

Pneumatic Expansion Engine with Freewheeling Clutch Coupling

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ABSTRACT

A new pneumatic motor is presented where the pneumatic to mechanical energy conversion is based on double effect linear actuators. The parallel running cylinders are pneumatically coupled through intake, transfer and exhaust valves which allow to combine constant pressure displacement work and thermodynamic expansion work in order to reach an augmented energy efficiency. The linear motion of the actuator assembly is transformed into a rotational movement through an original connecting-rod based mechanism pushing and pulling lever-arms which are coupled to freewheeling clutch bearings. The paper shows the different variable as forces, torque, power and produced mechanical work. The energetic performance of the new engine is analysed, and finally, a small dimension demonstrator is presented.

Keywords: Compressed air motor; linear actuators; energy efficiency; clutch bearings; coupling mechanism.

INTRODUCTION

Compressed air engines have been proposed in many contexts of use and in many different technologies. Their main advantages are in the simplicity and reliability of the technology, and in the acquisition costs of the equipments. Near standard industry applications in production of goods, handling or moving of objects, another application field in the energy sector concerns compressed air energy storage where conversion engines are used. Compressors are filling the storage reservoirs while in the recovery path expansion turbines or motors are placed between the storage vessel itself and the electrical generation machine (1-4). The classical compressed air machines go from rotary vane type to piston/crankshaft types, through gear type or turbine type motors.

In general, these motors have a low energetic efficiency due to their principle of conversion where only displacement work is produced in the machine (5-7). Recent developments in the field of pneumatic to mechanical conversion allow to strongly increase the energetic performances of such machines. These benefits are realised through the addition to the conversion principle of a second torque component resulting from a real thermodynamic expansion inside of the active volume or in an added supplementary chamber of larger dimensions (8-11).

Pneumatic drives for propulsion of vehicles have also been proposed with very different energetic performances. Transformed machines initially running with internal combustion of fuel have been proposed as zero-emissions vehicles, also dedicated architectures with increased efficiency (12-15). Previous work has described an original compressed air engine where the conversion of the compressed air energy is realized on the base of so-called PAMs, Pneumatic Artificial Muscles (16-18).

Instead of using a classical crankshaft for the coupling of the linear motion of the muscles to the rotation of the shaft, an original architecture has been proposed where the artificial muscles are driving the shaft of the engine with lever-arms which impose their sectorial rotation through freewheel clutch bearings (19-20). In a

further study, the coupling mechanism has been transformed into a specific mechanism allowing the coupling of both the pulling and the pushing effort of the actuator into a rotational movement (21).

This mechanism is re-used in the present studied engine. Instead of using a PAM as a primary actuator, the present engine uses classical linear piston-cylinders in an original assembly allowing to reach an increased energy efficiency (22, 11). Figure 1 illustrates the concept of the new engine.

The original assembly of the pneumatic actuator is based on the principle of combining displacement work and thermodynamic expansion. This principle uses a central bidirectional cylinder where the filling of the chambers is done with air at constant pressure. Then the air is expanded by a transfer from the central cylinder into the chambers of two lateral cylinders running in parallel.

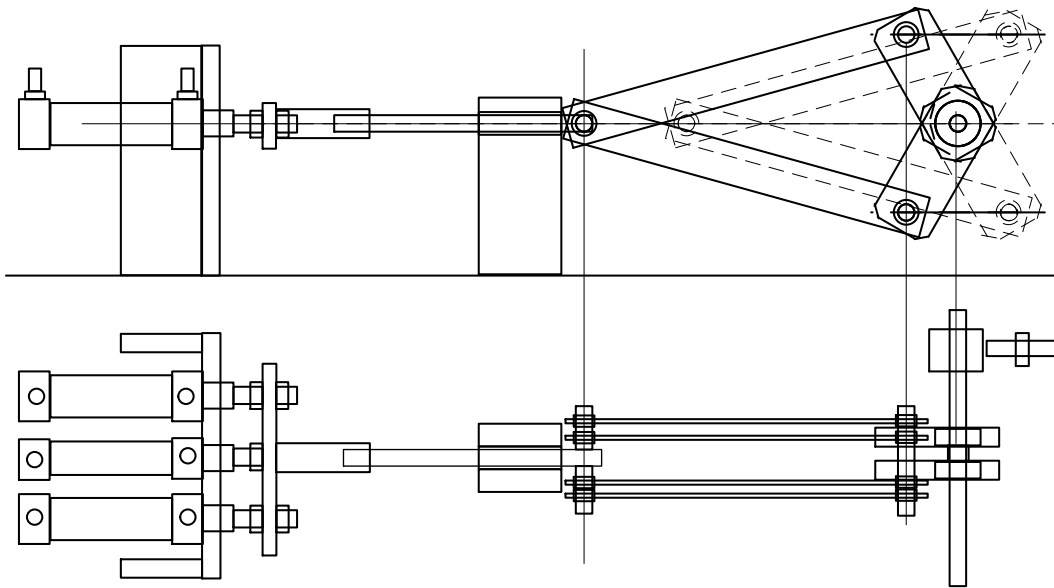


Figure 1 Structural diagram of the engine

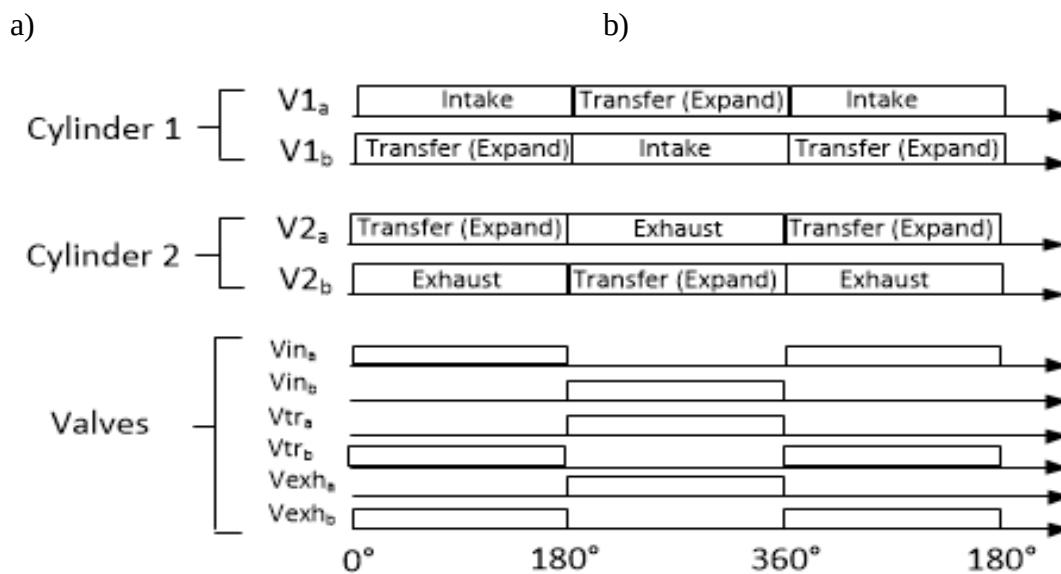
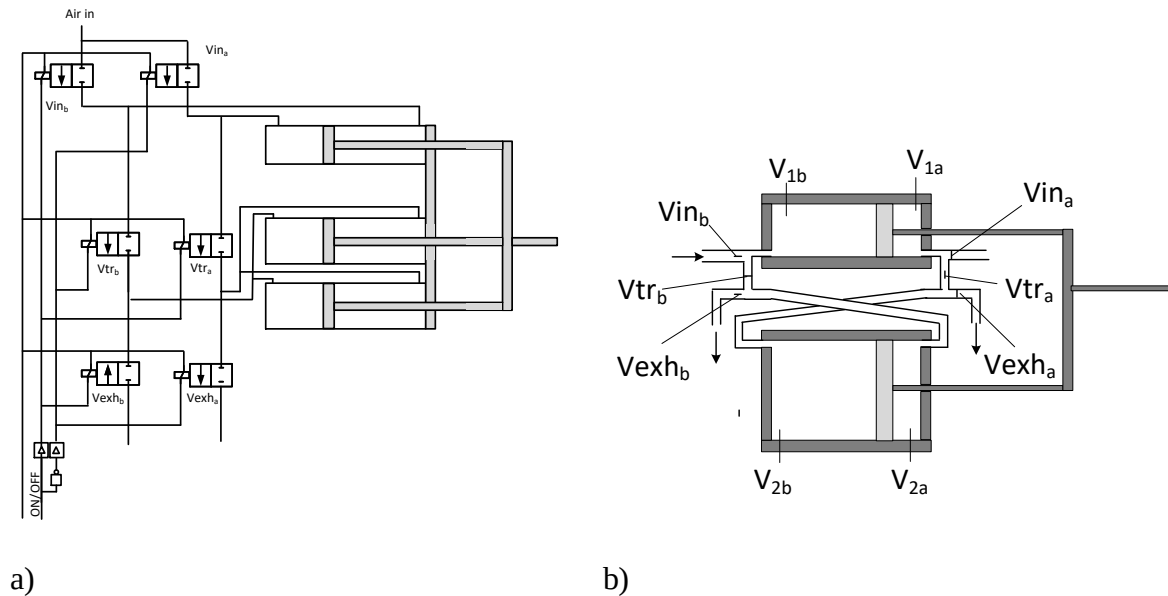
A Cylinder Assembly for an Actuator With Higher Efficiency

The principle of recovering the thermodynamic content of the pressurized air is applied to linear cylinders, where in addition to the simple displacement work usually produced in a conventional cylinder, the air is additionally expanded in a supplementary pneumatic chamber system, allowing to recover a significant part of the injected enthalpy [11].

Operating Principle and Control

The pneumatic actuator of the present engine is an assembly of individual cylinders as represented in Fig. 1 and 2a. The chambers of the central cylinder are fed from the air reservoir under constant pressure. Then the air is transferred into the chambers of the two other cylinders running in parallel. As represented in the diagram of Fig. 2a, two intake valves V_{in_a} and V_{in_b} are feeding alternately the pulling and pushing chambers of the first cylinder. The transfer of the air from the first cylinder to the two others is controlled through the two transfer valves V_{tr_a} and V_{tr_b} .

Finally, after the expansion, the air at the final expansion pressure is released to the external atmosphere through two exhaust valves, V_{exh_a} and V_{exh_b} . The sequences of operation are defined according to the diagram of Fig. 2c. The simulation of the system is considering a simplified model where the two parallel running cylinders are represented by one cylinder of the corresponding larger volume (Fig. 2b). In the studied example, a central cylinder of 12mm diameter is chosen with two lateral ones of each 16mm diameter. As a result, the volumetric expansion ratio is 3.55.



c)
Figure 2 Principle of operation a) cylinder arrangement with control valves, b) equivalent of volumes, c) Control diagram

The Pressure Variation during Expansion

The expansion of the air is supposed to be of the adiabatic type. This assumption is based on the fact that the air flows rapidly and does not rest in the cylinders, with the consequence that the thermal transfer time is too short. The resulting pressure P_2 in the chambers V_{2a} (first half-cycle) and V_{2b} (second half-cycle) and of the feeding chambers V_{1b} and respectively V_{1a} takes the value of

$$P_2 = P_{in} \left(\frac{V_{1max}}{V_{1a,b} + V_{2b,a}} \right)^\gamma \quad \text{with } \gamma = 1.4 \quad (1)$$

P_2 , V_{1a} , V_{2b} respectively P_2' , V_{1b} , V_{2a} are system variables which change during the expansion phases in dependency of the piston's displacement during the first, respectively the second half period. For example, V_{1a} is decreasing during the "push" stroke, while V_{2b} is increasing.

With the dimensions of the chosen cylinders of respectively 12 and 16 mm, the ideal volume ratio which appears at the end of the stroke becomes

$$\frac{V_i}{V_f} = \frac{l \cdot \pi D_1^2 / 4}{2 \cdot l \cdot \pi D_2^2 / 4} = \frac{0.012^2}{2 \cdot 0.016^2} = 0.281 \quad (2)$$

V_i corresponds to the initial (maximum) value of the chambers of the central cylinder, and V_f corresponds to the final value of the sum of the feeding and of the expanding chambers of the two parallel running lateral ones.

and the corresponding final pressure ratio becomes

$P_2/P_{in} = 0.169$ according to rel. (1).

In the real case, there is an asymmetry of expansions and forces between the right- and the left side of the pistons in the cylinders. The asymmetry is due to the presence of the piston rod inside of one of the cylinder chambers (at the pushing side) which reduces the pressure area. Table 2 shows the values of the parameters of the chosen cylinders, their active surfaces in the of the push and pull motions, together with the length of the strokes.

Cylinder type	Bore size	Pressure area (push)	Pressure area (pull)	Stroke
MI12-50	12 mm	113 mm ²	93.4 mm ²	50 mm
MI16-50	16 mm	201 mm ²	181 mm ²	50 mm

Table 2 Parameters of the cylinders

The asymmetry of the pressure areas affects first the produced force caused by a given pressure, and second it causes an asymmetry of the volumes, which further have their effect on the expansion ratio in the push and pull modes. The volume $V_{1a\max}$ at the pull side (right side) of the 12 mm cylinder is

$$V_{1a\max} = l \cdot \pi (D_1^2 - d_{rod}^2) / 4 = 0.050 \cdot \pi \cdot (0.012^2 - 0.006^2) / 4 = 4.24 \cdot 10^{-6} m^3 \quad (3)$$

d_{rod} being the diameter of the piston rod (Figure 3)

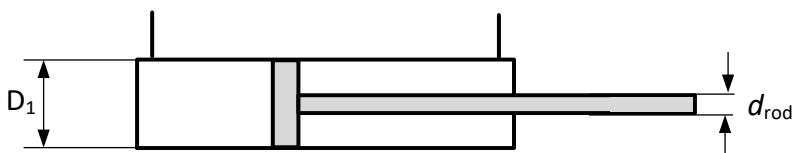


Figure 3 Diameters of cylinder and piston rod

At the push side (left side) $V_{1b\max}$ is

$$V_{1b\max} = l \cdot \pi D_1^2 / 4 = 0.050 \cdot \pi \cdot (0.012^2) / 4 = 5.65 \cdot 10^{-6} m^3 \quad (4)$$

For the 16 mm cylinder, the volumes are respectively $V_{2a\max}$ and $V_{2b\max}$

$$V_{2a\max} = l \cdot \pi (D_2^2 - d_{rod}^2) / 4 = 0.050 \cdot \pi \cdot (0.016^2 - 0.006^2) / 4 = 8.63 \cdot 10^{-6} m^3 \quad (5)$$

$$V_{2b\max} = l \cdot \pi D_2^2 / 4 = 0.050 \cdot \pi \cdot (0.016^2) / 4 = 10.05 \cdot 10^{-6} m^3 \quad (6)$$

As a consequence, the volume and pressure ratios become

$$\frac{V_{1a\max}}{2 \cdot V_{2b\max}} = \frac{4.24 \cdot 10^{-6}}{2 \cdot 10.05 \cdot 10^{-6}} = 0.210 \quad (7)$$

$$\frac{P_{21}}{P_{in}} = \left(\frac{V_{1a\max}}{2 \cdot V_{2b\max}} \right)^{1.4} = 0.114 \quad (8)$$

$$\frac{V_{1b\max}}{2 \cdot V_{2a\max}} = \frac{5.65 \cdot 10^{-6}}{2 \cdot 8.63 \cdot 10^{-6}} = 0.327 \quad (9)$$

$$\frac{P_{22}}{P_{in}} = \left(\frac{V_{1b\max}}{2 \cdot V_{2a\max}} \right)^{1.4} = 0.2094 \quad (10)$$

Modeling the Complete Engine.

The produced mechanical work by the proposed system must be evaluated first as the effort produced by the cylinder assembly and then as the torque transmitted to the engine's shaft. The actuator assembly produces pulling and pushing forces, as contributions of four main components namely the forces generated by the intake pressure on the surfaces of the first piston and the forces generated by the expansion and exhaust pressures on the surfaces of the first and second one. Additionally, the four force components must be evaluated for the pull and push motion of the actuator. An analytic calculation of the sum of these eight components is not realistic, due to the fact that the expansion pressure depends on the position of the pistons which is imposed by the coupling mechanism and the angular position of the engine's shaft. Thus, the different variables are calculated by simulation of the complete system.

Structure of the Coupling

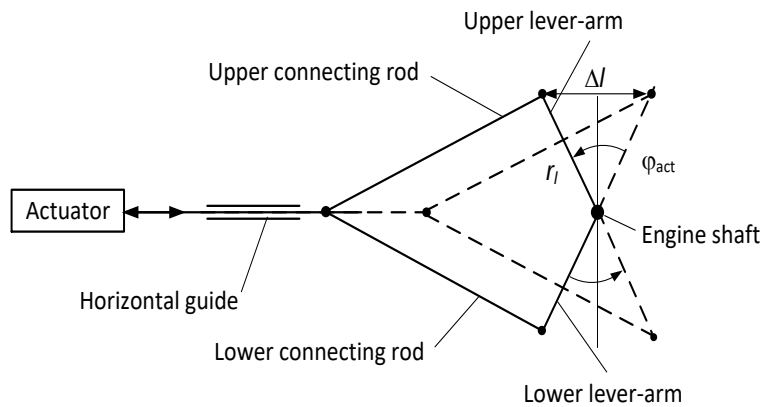
For the representation of the variables of the complete system, a mathematic description of the kinematics of the coupling of the cylinder's piston rods to the engine's shaft is needed. Figure 1 has given a first view of the coupling mechanism which uses short lever-arms driving freewheeling clutch bearings. The end connection of the three pistons is maintained in line by a horizontal linear guide and is coupled to two connecting rods (Figures 1 and 4). The positive force of the cylinder assembly (acting from the right to the left in Figure 4) is pulling the upper lever-arm and produces a torque which acts anticlockwise on the engine shaft. The clutch bearing of the lower lever-arm is freewheeling during this stroke. The negative force of the cylinder assembly (acting from the left to the right in Figure 4) is pushing the lower lever-arm and acts on the shaft with a torque in the same direction as the upper one during the previous stroke. The clutch bearing of the upper lever-arm is freewheeling during this stroke. The positive and negative actions of the cylinder assembly are defined over a limited angle during each half-period (limited oscillating angle of the lever-arms), and are positioned with a 180° phase-shift, assuming no overlap of their active modes.

In the present application, the active rotation angle φ_{act} is a result of the defined cylinder stroke Δl and the radius of gyration of the lever-arms r : $\varphi_{act} = 2 \cdot \arcsin(\Delta l / 2r)$ as represented in Figure 4.

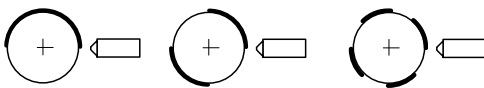
The radius of gyration of the lever-arms is chosen equal to the cylinder stroke, resulting to an oscillation angle φ_{act} of 60°.

From the simulation curves of the torque (see next sections) one can see the dead zone existing in the torque transmission. This intermittency can be corrected with multiple interventions triggered from an adapted angular position sensor (see Figure 4b).

A view of the coupling from the realized experimental set-up is given in Figure 5.



a)



b)

Figure 4 a) Scheme of the coupling mechanism, b) Different position sensors for the multiple torque interventions.

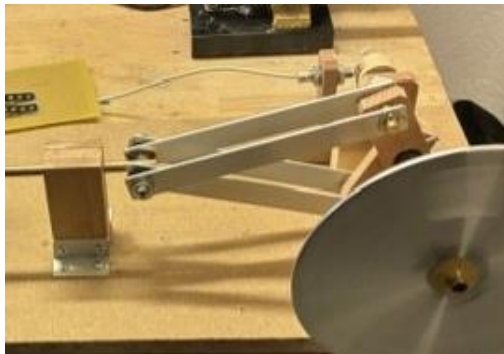


Figure 5 Detail view of the coupling (experimental set-up)

Simulation of the System

The simulation of the system is realized with a tool where the different elements are represented as functional blocks (Simulink). One period of rotation of the engine is simulated as a quasi-stationary phenomenon, and with an imposed value of the angular velocity ($\omega=6.28$ rad/s). The coupling angles of the clutch bearings are calculated from the running angle ωt .

In the simulation program all elements are represented with functional blocs. Trigonometric, algebraic and other classical analogic and logic functions are implemented directly.

Mathematical Description of the Lever-Arm Assembly

In Figure 6 the coupled cylinder assembly is represented with a connecting rod and a crankshaft used for the illustrating of the lever-arm principle. The parameters are indicated as r_l , the radius of gyration of the lever-arm, l_{cr} the length of the connecting rod, and j the angle of rotation of the crankshaft. The position of the end of the piston rod x is also indicated.

The relation between the position x of the pistons end and the rotation angle of the shaft is given through Rel. (17). This relation is valid for the status of the engaged clutch bearings.

$$x = r_l(1 - \cos \varphi) + \frac{\lambda}{2} r_l \sin^2 \varphi \quad (11)$$

where the connecting rod ratio λ is used and is defined as

$$\lambda = \frac{r_l}{l_{cr}} \quad (12)$$

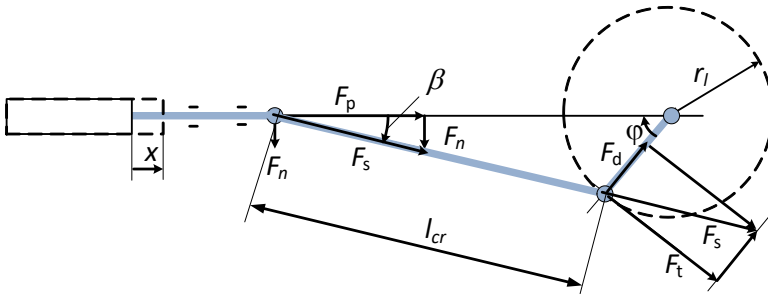


Figure 6 Diagram of the forces for the calculation of the torque and reactions

The torque on the crankshaft is given by Rel.13.

$$M_{mot} = F_t \cdot r_l \quad (13)$$

With the tangent force F_t which is given by (14)

$$F_t = F_s \cos(\pi / 2 - (\beta + \varphi)) \quad (14)$$

The force transmitted through the connecting rod F_s is calculated from the actuator's force F_p and the angle beta (15)

$$F_s = \frac{F_p}{\cos \beta} \quad (15)$$

This angle is given by Rel. (16)

$$\beta = \arcsin \left(\frac{r_l}{l_{cr}} \cdot \sin \varphi \right) \quad (16)$$

The power produced at the shaft's level is also defined

$$P_{mech} = M \cdot \omega \quad (17)$$

Simulation Results

The variables of the system as positions, forces, torques, power and transmitted energy are calculated for a complete period of a system. First, the position of the linear actuator assembly's end is calculated which is defined by the angular position of the lever-arm when the clutch bearings are locked. The curves of Figure 7 are representing the variations of the different volumes of the active chambers of the cylinders. The upper two curves represent the decreasing volume of the left chamber of the central cylinder, and the sum of the two increasing volumes at the right sides of the lateral cylinders. This corresponds to a pulling action of the assembly (right-to-the left action). The lower two curves represent the increasing and decreasing volumes of

the opposite chambers which act in the pushing mode (left-to-the-right motion of the actuator). In these representations the asymmetries of the volume variations can be well recognized.

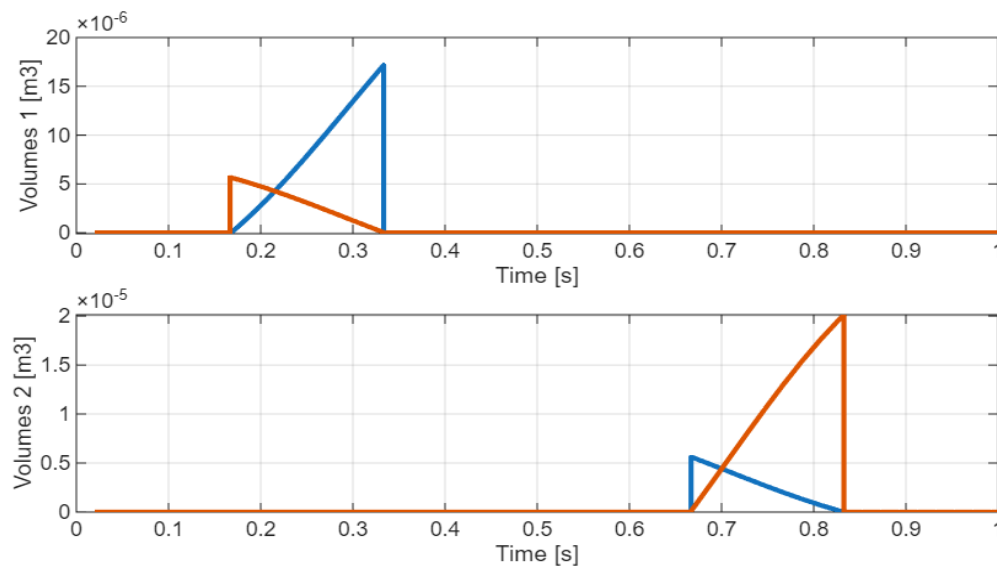


Figure 7 Volumes of the cylinder chambers.

The resulting expansion pressures in the corresponding chambers are represented in Figure 8. For both expansions, meaning in the first and second half-wave, the pressure at the beginning of the expansion corresponds to the intake pressure, the pressure of the air drawn in the chambers of the central cylinder.

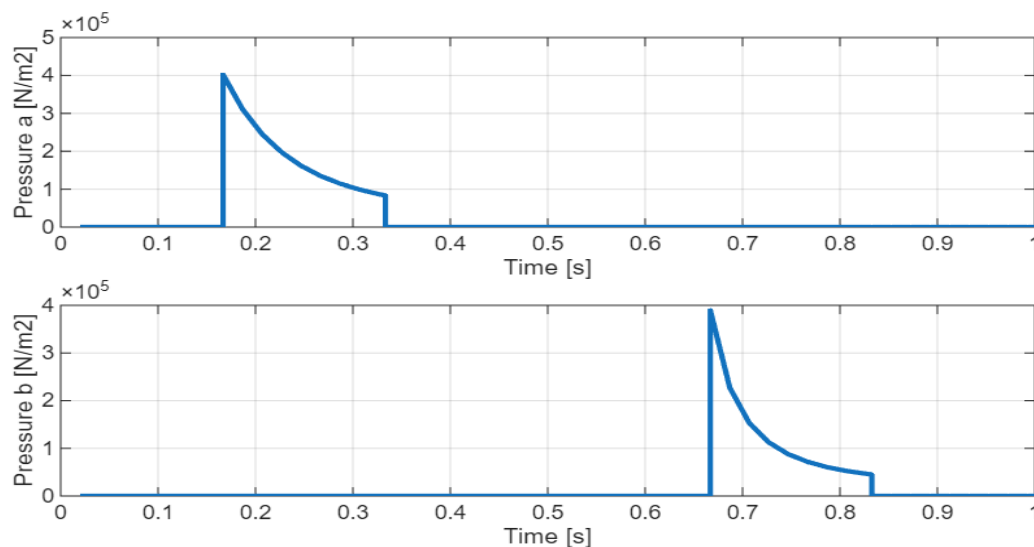


Figure 8 Pressures in the expansion volumes

Figure 9 shows the effects of the pressures on the two sides of each cylinder, together with the resulting force of the actuator in the first and second half-wave. The rising curve (blue, left) corresponds to the total force of the central cylinder. It is the result of the sum of the constant positive force on the right side of the central piston during intake and of the decreasing force on the left side of this piston due to the transfer of air into the other two cylinders (expansion). The smaller decreasing force (left, yellow) is the total force exerted by the two lateral cylinders. It is the result of the sum of the expansion force of the air on the right sides of the two lateral pistons, minus the effect of the atmospheric pressure on the left sides of these pistons during exhaust of the expanded air. The group of curves during the second half-wave shows the forces during the return stroke of the actuator, and again show the asymmetries of the effects due to the presence of the piston rods inside of the left chambers of the cylinders.

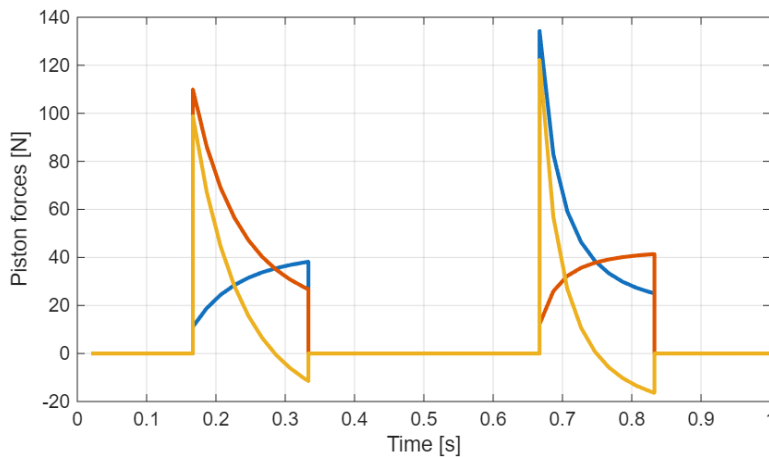


Figure 9 Forces produced by the pistons

The upper curves of figure 10 show the effect of the different forces transmitted by the coupling mechanism to the engine shaft. The blue curve is the resultant force of the actuator (F_P , Fig. 6), and the red curve is the force transmitted by the connecting rod (F_s , Fig. 6). The yellow curve corresponds to the tangential force (F_t , Fig. 6). The lower curve of Figure 10 represents the torque transmitted to the engine's shaft.

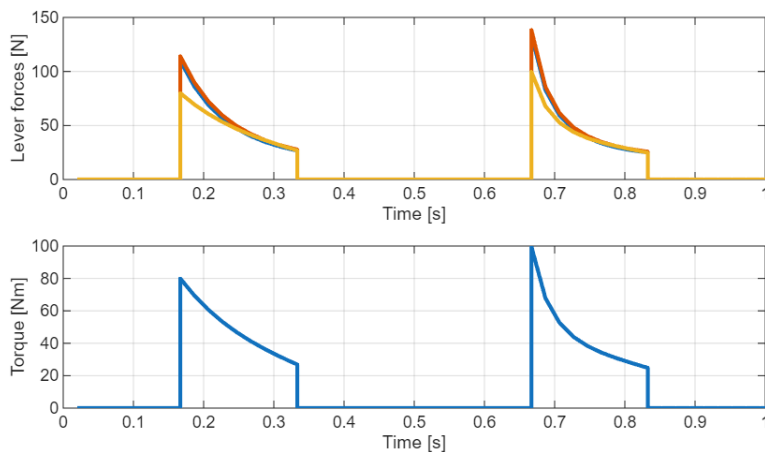


Figure 10 Forces of the coupling mechanism and resultant torque.

Finally, Figure 11 shows the power produced by the engine and the energy converted or the work produced by the engine.

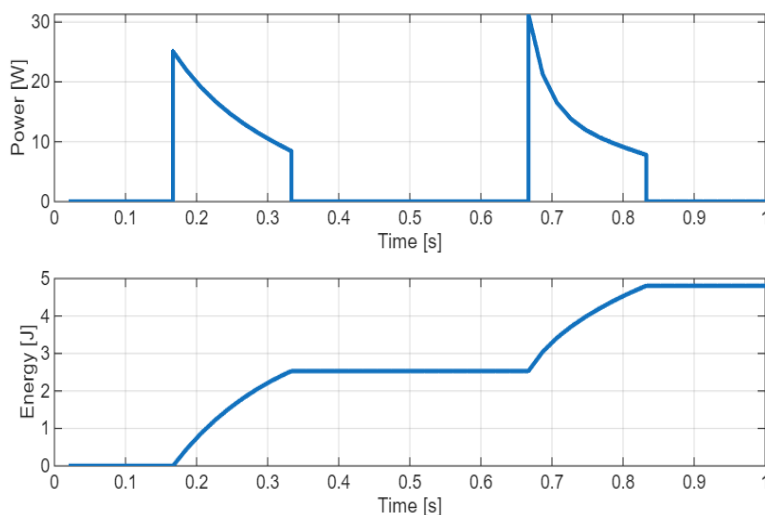


Figure 11 Power produced by the engine and energy converted

Energy Efficiency

To evaluate the energy efficiency of an air engine, the ratio of the mechanical work produced, W_{out} , to the enthalpy, H_{in} , of the injected air is calculated (Rel. 18).

$$\eta_{conv} = \frac{W_{out}}{H_{in}} \quad (18)$$

The value of the produced mechanical work W_{out} is obtained from the integration of the instantaneous power P_{mech} over one period of the engine's cycle.

$$W_{out} = \int_0^t P_{mech}(t) dt \quad (19)$$

From the simulation results given in Figure 11 the produced mechanical work over one period is equal to $W_{out} = 4.807 \text{ J}$.

Due to the asymmetry of the volumes of the cylinders, the enthalpy injected for one cycle must be calculated in two steps, namely for the filling of the right (a) and left (b) chambers of the first cylinder.

For the enthalpy injected into the two chambers two terms remain to be evaluated, namely the internal energy U and the transfer term PDV.

$$H_{in} = U_a + P_{in} \cdot \Delta V_{1a} + U_b + P_{in} \cdot \Delta V_{1b} \quad (20)$$

Air transfer

$$P_1 \cdot V_{1a} = 4 \cdot 10^5 \text{ N/m}^2 \cdot 4.24 \cdot 10^{-6} \text{ m}^3 = 1.696 \text{ J} \quad (21)$$

$$P_1 \cdot V_{1b} = 4 \cdot 10^5 \text{ N/m}^2 \cdot 5.65 \cdot 10^{-6} \text{ m}^3 = 2.26 \text{ J} \quad (22)$$

The volumes of the right and left chambers being

$$V_{1a} = l \cdot \pi (D_1^2 - d_{rod}^2) / 4 = 0.050 \cdot \pi \cdot (0.012^2 - 0.006^2) / 4 = 4.24 \cdot 10^{-6} \text{ m}^3 \quad (23)$$

$$V_{1b} = l \cdot \pi D_1^2 / 4 = 0.050 \cdot \pi \cdot (0.012^2) / 4 = 5.65 \cdot 10^{-6} \text{ m}^3 \quad (24)$$

Internal energies

The internal energies U_a and U_b of the injected compressed air are calculated as

$$U_a = P_1 \cdot V_{1a} \left(\ln \frac{P_1}{P_a} - 1 + \frac{P_a}{P_1} \right) = 1.696 \text{ J} (0.636) = 1.079 \text{ J} \quad (25)$$

$$U_b = P_1 \cdot V_{1b} \left(\ln \frac{P_1}{P_a} - 1 + \frac{P_a}{P_1} \right) = 2.26 \text{ J} (0.636) = 1.43 \text{ J} \quad (26)$$

They correspond to an isothermal expansion from a pressure P_1 to an atmospheric environment (Ref. [1]).

The efficiency calculated over a period of 360° comprising a right-to-the-left and left-to-the right motion of the piston assembly becomes

$$\eta = \frac{W}{P_1 \cdot V_{1b} + U_b + P_1 \cdot V_{1a} + U_a} = \frac{4.807 J}{1.69 J + 1.079 J + 2.26 J + 1.43 J} = \frac{4.807 J}{6.46 J} = 0.744$$

Experimental Set-up

Figure 12 shows the realized experimental set-up with the original cylinder assembly and the control valves (left). At the right side of the picture the coupling mechanism can be seen.

On the front side of the engine shaft the flywheel disc is mounted, and at the rear side the angular position sensor is visible which is realised with a rotating piece with sectors made of electromagnetic material and a proximity sensor (see also Figure 4b).

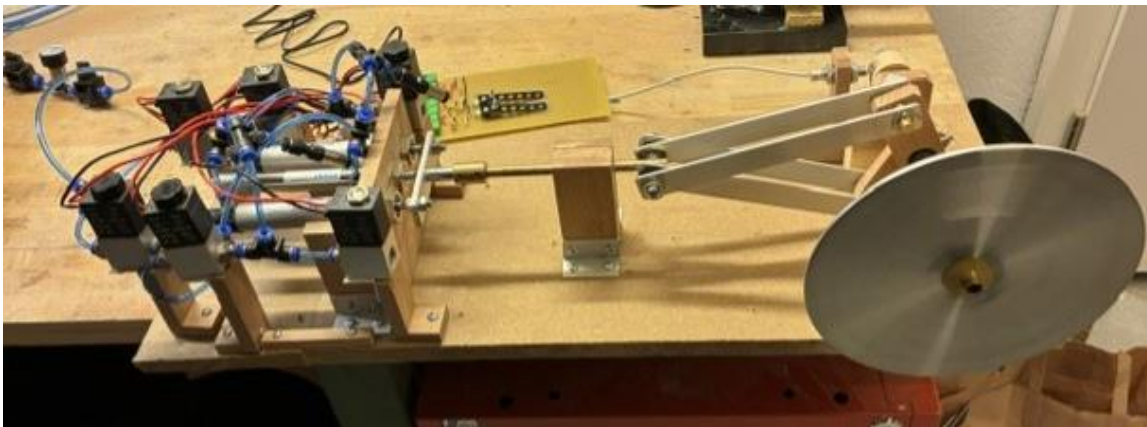


Figure 12 Experimental set-up

Measurements on the experimental set-up

Figure 13 shows the values of the measured pressures during a filling stroke (constant pressure) in the central cylinder followed by an expansion into the two lateral cylinders. This measurement shows a specific effect of the existing dead volumes in the path of air transfer between cylinders, due to tubing and valves. When the central cylinder is filled at a pressure P_1 , the expansion phenomenon begins not at that value, but at a pressure P_1' . The change from P_1 to P_1' is due to the transfer of air into the dead volume, causing a pre-expansion [23]. This effect is particularly marked for a system with small cylinders, what is the case of the realized demonstrator.

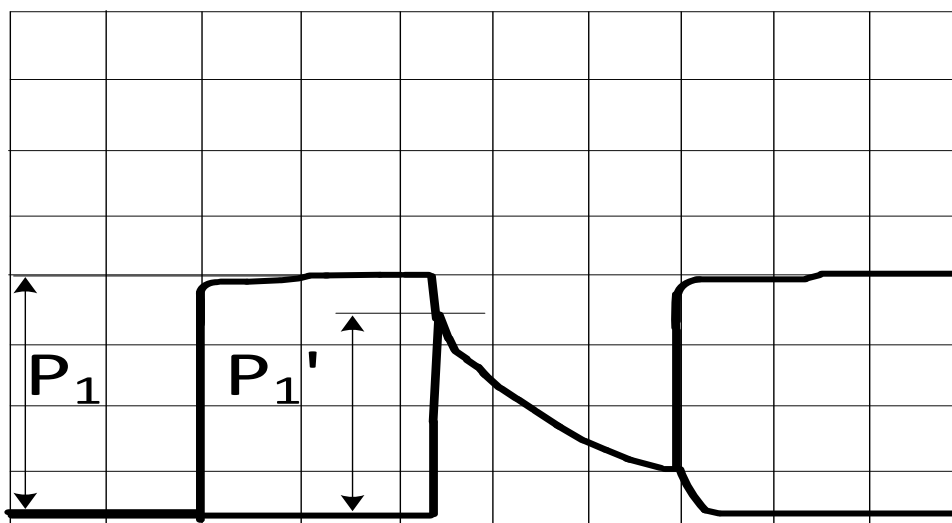


Figure 13 Pressure in the cylinders during filling and expansion ($1 \cdot 10^5 \text{ N/m}^2$ per division).

Additionally, the friction of the pistons in the cylinders would be a second source of deviation from the theoretical values in terms of efficiency. Especially for small cylinders, when the friction force of the piston is proportional to the diameter and the pressure force proportional to the square of the diameter.

Under these conditions, any measurement of torque, power and mechanical work cannot be used for a practical verification of the simulated performances. Such measurements should be made for a demonstrator of larger dimensions.

The realized small demonstrator could however be considered as a good proof of concept.

CONCLUSIONS

A new compressed air engine has been presented and analysed. The forces developed by a dedicated cylinder assembly have been established by simulation, as also the torque transmitted to the engine's shaft. Produced power and mechanical work, together with the injected enthalpy have allowed to calculate the energetic performance of the motor. The well augmented energy efficiency is the result of the concept of combining constant pressure displacement work with real thermodynamic expansion energy inside of the cylinder assembly.

A comparison of the efficiency obtained with previously investigated systems (ref. 8-11) show that similar values are obtained with the new concept. The relatively high value of more than 0.7 is mainly due to the addition of an expansion process of the injected air. The coupling mechanism plays only a minor role, because of all elements of the coupling are not affected by significant losses. These losses can be evaluated as a small part only of the added expansion energy.

The use of double effect single rod pneumatic cylinders has the disadvantage of presenting different pressure areas at the push and pull sides of the pistons in the cylinders, leading to asymmetries as well in the linear forces as in the produced torque of the engine.

The operating principle of the cylinder assembly and of the original coupling mechanism have been verified with a small experimental set-up.

REFERENCES

1. Rufer, A., Energy storage-Systems and components, CRC Press, Taylor & Francis group, Boca Raton FL, November, 2017.
2. F. Crotogino, K.-U. Mohmeyer, R. Scharf, Huntorf CAES: More than 20 Years of Successful Operation, Spring 2001 Meeting Orlando Florida USA 15-18 April 2001
3. A.J. Giramonti, R.D. Lessard, W.A. Blecher, E.B. Smith, Conceptual design of compressed air energy storage electric power systems, Appl Energy, vol. 4 (4) pp. 231-249 1978.
4. R. Saidur, N. A. Rahim, M. Hasanuzzaman, A review on compressed-air energy use and energy savings, Renewable and Sustainable Energy Reviews, vol. 14, 4, pp. 1135–1153, 2010.
5. Devang Marvania, Sudhakar Subudhi, A comprehensive review on compressed air powered engine, Renewable and Sustainable Energy Reviews, Volume 70, 2017, Pages 1119-1130, ISSN 1364-0321, <https://doi.org/10.1016/j.rser.2016.12.016>.
(<https://www.sciencedirect.com/science/article/pii/S1364032116310681>).
6. Atlas Copco, Pocket guide to air motors, 9833 9067 01 2015:1 Jetlag/Boardwalk. Printed in Sweden. © Atlas Copco Industrial Technique AB.
7. Dhumal, A., Ambhore, N., Naik, A. et al. Design and Optimization of Air Motor with Gear Rotor System for Enhanced Performance. J. Inst. Eng. India Ser. C 106, 367–376 (2025). <https://doi.org/10.1007/s40032-024-01157-6>.
8. Rufer, A., A High Efficiency Pneumatic Drive System Using Vane-Type Semi-Rotary Actuators, Facta Universitatis, Series: Electronics and Energetics Vol. 34, No 3, September 2021, pp. 415-433, <https://doi.org/10.2298/FUEE2103415R>.

9. Rufer, A., A High Efficiency Pneumatic Motor Based on Double-Acting Linear Cylinders, *World Wide Journal of Multidisciplinary Research and Development*, WWJMRD, 2021, 7(1), 25-36.
10. Rufer, A., A four-cylinder pneumatic expansion motor with high efficiency, *Academia Green Energy*, 2(2), April 2025, DOI: 10.20935/AcadEnergy7625.
11. A. Rufer, *Semi-Rotary and Linear Actuators for Compressed Air Energy Storage and Energy Efficient Pneumatic Applications*, Bentham Science Publishers Pte, Ltd, Singapore, 2023, ISBN 978-981-5179-11-8.
12. H. Liu, Y. Chen, G.L. Tao, G.Z. Jia, W.H. Ding, Research on the displacement and stroke-bore ratio of the air-powered engine, *Proceedings of the Sixth International Conference on Fluid Power Transmission and Control* (2005), pp. 381-384
13. Yidong Fang, Yiji Lu, Anthony Paul Roskilly, Xiaoli Yu, A review of compressed air energy systems in vehicle transport, *Energy Strategy Reviews*, Volume 33, 2021, 100583, ISSN 2211-467X,
14. Truglia, Vito Gianfranco, High efficiency engine driven by pressurized air or other compressible gases, US Patent, US 9,677,400 B2 Jun. 13, 2017.
15. Nègre, Guy, Engine with an active mono-energy and/or bi-energy chamber with compressed air and/or additional energy and thermodynamic cycle thereof, US Patent, US 7,469,527 B2, Dec. 30, 2008
16. Koruku, S., Samtas, G., Soy, G., Design and Experimental Investigation of Pneumatic Movement Mechanism Supported by Mechanic Cam and Crank Shaft, *TEM Journal*, Volume 4 / Number 1 / 2015, www.temjournal.com.
17. FESTO, Fluidic muscle,
https://ftp.festo.com/Public/PNEUMATIC/SOFTWARE_SERVICE/Documentation/2020/DE/DMSP-MAS_DE.PDF.
18. Rufer, A., A New Compressed Air Engine Based on Pneumatic Artificial Muscles (PAM's) , *Worldwide Journal of Multidisciplinary Research and Development*, WWJMRD, 2026, 12(05), pp 14-20.
19. A. Rufer, Clutch Bearing Coupled PAM's (Pneumatic Artificial Muscle) for a Compressed Air Engine, WWJMRD, *Worldwide Journal of Multidisciplinary Research and Development*, February 2026.
20. Freewheels – Their functions and applications,
<https://hard-met.pl/wp-content/uploads/2019/06/Freewheel-Clutch.pdf>, accessed April 18th 2026.
21. Rufer, A. Deformation Work Recovery for a Compressed Air Engine Driven by Pneumatic Artificial Muscles, *Facta Universitatis, Series Electronics and Energetics*, Vol. xx, No y 2026, pp pp
22. Rufer, A. Linear Pneumatic Cylinder Assembly with Reduced Air Consumption, October 2023
DOI: 10.2174/9789815179095123010007 In book: *Semi-rotary and Linear Actuators for Compressed Air Energy Storage and Energy Efficient Pneumatic Applications*.
23. A. Rufer, *Semi-Rotary and Linear Actuators for Compressed Air Energy Storage and Energy Efficient Pneumatic Applications*, Sect. 6.6.1, Bentham Science Publishers Pte, Ltd, Singapore, 2023, ISBN 978-981-5179-11-8.