19th Brazilian Congress of Thermal Sciences and Engineering
November 06th-10th, 2022, Bento Gonçalves - RS - Brazil

ENCIT-2022-0253
THERMODYNAMIC MODEL FOR A COMPRESSED AIR MOTOR

De Aguiar, João B.
De Aguiar, José M.
UFABC, Av. dos Estados, 5001, Bangú, Santo André, SP, Brasil, 09210-580
FATEC_SP, Av. Tiradentes, 615, Bom Retiro, São Paulo, SP, Brasil, 01124-060
joao.aguiar@ufabc.edu.br, josemaguiar@gmail.com

Abstract. *Lately engines driven with compressed air have become an important subject of research, even though these motors are not new. First compressed air vehicles were built in Paris between 1838 and 1940. But the small amount of energy stored in the compressed air of a typical tank was considered insufficient for practical use. Nowadays, however, new motors have been proposed, like the one by MDI that developed the Air Pod, a compressed air vehicle equipped with a small engine of 430 cm3, 7 kw power at 1500 rpm, maximum torque of 45 Nm, capable of running 130 km in urban conditions using two tanks of air, each one with 125 l, as claimed. For each inlet condition, a different amount of theoretical work can be obtained, according to the thermodynamic open cycle model hypotheses. Yet the real work produced in the expansion cycle will differ from the theoretical one, as it depends on the construction elements of the machine. These two questions are considered for single stage processes in this work. An open cycle approximation model is used to determine the energy exchange, thermodynamic state variables and efficiency. These ideal results are used to qualify the performance of an air motor. Discussion of the results for modified models and some improvements are also considered.*
Keywords: *thermodynamic cycle, work output, model, efficiency*

1. INTRODUCTION

In recent times, serious environmental issues brought concerns in human society related to the degradation of environment caused by greenhouse effect derived from the massive use of fossil fuels. In this sense many solutions were proposed in the transportation industry to deal with the problem. Electric, (Salvi and Subramanian, 2016), hydrogen and nitrogen cars (Negre et al, 2015), for example, appeared with appealing promises. Older solutions, like the pneumatic vehicles, with the use of high pressure air tanks were specially considered due to their low cost, simplicity of use, due in large regards to existing technology, Fig. 1.

By far the most promising solution was the one pushed forth by compressed air car builder (MDI, 2016) that developed the Air Pod, a compressed air vehicle equipped with a small engine of 430 cm3, 7 kw power at 1500 rpm, maximum torque of 45 Nm, capable of running 130 km in urban conditions using two tanks of air, each one with 125 l, as claimed, Fig. 2.

Figure 1. An arrangement of parts in a simple compressed air system: (1) Atmospheric, (2)+(3) Pressurized, (4) Expanded Air

Figure 2. An Air Pod in test in Europe

2. THERMODYNAMIC MODEL

Air motors work in open-close thermodynamic cycles, because of the nature of the admission and exhaust processes, (Papson et al, 2019). It is therefore difficult to come up with a simple thermodynamic description of the states involved in the process. Non-permanent, non-equilibrium conditions exist during the process, due to turbulence and mixture that takes place inside the cylinder of the machine, even though no combustion takes place. In this sense an approximate simple approach is presented here in the intent of qualifying the performance of the motor. In the procedure some parameters related to the settings are introduced to allow some amplitude to the solution.

A sketch of the cycle representation is presented in figure 3. It comprises an admission phase occurring during the short time the admission valve is open. It is followed by an expansion when the piston in impelled by the high pressure from the air of the supply tank. Close to the end of the ride, a lateral exhaust valve opens, for the exhaustion of air, with return of piston back to initial position coming next.

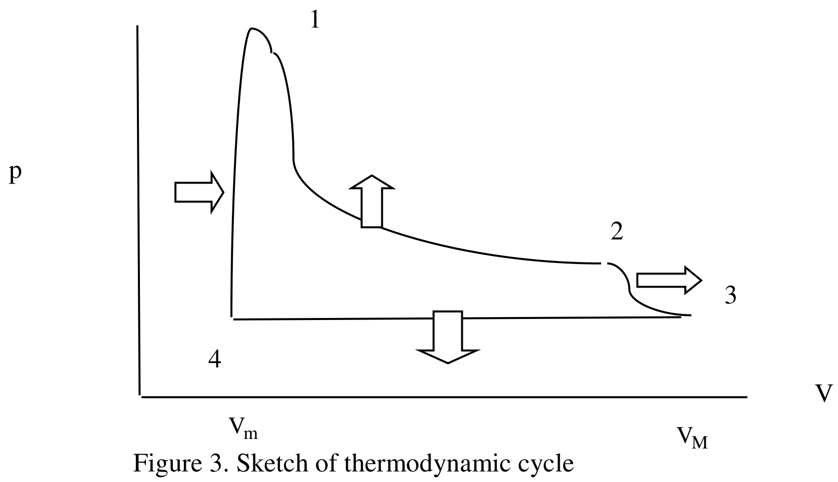

Figure 3. Sketch of thermodynamic cycle

2.1 Air Admission

The admission of air into the cylinder, sector 4-1 in the diagram, starts with the opening of the admission valve, around the top dead position of the piston motion, where volume is minimum, V_m. At that point, the air remaining inside the cylinder, from last cycle, has mass $\rho_4 V_4$ while at the end of admission, total mass is $\rho_1 V_1$. The energy equation for a control volume $V_{cv} = V_m$ in rate form is (Sontag and wan Wylen,2005):

$$\dot{Q}_{CV} + \dot{m}_i(h_i + k_i + p_i) = \dot{E} + \dot{m}_e(h_e + k_e + p_e) + \dot{W}_{CV} \quad (1)$$

Air from reservoir has internal energy density μ_o, pressure p_0 and specific volume v_o. Heat flux rate crossing the control volume is $\dot{Q}_{CV}$ and $\dot{m}_i$ the mass influx rate. Efflux quantities are denoted with an e-subscript. $\dot{W}_{CV}$ is the boundary work rate. Specific enthalpy $h = \mu + pv$, kinetic energy k and potential energy p appear in the flow terms. Once integrated, up to some virtual t in the process, $t_4 \le t \le t_1$, it leads to:

$$ {}_4Q_t + m_i(t)h_i(t) = {}_4E_t + m_e(t)h_e(t) + {}_4W_t \tag{2} $$

where kinetic and potential energy terms were disregarded, and

$$ {}_4E_t = \rho_t V_t \mu_t - \rho_4 V_4 \mu_4 \tag{3} $$

The complete integrated equation, interval 4 to 1, is:

$$ {}_4Q_1 + \alpha\Delta m(h_o) = (1-\alpha)\Delta m(\mu_1 - \mu_4) + \alpha\Delta m h_1 + {}_4W_1 \tag{4} $$

The work term includes the boundary ${}_1W_4{}^b = \langle p \rangle (V_1 - V_4)$, being the average intake pressure $\langle p \rangle = \frac{p_1 + p_4}{2}$ plus some lost work due to irreversibility in the admission process - turbulence and mixture. These contributions are assumed small. The volume at the end of admission is $V_1 = V_m(1+\beta)$ being β an admission valve control measure, related to some crankshaft angle.

Mass flow conservation requires that at the end of process, 1:

$$ m_4 + (m_i - m_e)_{t_4}^{t_1} = m_1 \tag{5} $$

This is, the mass $m_1 = \rho_1 V_1 = \Delta m$ inside the cylinder after charging minus the remaining mass in the cylinder in 4, $m_4 = \rho_4 V_4 = (1-\alpha)\Delta m$, equals the mass admitted into the cylinder $\alpha\Delta m$. Variable α is dependent upon the valve opening settings. This mass fed is also equal to the air consumed from the reservoir $\rho_0 Q_{0\,t_4}\Delta_{t_1}; Q_0 = A_i v_i$ at every cycle. Here the influx volume flow rate is Q_0, the influx valve cross sectional area is A_i and influx velocity is v_i. The reservoir conditions are supposed fixed and fully known.

Introduction of this result into Eq. (4), brings the new form:

$$ {}_4q_1 + \alpha h_o \cong (1-\alpha)(\mu_1 - \mu_4) + \alpha h_1 + {}_4w_1 \tag{6} $$

with quantities now referred to the working mass Δm.

In case air behaves like a perfect gas, specific internal energy depends only on temperature, $\mu_t = C_v T_t$, where $C_v(T)$ is the specific heat at constant volume. Taking the approximated value C_{vo} - at zero pressure - it turns out that:

$$ {}_4q_1 + \alpha h_0 \cong (1-\alpha)C_{vo}(T_1 - T_4) + \alpha(C_{vo}T_1 + p_1 v_1) + {}_4w_1 \tag{7} $$

with $p_1 v_1 = R_{air} T_1$ for a perfect gas, being $R_{air} = \frac{R}{M_{sir}}$ with R the universal constant of gases, and M_{air} the mass of a mole of air. This is an upper value of the of the mass transferred term, as the counter pressure has been ignored.

2.2 Expansion Phase

While the admission process is characterized by variable mass, open condition, the expansion process occurs with constant mass in the cylinder, Δm. This phase starts at state 1 and goes on up to the opening of the exhaust valve, state 2, for two stroke machines - in the thermodynamic diagram, $t_2 \le t \le t_3$.

The expansion depends very much on the construction of the motor and its operation. A wide range of behavior may be explained assuming a polytropic behavior for the air, $pV^n = \lambda$, being n the polytropic exponent. According to the value of the exponent n, isobaric, isothermal, isomeric and isoentropic expansions may be modeled. In the case of air, the exponent n is constant for a large range of temperatures. Under these circumstances, states at initiation and ending of expansion may be related by (Reynolds and Perkins, 1977):

$$p_2 = p_1[\frac{V_1}{V_2}]^n \tag{8}$$

In this expression, the volumes involved are known: the closure of the admission valve is at V_1 and the opening of the exhaust valve is at V_2, being $V_2 = V_M(1-\delta)$ with the maximum volume V_M corresponding lower dead point of the piston stroke and δ an exhaust valve control variable. These volumes may be related to specific positions of the piston. In the first, distance Δ from cylinder head, so that $V_m = A_p\Delta$, being A_p the piston area. For the second, $V_M = A_p(\Delta + L)$, being L the length of the connecting rod. Therefore, if:

$$f = \frac{V_1}{V_2} = \frac{(1+\beta)}{(1-\delta)r} \tag{9}$$

where $r = \frac{V_M}{V_m} = \frac{L}{\Delta} + 1$ is the stroke relation. Introduction of the above into Eq. (8) defines the pressure ratio in the expansion process. Having that, temperatures may be obtained at once. At 1:

$$p_1 v_1 = R_{air} T_1 \tag{10}$$

with $v_1 = \frac{V_1}{\Delta m} = \frac{V_m}{\Delta m}(1+\beta) = v_m(1+\beta)$. Likewise at 2:

$$p_2 v_2 = R_{air} T_2 \tag{11}$$

where $v_2 = \frac{V_2}{\Delta m} = \frac{V_M}{\Delta m}(1-\delta) = v_M(1-\delta)$. From these results it is clear that:

$$T_2 = T_1(1+\beta)^{n-1}[(1-\delta)r]^{1-n} = T_1 f^{1-n} \tag{12}$$

Therefore, the work produced in the expansion process may now be computed from:

$${}_1W_2 = \int_{V_1}^{V_2} p dV; \quad p = \lambda V^{-n} \tag{13}$$

what leads to:

$${}_1W_2 = \frac{p_2 V_2 - p_1 V_1}{1-n} \tag{14}$$

As a close process is involved, first law of thermodynamics, ${}_1Q_2 = \Delta_1 U_2 + {}_1W_2$, may be used to find the exchanged heat per unit mass:

$${}_1q_2 = C_{vo}(T_2 - T_1) + \frac{p_2 v_2 - p_1 v_1}{(1-n)} \tag{15}$$

It is worth mentioning that when a counter pressure exists in the expansion process, relative pressures to the atmospheric pressure appear above.

2.3 Exhaust Phase

The exhaust process resembles the admission process. For some types of valves, it starts with when the piston uncovers a lateral exhaust valve, in general just after point 2, and ends when the maximum displacement of the piston is attained, point 3, volume V_M when the piston motion inverts.

Considering a fixed control volume at the beginning of exhaustion, $V_{CV} = V_2$, from Eq. (1) heat exchange, mass variation and work are involved so that:

$${}_2Q_t = E_t - E_2 + m_e(t)h_e(t) + {}_2W_t; \quad t_2 \le t \le t_3 \tag{16}$$

The work required to expel the mass $\alpha\Delta m$ to the atmosphere, flow work, has an increment $\partial\delta W^f = (p - p_a)A_e v_e \partial t = (p - p_a)Q_e \partial t$. Symbols follow influx notation. Being $v = \rho^{-1}$ the specific volume, $Q_e \partial t = v\partial m_e$. Only pressure work is considered, no shear effort. Like before, turbulence and mixture takes places, with boundary and lost work occurring.

In the process separation takes place, with the part $\alpha\Delta m$ of mass being thrown out and another, $(1-\alpha)\Delta m$, remaining in the cylinder. Upon considering integration from 2 up to 3, Eq. (16) reads:

$$ {}_2Q_3 = (1-\alpha)\Delta m\mu_3 - \Delta m\mu_2 + \alpha\Delta m h_3 + {}_2W_3 \tag{17}$$

Expanding the internal work terms - whole $2 \rightarrow 3$ exhaust interval:

$$ {}_2Q_3 = \Delta m C_{vo}(T_2 - T_3) + \alpha\Delta m p_3 v_3 + {}_2W_3 \tag{18}$$

As $p_3 v_3 = R_{air} T_3$:

$$ {}_2q_3 = C_{vo}(T_3 - T_2) + \alpha R_{air} T_3 + {}_2w_3 \tag{20}$$

A lower limit for the exhaust pressure is that $p_3 = \zeta p_a, \zeta \geq 1$, for no suction, being p_a the atmospheric pressure and ζ an exhaust pressure factor. Under this condition, constitutive equation for perfect gas requires that in 3:

$$ T_3 = \frac{\zeta p_a}{(1-\alpha)R_{air}} v_M \tag{21}$$

where $v_M = \dfrac{V_M}{\Delta m} \neq v_3$

2.4 Compression Phase

Final phase of the work cycle starts when the air $(1-\alpha)\Delta m$ remaining in the cylinder is compressed in the return motion of the piston – with the exhaust valve closed by the controller, up to top. Again assuming polytropic behavior in $3 \rightarrow 4$ stretch allows a broad representation of possible compression representations.

$$ p_3 V_3^{\,n} = p_4 V_4^{\,n} \tag{22}$$

so that $p_4 = r^n p_3$ as $r = \dfrac{V_3}{V_4} = \dfrac{V_M}{V_m}$. From it, temperature at end of compression may be obtained easily as:

$$ T_4 = \frac{\zeta p_a v_4}{R_{air}}; \quad v_4 = \frac{V_m}{(1-\alpha)\Delta m} \tag{23}$$

As temperature T_3 is usually low, refrigeration processes may be coupled to this cycle, with cogeneration. Defined these quantities, the work required to compress the air in this phase is:

$$ {}_3W_4 = \frac{r^n V_m - V_M}{(1-n)} p_3; \quad p_3 = \zeta p_a \tag{24}$$

Again from first law for the compression step:

$$ {}_3Q_4 = \Delta_3 U_4 + {}_3W_4 \tag{25}$$

Being the internal energy density of a perfect gas dependent only on the temperature $\mu = \hat{\mu}(T)$:

$$\Delta_3 U_4 = (1-\alpha)\Delta m(\mu_4 - \mu_3) \tag{26}$$

Resulting that the heat inflow will be:

$$_3q_4 = [C_{vo} + 1](T_4 - T_3) \tag{27}$$

with C_{v0} the constant volume specific heat of the air.

2.5 Energy Balance

For the continuous part of mass, $(1-\alpha)\Delta m$, mass that even though renewed at every cycle, is permanently in the same thermodynamic conditions and amount, for fixed running conditions of the engine and environment, first law may be considered in close cycle to give (Holman, 1980):

$$(1-\alpha)\Delta m[_1q_2 + _2q_3 + _3q_4 + _4q_1] = (1-\alpha)\Delta m[_1w_2 + _2w_3 + _3w_4 + _4w_1] \tag{28}$$

Upon addition of the respective equations - equations (15), (20), (27) and (7) - it turns out that:

$$T_1 = \frac{h_o}{R_{air}} - [1 + \frac{C_{vo}}{R_{air}}]T_3 \tag{29}$$

The value of temperature T_3 cannot be found unless the parameters of the machine are known, mostly air consumption rate and power. If the shaft power of the engine is P_m when running at n rotations per minute, then the time period of each cycle is $\tau = N^{-1}$; $N = \frac{n}{60}$. Hence in one cycle the resulting work, net, produced is $W_{net} = {}_1W_2 + {}_2W_4$:

$$W_{net} = \frac{P_m \tau}{\eta_m} \tag{30}$$

where η_m is the mechanical efficiency. Expansion and compression work expressions, derived above, may be used to find maximum pressure p_1 as a function of exhaust pressure $p_3 = \zeta p_a$. From Eqs, (14) and (24), upon substitution of the constitutive equation for perfect gas, it results that:

$$\frac{P_m \tau}{\eta_m} = \frac{f^n V_2 - V_1}{(1-n)} p_1 + \frac{r^n V_4 - V_3}{(1-n)} p_3 \tag{31}$$

Once found p_1, for the specified consumption, temperature T_1 results:

$$T_1 = \frac{(1-n) P_m \tau \eta^{-1}{}_m - (r^n V_4 - V_3)\zeta p_a}{R_{air}(f^n V_2 - V_1)\alpha\Delta m} V_1 \tag{32}$$

The consumption of air C is measured in terms of the volume of air, generally taken in liters, per minute at 20 ºC and one atmosphere of pressure. An accuracy of about 5% is required. It is represented as Nl/min. Hence it may be written in terms of the inflow rate from the supply tank, converted to the required conditions, as:

$$\alpha\Delta m = \frac{60C}{1000} \tau \rho_{npt} \tag{33}$$

with the density ρ of air taken at reference conditions of temperature and pressure.

A measure of the thermodynamic performance of the motor may be set by the ratio of the useful work produced against the energy required:

$$\eta_t = \frac{W_{net}}{_4Q_1} \tag{34}$$

3. RESULTS AND ANALYSIS

The thermodynamic model for the ain driven motor presented above may be coded in computer to allow fast determination of the temperatures and pressures involved, for different settings of the machine. This was done with help of a matrix solver, (MATLAB, 2020). The results depend on the size, power, operation conditions and parameters put forth in each case. Table 1 shows typical values for an air motor based on a two stroke cycle, with one cylinder, under constant turning velocity and fixed settings.

Table 1: Engine Specifications

VARIABLE	VALUE
Bore D_P	200 mm
Stroke L	100 mm
Consumption C	3.141 l
Intake valve diameter D_i	20 mm
Exhaust valve diameter D_e	20 mm
Compresion ratio r	10
Mechanical Power P_m	8400 W
Rotational Velocity n	1800 RPM

3.1 Application

The engine specified in Tab. 1 is considered as demonstration of a mixed open-close cycle. For it, maximum volume in the cylinder is found first, $V_M = A_p L$, being the piston area $A_p = \frac{\pi D_p^2}{4}$. Minimum value, on the other hand is $V_m = A_p \Delta$. Both volumes are related by the compression ratio, $r = \frac{V_M}{V_m}$, meaning than that $r = \frac{L}{\Delta} + 1$, what determines $V_m = 3.141e-4\ m^3$, with V_M ten times this value. The other important volumes, the volume when admission valve closes, $V_1 = (1+\beta)V_m$, and the exhaust initiation volume, $V_2 = (1-\delta)V_M$, follow easily. Values of $\beta = 0.2$ and $\delta = 0.3$ were chosen.

Thermodynamic properties of air required for the computations of specific heat at constant pressure and volume, as a function of temperature, in case of coding, considered that $C_v = C_p - R_{air}$ according to the expression, for absolute temperature T:

$$C_v = 1.9327e-10T^4 - 7.9999e-7T^3 + 1.1407e-5T^2 - 4.4890e-1T + 770.45 \tag{35}$$

For approximate estimations, the constant C_{vo} of air at one atmosphere can be considered, as well as the values of molecular weight and universal constant of gases as shown in table 2 (Air Properties, 2022)

Table 2: Air Properties

PROPERTY	VALUE
Specific heat C_{vo}	718 $JKG^{-1}K^{-1}$
Air constant R_{air}	287.042 $JKG^{-1}K^{-1}$
Polytropic exponent	1.4
Atmospheric pressure p_a	1.013250e+5 Pa

Working conditions are fixed by control set $<\beta, \delta, \zeta, \alpha>$. With $\eta_m = 0.90$, from values in Table 1, the work per cycle is $W_{net} = 311.1$ Nm, Eq. (30). The consumption is $\alpha\Delta m = 3.871e-3$ kg also per cycle, eq. (33). From the relationship Eq. (31), a value of maximum pressure $p_1 = 34.7$ atm is found, associated to $\zeta = 1.2$. The maximum temperature $T_1 = 1221.05$ K, from Eq. (10). Temperature $T_2 = fT_1 = 601.9$ K, Eq. (12), is obtained, with associated pressure $p_2 = 2.93$ atm. Atmospheric pressure is 1.013250e+5 Pa, or 1 bar.

Completing the operating setting with $\alpha = 0.4$, for example, an operating mass of air $\Delta m = 9.454e-3$ kg is needed. With that, exhaust temperature $T_3 = 234.6$ K is gotten, Eq. (21). When compression is completed, $p_4 = r^n p_3 = 30.14$ atm, The corresponding temperature is $T_4 = 589.2$ K. The required supply conditions at the storage tank is found from El. (29): $h_0 = 586.2e+3$ J/kg, with associated temperature $T_0 = 583.3$ K.

Instead of machine variables determining the required supply conditions, the contrary may be used. Larger machines would employ much higher pressures to obtain more work.

3.2 Comparison

Two thermodynamic cycles, Ericsson and Stirling, are normally considered as acceptable descriptors of the operation of alternative air machines. Many times the results produced by them is not very good as most machines do not operate under the ideal conditions they preconize. Here the Stirling cycle will be considered for comparison purposes.

In Figure 3, a schematic drawing of a Stirling cycle operating between high T_h and low T_l temperatures is shown. In this cycle the admission process when heat is taken in by the working fluid, 4-1, occurs at constant volume. It is followed by the expansion phase, 1 to 2, which occurs at a constant temperature, T_h. The exhaust phase occurs, as the admission, at constant volume, 2 to 3, when temperature decreases to T_l. Cycle is completed with the compression phase, 3 to 4, at constant low temperature.

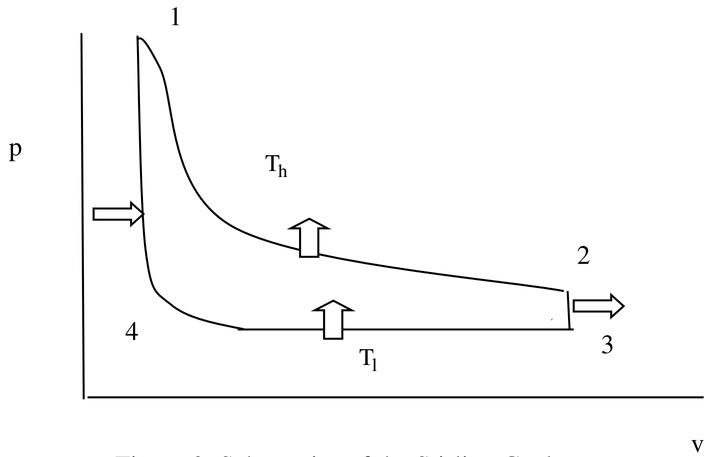

Figure 3: Schematics of the Stirling Cycle

The heat influx per unit mass, from low T_l to high T_h temperature, in the admission process, for the mass that is held inside the cylinder is obtained from:

$$ {}_4q_1 = C_{vo}(T_h - T_l) \tag{36}$$

which cancels out the one for the exhaust process:

$$ {}_2q_3 = C_{vo}(T_l - T_h) \tag{37}$$

In both cases there is no work – constant volume process. For the expansion process, at constant temperature T_h, the work output is found from ${}_1w_2 = \int_{v_2} p dv$ with $p = c_h v^{-1}; c_h = R_{air} T_h$ to give:

$$ {}_1w_2 = c_h \ln\frac{v_2}{v_1} \tag{38}$$

with identical form for the compression phase, where constant is c_l, ${}_3w_4 = c_l \ln\frac{v_4}{v_3}$, being again $v_4 = v_1 = v_m$ and $v_2 = v_3 = v_M$. The total work output is the sum of these two terms. Heat exchange also includes ${}_1q_2 = T_h(s_2 - s_1)$ for the high temperature part. The entropy densities are shown as s. For a perfect gas, $s_2 - s_1 = C_v \ln\frac{T_2}{T_1} + R_{air} \ln\frac{v_2}{v_1}$, with $T_1 = T_2 = T_h$. For the compression sector, the form of expression applies at lower temperature, T_l. The efficiency will consider the total work in each cycle, $w_{net} = {}_1w_2 + {}_3w_4$ against the payed for heat:

$$\eta_S = \frac{w_{net}}{{}_4q_1} \quad (39)$$

The procedure for finding the temperatures involved here is simpler, as it is a close cycle. If the same resultant work is considered, for the same mass of operating air, appreciable differences in temperature are obtained in comparison with the proposed approach presented above. In particular, high temperature $T_h = 317.6$ K and $T_l = 234.6$ K. The pressures are going to be, $p_1 = 16.24$ atm, with $p_2 = \frac{p_1}{10}$. Also $p_3 = 1{,}2$ atm, which is the same as p_4. The efficiency is about 0.23.

4. CONCLUSIONS

The presented procedure shows good versatility as it is able to model various settings for the opening and closing times of the valves in the motor. Variables β and δ may be related to angular positions of the injection/exhaust valves. The model should help in the design process when used in numerical, coded, format, mostly. It allows for optimization of variables with development of specific expressions, as well. However experimental verifications should be conducted before deciding which combinations of polytropic alternatives best compares with reality, for different motor arrangements and designs. Comparison with Stirling cycle shows, in the example considered, different $< p, v >$ and $< T, s >$ domains.

5. REFERENCES

Salvi, B. and Subramanian, K., 2016, "Sustainable Development of Road Transportation Sector Using Hydrogen Energy System", *Renewable and Sustainable Energy Reviews*, vol. 51, pp. 1121-1155

Nègre et al, 2015, "Engine with Active Mono-Energy and/or Bi-energy Chamber with Compressed Air and/or Additional Energy and Thermodynamic Cycle Thereof", US Patent, 746952B2

MDI, Moteur Development International, https://www.mdi.lu/

Papson, A., Creutzig, F, and Schipper, L, 2019, *Compressed Air Vehicles:Drive-Cycle Analysis of Vehicle Performance, Environment Impacts and Economic Costs, Transportation Research Record 2191, pp.67-74*

Sontag, R.E. and Van Wylen,G.J, 2005*, Fundamentals of Classical Thermodynamics, John Wiley & Sons*, 4th edition, Michigan, USA

Reynolds, W.C. and Perkins, H.C, 1977, *Engineering Thermodynamics, McGraw-Hill Book Co., 2nd edition New York, USA*

Holman, J.P., 1980, *Thermodynamics*, *McGraw-Hill Book Co., 3rd edition., New York*, USA

MATLAB, 2020, *MathWorks Inc., R2020 in* https://www.mathworks.com/

Air Properties, Online Calculator, https://peacesoftware.de/einegewerte/luft_1.html

6. RESPONSIBILITY NOTICE

The authors are responsible for the printed material included in this paper.