is about 170 kg for a vehicle base mass of 1503 kg and a power-to-weight ratio of 67 W/kg. By contrast, the HPE concept does not necessitate any heavy additional system components. The ICE is used for all the extra functionalities, removing the need for an extra compressor or an expansion device. The minimal additional weight of an HPE system comes mainly from the weight of the pressure tank and the valve actuation system. The additional weight for a pressure vessel and the hydraulic components is estimated to be just 25 kg. Furthermore, such a pressure tank is not subject to aging, a phenomenon that is a major problem for batteries used in HEVs [12].

Another major issue to be considered is added cost, especially when comparing pneumatic hybridization to electric hybridization. For HEVs, reliable battery packs are expensive, whereas the only significant cost-adding component for the HPE is the fully variable valve actuation system needed to achieve the different modes of operation. Such valve systems are significantly cheaper than battery packs and electric motors, especially if the number of fully variable actuated valves can be reduced, an issue that is addressed in this paper.

B. Approach

This paper focuses on the fuel consumption calculation of gasoline engine based HPVs using the well-known dynamic programming (DP) technique.

Four aspects of the HPE concept are investigated with regard to their effect on the vehicle's fuel consumption:

- Downsizing of the ICE, while keeping the rated power of the engine constant using a turbocharger.
- Pneumatic hybridization on the basis of fixed camshafts (FCS) with one fully variable charge valve per cylinder.
- Pneumatic hybridization on the basis of fully variable (FV) actuated intake, exhaust and charge valves.
- Removing the engine's camshafts without adding hybrid functionality (camless conventional engine).

The last aspect has to be investigated to evaluate the hybridization effect only on the basis of fully variable valves. The baseline engine is a 2.0 liter naturally aspirated engine. Additionally, downsized and supercharged engines with displaced volumes of $1.6, 1.4, 1.2$, and 1.0 liters are investigated.

The paper is structured as follows: In section II, the mathematical models of the elements of all HPV configurations are introduced. In section III, the application of the DP method to the HPE concept for the fuel consumption calculation is shown. The fuel consumption calculation and results from a modified real engine are shown in section IV, and the conclusions are drawn in section V.

II. CONTROL ORIENTED MODELING OF HYBRID PNEUMATIC VEHICLES

The behavior of the HPV is modeled in a quasi-static way [10]. All signals go backwards from the wheels to the engine in a physical non-causal sense. The discretized model is described as

$$x_{k+1} = f(x_k, u_k, v_k, a_k, i_k), \tag{1}$$

where x_k represents the internal energy U_{tank} of the pressure tank, the control input u_k is the chosen engine mode, v_k is the vehicle speed, a_k is the vehicle acceleration, and i_k stands for the gear, each at the time instant k. Since the tank volume V_{tank} does not change, the internal energy is a function of the tank pressure only:

$$U_{tank} = \frac{p_{tank} \cdot V_{tank}}{\kappa - 1}, \tag{2}$$

where κ is the ratio of specific heats of air. With U_{tank} being the level variable of the vehicle's pneumatic energy buffer, it can be called the state of charge, analogously to its usage for HEVs. Of course, the state of charge is only relevant for the investigated hybrid configurations. The elements of the HPV model are described in the following subsections.

TABLE I
VEHICLE PARAMETERS

Parameter	Symbol	Value	Note
vehicle base mass	m_{veh0}	1450 kg	
ICE mass	m_{ice}	$V_d \cdot 67 \frac{kg}{l}$	
air tank mass	m_{tank}	25 kg	hybrid only
air tank volume	V_{tank}	30 l	hybrid only
gearbox efficiency	η_{gb}	0.93	
auxiliary power	P_{aux}	400 W	
rolling friction coefficients	c_{r0} c_{r1} c_{r2}	0.0065 $1.22 \cdot 10^{-5} \frac{s}{m}$ $2.34 \cdot 10^{-6} \frac{s^2}{m^2}$	
air resistance	$A_f \cdot c_d$	0.83	
wheel radius	r_w	0.31 m	
air density	ρ_{air}	$1.293 \frac{kg}{m^3}$	

A. Vehicle

For this investigation, a mid-size vehicle was chosen, whose parameters are shown in Table I. The total vehicle mass is

$$m_{veh} = m_{veh0} + m_{ice} + m_{tank}, \tag{3}$$

with m_{ice} depending on the displaced volume V_d of the engine as described in [11]. In order to calculate the signals that are relevant for the HPE, the input signals given by the drive cycle (vehicle speed v_{veh} and acceleration a_{veh}) have to be considered.

The force on the vehicle F_{veh} is determined by the vehicle acceleration force F_a, the air resistance F_{air}, and the rolling friction force F_f:

$$F_{veh} = F_a + F_{air} + F_f \tag{4}$$

with

$$F_a = m_{veh} \cdot a_{veh} \tag{5}$$

$$F_{air} = 0.5 \cdot c_d \cdot A_f \cdot \rho_{air} \cdot v_{veh}^2 \tag{6}$$

$$F_f = m_{veh} \cdot g \cdot (c_{r0} + c_{r1} \cdot v_{veh} + c_{r2} \cdot v_{veh}^2). \tag{7}$$

The rolling friction is a second-order polynomial fitted to measured data. Assuming a standard 5-speed gearbox with efficiency η_{gb}, F_{veh} and v_{veh} translate to a demanded engine rotational speed ω_e and a demanded engine torque T_e as follows:

$$\omega_e = \frac{v_{veh}}{r_w} \cdot \gamma(i) \tag{8}$$

$$T_e = \frac{F_{veh} \cdot r_w}{\gamma(i)} \cdot \eta_{gb}{}^{-sign(F_{veh})} + T_{VVA} + \frac{P_{aux}}{\omega_e} \tag{9}$$

where T_{VVA} is the torque demand, for instance for the hydraulic pump for a variable valve actuation (VVA) system, P_{aux} is the electric power needed for auxiliaries, r_w is the wheel radius, and $\gamma(i)$ is the gear ratio of the gear chosen. The gear switching strategy is a baseline vehicle configuration switching strategy. It is adopted for all downsized engines.

B. Internal Combustion Engine (ICE)

The conventional combustion mode of the ICE is modeled using a standard Willans approximation (see [13] for further details):

$$p_{me} = \frac{4 \cdot \pi \cdot T_e}{V_d} \approx e(\omega_e) \cdot p_{mf} - p_{me0}(\omega_e), \tag{10}$$

where p_{me} is the mean effective pressure, $e(\omega_e)$ is the internal efficiency dependent on engine speed, and $p_{me0}(\omega_e)$ is the drag mean effective pressure. The fuel mean effective pressure p_{mf} is defined as

$$p_{mf} = \frac{H_l \cdot m_f}{V_d}, \tag{11}$$

where $H_l = 42.6$ MJ/kg is the lower heating value of gasoline and m_f is the burned fuel mass during one combustion cycle.

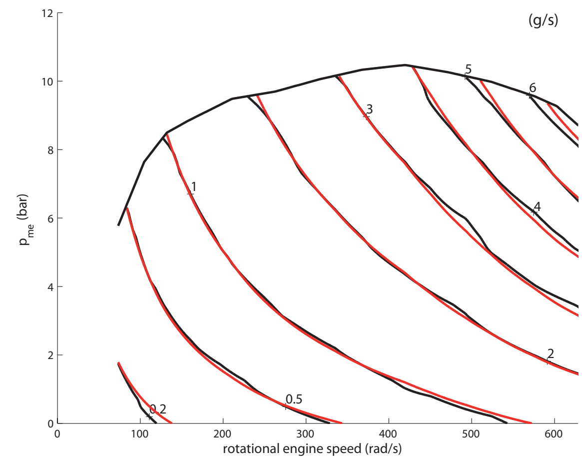

Fig. 2. Measured (black) and fitted (red) fuel consumption in g/s for a 1.6l naturally aspirated engine for loads up to 90%.

The Willans approximation works especially well if the engine is operated in a strictly stoichiometric way. To achieve peak torques, engines are often operated in rich conditions. Consequently, the Willans approximation in Fig. 2 for a 1.6 l naturally aspirated gasoline engine was carried out only for up to 90% of the peak load.

The drag mean effective pressure p_{me0} calculated from the Willans fit consists of three parts: the mechanical friction mean effective pressure p_{mfric}, the pumping losses that can be expressed as a mean effective gas transport pressure p_{mgas}, and other effects contributing to the drag torque such as heat losses, expressed as p_{mheat}:

$$p_{me0} = p_{mfric} + p_{mgas}(p_{me}) + p_{mheat} \tag{12}$$

with

$$p_{mfric} = r_0 + r_1 \cdot \omega_e + r_2 \cdot \omega_e^2. \tag{13}$$

The second-order polynomial (13) was fitted to measurement data, whose parameter values are listed together with the other engine variables in Table II.

C. Downsizing and Supercharging

The advantage of using a Willans approximation as shown in the previous subsection is that it can be easily scaled to other engine sizes. The baseline engine configuration is

TABLE II

BASE ENGINE PARAMETERS

Parameter	Symbol	Value
# cylinders	z	4
# intake valves per cylinder	z_{IV}	2
# exhaust valves per cylinder	z_{EV}	2
compression ratio	ε	10.5
idle speed	$\omega_{e,idle}$	$75\frac{rad}{s}$
min. engine speed for fuel cut-off	$\omega_{e,cutoff}$	$115\frac{rad}{s}$
mechanical friction coefficients	r_0 r_1 r_2	$3.7 \cdot 10^4$ $40.58\frac{s}{rad}$ $0.137\frac{s^2}{rad^2}$
max. power	P_{max}	99 kW

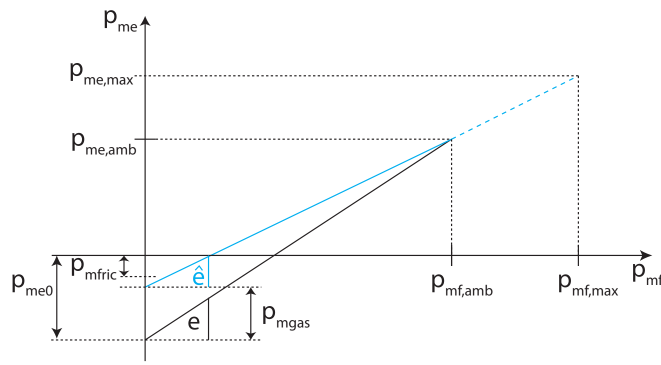

Fig. 3. Willans approximation for downsized and supercharged FCS engine configuration (black solid and blue dashed lines) and for downsized FV engine configuration (blue solid and blue dashed lines).

chosen to be a 2.0 liter naturally aspirated (NA) gasoline engine with standard fixed camshafts. The previously obtained Willans fit is also adapted for all turbocharged (TC) engines. The following modeling assumptions regarding engine scaling are made:

- The mean pressures p_{mfric}, p_{mheat} and p_{mgas} remain constant for all engine sizes.
- The internal efficiency $e(\omega_e)$ depends on the compression ratio ε of the engine as follows:

$$e(\omega_e) = k(\varepsilon) \cdot e_{\varepsilon=10.5}(\omega_e), \tag{14}$$

  where $k(\varepsilon)$ was determined using an engine process simulation. It is listed for all engine sizes in Table III.
- All engines are operated in a strictly stoichiometric manner. The fuel mean effective pressure at ambient intake pressure $p_{mf,amb}(\omega_e)$ as depicted in Fig. 2 stays the same for all engines.
- The rated power of the downsized engines (compared against the baseline of 2.0 l) is at least equal to the power of the baseline engine due to the use of turbochargers.
- All turbochargers are designed for maximum efficiency, since high instantaneous torque can be provided using the supercharged mode. This means that the intake pressure level is assumed to be equal to the exhaust pressure level for intake pressures higher than 1 bar.

The last assumption directly indicates that for intake pressures higher than 1 bar, the internal efficiency e can be assumed to be equal to the internal efficiency of an unthrottled engine since there the intake and exhaust pressures coincide as well. The next subsection explains how the internal efficiency $\hat{e}$ of an unthrottled engine is modeled.

D. Camless Engine

The Willans parameters $e(\omega_e)$ and $p_{me0}(\omega_e)$ were identified for a throttled engine. For a dethrottled engine, however, it can be assumed that there are no pumping losses for the whole operating range. The maximal pumping losses for a throttled engine are assumed to be 0.8 bar. The pumping losses depend on the nominal p_{me} as follows for every engine size:

$$p_{gas}(p_{me}) = 0.8 \cdot \frac{p_{me,amb} - \min(p_{me}, p_{me,amb})}{p_{me,amb} + p_{me0}}. \tag{15}$$

The internal efficiency of an unthrottled engine $\hat{e}$ is calculated as follows:

$$\hat{e}(\omega_e) = \frac{p_{me,amb} + p_{me0} - p_{mgas}(p_{me} = -p_{me0})}{p_{mf,amb}}, \tag{16}$$

where $p_{mf,amb}$ is calculated from (10).

For a visualization of this approach see Fig. 3. The blue dashed line is valid for both FCS and FV configurations for intake pressures higher than 1 bar.

In addition to these considerations, it is assumed that an electro-hydraulic valve system is used for the valve actuation, so mechanical friction due to camshaft components friction has to be eliminated for all FV engine configurations. It is assumed that the mechanical friction of the engine is reduced by 25%. In turn, the energy necessary for the hydraulic power has to be taken into account for all fully variable valves, which is done in the next subsection. These adjustments are not shown in Fig. 3.

TABLE III

ENGINE COMPRESSION RATIOS

Engine Type	V_d	ε	$k(\varepsilon)$	Name
NA FCS, FV	2.0 l	10.5	1	NA20, FV20
TC FCS, FV	1.6 l	9.5	0.978	TC16, FV16
TC FCS, FV	1.4 l	9.5	0.978	TC14, FV14
TC FCS, FV	1.2 l	9.0	0.966	TC12, FV12
TC FCS, FV	1.0 l	9.0	0.966	TC10, FV10

E. The Electro-Hydraulic Valve System

For the pneumatic hybridization it is necessary to have a variable valve actuation (VVA) system for the charge valve. The VVA system can of course also be used to actuate the intake and exhaust valves, which leads to a completely camless engine. Here, the Bosch electro-hydraulic valve system (EHVS) [14] was considered. In Table IV, the EHVS

TABLE IV
DISPLACED HYDRAULIC VOLUME FOR ONE ACTUATION

Valve	EHVS Type	$V_{hyd,i}$
Intake Valves (IV)	V0.7	407 mm^3
Exhaust Valves (EV)	V0.7	385 mm^3
Charge Valve (CV)	V0.5	414 mm^3

types and the assumed displaced hydraulic volume V_{hyd} per one valve actuation are listed. The total hydraulic volume $V_{hyd,tot}$ to be displaced by the pump during two engine revolutions yields

$$V_{hyd,tot} = \sum_i n_i \cdot V_{hyd,i} \qquad i = IV, EV, CV \quad , \tag{17}$$

where n_i is the number of the respective EHVS actuated valves during two engine revolutions for all cylinders needed for the engine mode chosen. The additional torque at the crankshaft needed for the actuation of the hydraulic pump of the EHVS is calculated as follows:

$$T_{VVA} = \frac{V_{hyd,tot} \cdot p_{hyd}}{4 \cdot \pi} \cdot \frac{1}{\eta_{hyd}}, \tag{18}$$

where the mean efficiency of the hydraulic pump is assumed to be $\eta_{hyd} = 0.6$. The hydraulic pressure p_{hyd} is chosen such that it always ensures that the charge valve resists the gas force of the pressurized air in the tank. This is assumed to be guaranteed for $p_{hyd} = 8 \cdot p_{tank}$. Respecting the operating boundaries for the EHVS system $p_{hyd,min} = 50$ bar and $p_{hyd,max} = 200$ bar, the hydraulic pressure can thus be defined as

$$p_{hyd} = \min(\max(8 \cdot p_{tank}, p_{hyd,min}), p_{hyd,max}). \tag{19}$$

F. The Pneumatic Modes

The optimization of all pneumatic modes for two- and four-stroke based engines can be found in [15]. They are the basis of all actuation laws of the engine valves and the resulting enthalpy maps. The enthalpy transferred from or to the tank ΔH_{tank} thus is a function of the tank pressure p_{tank} and the demanded engine torque T_e.

The assumptions for the pneumatic modes are the following:

- Air is an ideal and calorically perfect gas.
- The valves are idealized by neglecting opening and closing times as well as flow restrictions.
- The mechanical friction and heat losses (p_{mfric} and p_{mheat}) considered for the pneumatic modes are the same losses as those that occur when dragging an internal combustion engine. The pumping losses, however, depend on the pneumatic operating mode and point, as detailed in [15].
- The air tank is modeled adiabatically. This assumption seems to be quite optimistic. However, since the tank can be heated using exhaust gas, as proposed in [1], for the sake of simplicity, it is assumed here that the heat losses equal the heat gains.

Since the tank is assumed to be adiabatic, the change in internal energy is equal to the enthalpy transferred:

$$\Delta U_{tank} = \Delta H_{tank}. \tag{20}$$

The two-stroke pneumatic modes require variable valve actuation for all cylinder valves, while the four-stroke pneumatic modes require one fully variable charge valve only. Fuel is cut off for all pneumatic modes.

1) The Two-stroke Pneumatic Motor Mode: This mode is characterized by one air power stroke and one exhaust stroke per revolution per cylinder. The air power stroke takes place during the downward motion of the piston, while pressurized air from the tank is injected into the cylinder. During the upward stroke, the air is exhausted either via the intake valves or via the exhaust valves.

2) The Two-stroke Pneumatic Pump Mode: For this mode, there is an upward compression stroke with air transfer to the pressure tank once cylinder pressure has reached tank pressure. The subsequent downstroke is an expansion stroke with an air intake phase once the cylinder pressure has reached ambient pressure. This happens at every engine revolution for each cylinder.

3) The Four-stroke Pneumatic Motor Mode: This mode works like a conventional combustion engine cycle with fuel cut-off. However, during the expansion stroke, pressurized air is injected to generate positive torque.

4) The Four-stroke Pneumatic Pump Mode: This mode works like a conventional combustion engine cycle with fuel cut-off as well. In this case, however, air is transferred during the compression stroke to the pressurized tank once cylinder pressure has reached tank pressure.

5) Cylinder Deactivation: For the FV configuration, it is possible to use engine cylinders as gas springs, where all cylinder valves remain closed.

G. The Recharge Modes

Similarly to HEVs, HPVs are also capable of moving the operating point of the internal combustion engine to a higher and thus more efficient operating point while the excess power is used to increase the state of charge of the energy buffer. For HPEs this is simply achieved by running some cylinders in a conventional combustion mode while the other cylinders are pumping air into the pressure tank. This method clearly favors an even split of working and energy-consuming cylinders since then the engine runs smoothly. Since it is assumed that all engines have four cylinders, the split is two-by-two.

Obviously, for the FCS configuration, the four-stroke pneumatic pump mode is used while the FV configuration uses the two-stroke pneumatic pump mode.

For the sake of simplicity, it is assumed here that it is best to run all cylinders at ambient intake pressure, since a lower intake pressure lowers both conventional engine efficiency and pump efficiency. The turbocharger is assumed to be inactive during recharge phases.

TABLE V

NUMBER OF CYLINDERS IN DIFFERENT OPERATION TYPES FOR EACH ENGINE MODE

u_j	Engine Mode Name	#Cyl ICE	#Cyl Deact.	#Cyl Pump	#Cyl PMot
1	STOP	0	0	0	0
2	**FV**: ICE4	4	0	0	0
3	**FV**: ICE2	2	2	0	0
4	**FV**: Recharge	2	0	2	0
5	**FV**: Pump4	0	0	4	0
6	**FV**: Pump2	0	2	2	0
7	**FV**: PMot4	0	0	0	4
8	**FV**: PMot2	0	2	0	2
9	**FCS**: ICE	4	0	0	0
10	**FCS**: Recharge	2	0	2	0
11	**FCS**: Pump	0	0	4	0
12	**FCS**: PMot	0	0	0	4

TABLE VI

CONFIGURATIONS INVESTIGATED: ALLOWED ENGINE MODES

Configuration	Name	Allowed u_j
FCS Downsized	**FCS A**	9
FCS Downsized Start/Stop	**FCS B**	1,9
FCS Downsized Start/Stop Hybrid	**FCS C**	1,9,11,12
FCS Downsized Start/Stop Hybrid Recharge	**FCS D**	1,9,10,11,12
FV Downsized	**FV A**	2
FV Downsized Start/Stop	**FV B**	1,2
FV Downsized Start/Stop Hybrid	**FV C**	1,2,5,7
FV Downsized Start/Stop Hybrid Recharge	**FV D**	1,2,4,5,7
FV Downsized Start/Stop Hybrid Recharge Cyl.Deactivation	**FV E**	1,2,3,4,5,6,7,8

III. DYNAMIC PROGRAMMING

Every HPE propulsion system has an additional degree of freedom compared with a conventional drive train, similarly to HEVs. This is due to the energy buffer represented by the internal energy of the pressurized air tank. In order to properly evaluate the theoretical potential of such a propulsion system, the dynamic programming technique [16] is used to provide an optimal supervisory control algorithm that ensures minimal fuel consumption for every engine configuration considered. Of course, the algorithm must also guarantee that the state of charge, i.e. the tank pressure, must not be lower at the end of a drive cycle than at its beginning. This enables a fair comparison to non-hybrid powertrains.

To use the deterministic dynamic programming technique, the HPV model introduced in section II has to be reduced and reorganized. First, the control options have to be examined. Table V lists all possible and previously described engine modes, taking into account that the first pair of cylinders can act in a different mode than the second pair. Each mode choice is equal to a control input u_j. This control input u_j substitutes low-level control inputs such as throttle or charge valve actuation. To enable a distinction of separate effects of the pneumatic hybridization, various configurations are defined in Table VI. Depending on the engine configuration

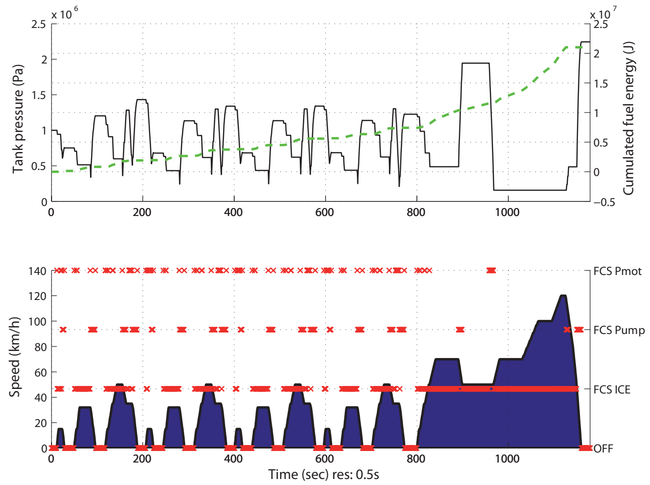

Fig. 4. Simulated state of charge, fuel consumption, vehicle speed and mode usage for the TC14 Engine in the MVEG-95 drive cycle.

chosen, some modes are available and some are not. Consequently, the number of admissible control input values u_j is equal to the number of allowed modes.

Since the drive cycle is assumed to be known in advance for every time step k, the operating point variables ω_e and T_e can be included into each discrete state change function f_k:

$$x_{k+1} = f_k(x_k, u_k), \qquad k = 0, 1, 2, ..., N-1. \tag{21}$$

Furthermore, the same can be done with the cost function $g_k(x_k, u_k)$ for every time step k. The cost is the fuel energy consumed:

$$\sum_{k=0}^{N-1} g_k(x_k, u_k) = H_l \sum_{k=0}^{N-1} \Delta m_f(x_k, u_k) \tag{22}$$

This formulation enables the use of the well-known dynamic programming technique, for details see e.g. [17].

The initial state is chosen to be $x_0 = p_{tank,0} = 10$ bar. The final cost is set to ∞ for $x_N < x_0$, and it is set to zero otherwise. This way, charge sustenance is always enforced.

The state space was limited here from 1.3 bar to 24.2 bar, the state space discretization was chosen to be 0.1 bar. The discretization in time was chosen to be 0.5 s.

IV. RESULTS

A. Dynamic Programming Results for Fuel Consumption

For every engine size and every engine configuration, the dynamic programming algorithm produces the optimal mode choice sequence that minimizes fuel consumption. Such an optimal mode sequence and the resulting state of charge as a function of time is shown in Fig. 4 for the TC14 hybrid engine configuration FCS C.

Here, the algorithm mainly decides when to use the recuperated energy for pneumatic propulsion and when to use the conventional combustion mode instead. The pneumatic motor mode is always used for starting the engine. The pneumatic pump mode is always used for recuperation, with

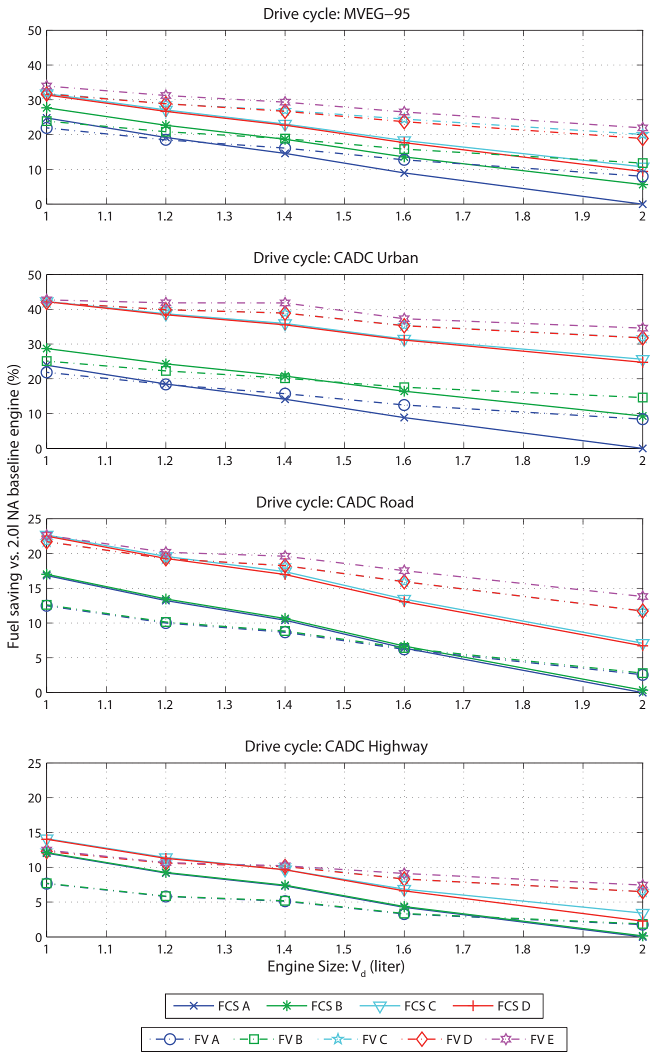

Fig. 5. Simulated fuel saving relative to baseline engine NA20 for various engine configurations and drive cycles. Baseline fuel consumption values are $\{8.00, 10.18, 6.68, 9.38\}$ l/100km

the exception of a part of the last braking phase. There the algorithm chooses the conventional mode since it does not yield any cost because of the fuel cut-off. And since all states $x_N \geq x_0$ have zero cost, pumping does not yield an advantage over the conventional combustion mode.

Fig. 5 shows the fuel saving results for various drive cycles.

Several points are worthwhile mentioning. Using the MVEG-95 cycle as an example, it is found that:

- Downsizing an engine can save as much fuel as 25%. It is the most important contributing factor to the overall fuel gain. Note that without the HPE concept, the "turbo-lag" would be an issue.
- The relative downsizing effect for camless engines is smaller than for FCS engines.
- Adding a start/stop capability to the engine results in 6% fuel saving for the baseline FCS engine. This effect is lower for downsized engines and camless engines, since the engine idling losses are the highest for large FCS engines.
- Adding an EHVS system for all engine valves leads to a fuel saving of 12.5% (8% without cylinder deactivation effects) for non-downsized engines. For heavily downsized engines, the fuel burned due to the higher torque resulting from the hydraulic system energy need exceeds the advantage of having a de-throttled engine.
- The pneumatic hybridization for a fixed camshaft system in combination with downsizing the engine yields a fuel saving of 32%.
- An overall fuel saving of 34% is achieved when considering a downsized camless engine with pneumatic hybridization and cylinder deactivation.
- The fuel saving induced by pneumatic hybridization is higher for a camless non-downsized engine than for a non-downsized engine with fixed camshafts. This is a result of the two-stroke capability of the FV engine configuration.

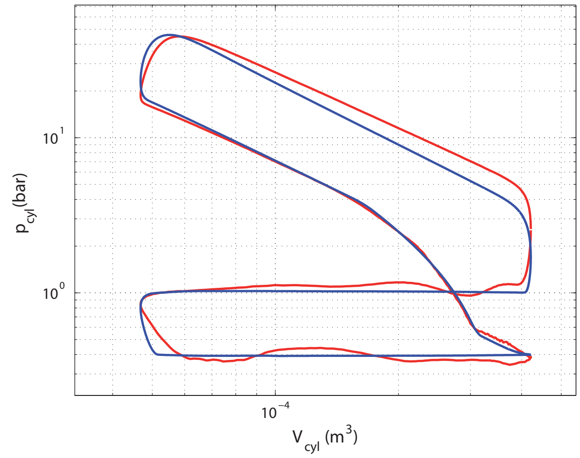

Fig. 6. p-V diagram of the supercharged mode of the modified engine. Blue: Measurement. Red: Process Simulation. $N = 2000$ rpm, $\lambda_{in} = 0.4, p_{mi} = 8.4bar, p_{tank} = 9bar$

B. First Experimental Results

For this project, a two-cylinder multi-purpose engine (MPE750, $V_d = 0.75$ l) manufactured by Weber Automotive GmbH was adapted. It is a port-fuel injection gasoline engine that is available in various configurations. The chosen engine has a compression ratio of $\varepsilon = 9.0$ and can be equipped with a turbocharger. The engine was modified by replacing one exhaust valve per cylinder by an EHVS actuated charge valve. The remaining exhaust valve and the two intake valves per cylinder remain camshaft-driven. The detailed engine setup is described in [18]. An initial test of the supercharged mode is shown in Fig. 6. The air-fuel mixture inducted has to be extremely rich ($\lambda_{in} = 0.4$) to achieve a large instantaneous torque step at an intake pressure of 0.4 bar. Once the intake valves are closed, the charge valve is opened to inject the amount of air that is necessary to achieve a stoichiometric mixture. The behavior of the supercharged mode can be predicted quite accurately using a modified process simulation.

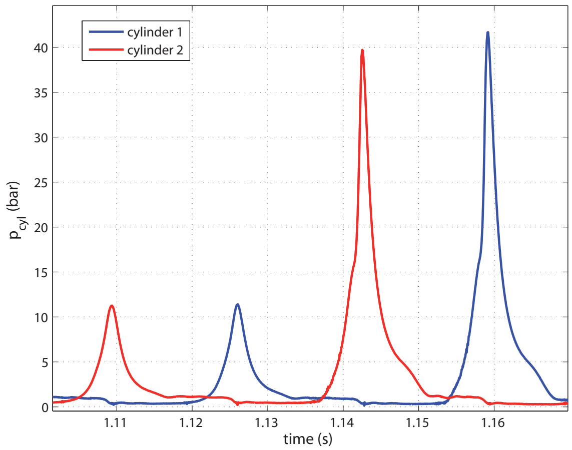

Fig. 7. Measurement: Torque step with supercharged mode with constant intake pressure; $p_{intake} = 0.4bar$, $N = 2000$ rpm, $p_{tank} = 9bar$

In Fig. 7, the two cylinder pressures are shown for a torque step using the supercharged mode at a constant intake pressure of 0.4 bar. This result clearly shows that for a downsized HPE, the turbocharger can be designed for maximum efficiency, since the supercharged mode can be used for transients with high load demands, providing instantaneous torque during the speed-up of the turbocharger. The torque step response is comparable to what is possible with a large electric motor. Further experimental results obtained with this engine are presented in [18].

V. CONCLUSIONS AND FUTURE WORK

A. Conclusions

This contribution shows the theoretical potential for lowering the fuel consumption of an ICE based vehicle propulsion system using engine downsizing and pneumatic hybridization. The dynamic programming technique provides the optimal mode choice sequence for every configuration and thus the benchmark for the optimal fuel consumption.

The results show that pneumatic hybridization is an especially powerful concept when it is combined with engine downsizing. The problem of the "turbo-lag" usually associated with downsizing and supercharging of engines is overcome by injecting additional air during transients. First measurements show the validity of the concept.

Using variable valve actuation for all engine valves enabling two-stroke pneumatic modes and de-throttled combustion operation does not prove to be more fuel efficient when comparing downsized hybrid engines. This is due to the increased need of energy consumption for the electro-hydraulic valve actuation system.

Finally, the hybrid pneumatic engine concept based on fixed camshafts combined with a downsized engine has the potential to be more cost-efficient than fully variable hybrid pneumatic engines and hybrid electric propulsion systems.

B. Future Work

Future Research will focus on the implementation of the turbocharger and the validation of all engine modes for steady-state and transient conditions. Furthermore, the fuel consumption of the hybrid pneumatic engine will be validated for complete drive cycles. The final goal of this project is the integration of a hybrid pneumatic engine into a passenger car to demonstrate the driving performance and the fuel consumption.

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