Problems of conversion of diesel locomotives to gas-diesel cycle

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Abstract. The article focuses on the problem of ensuring the required parameters for gas-air mixture in light load and idle modes. A mathematical modeling of the process in the gas-air duct of the diesel generator 1-PDG-4 was conducted during air blow-off in compressor and throttling at the diesel inlet. It was found that conversion of locomotive engines to gaseous fuel at idle speed can meet the required stoichiometric ratio of gas-air mixture only if air throttling processes at the engine inlet are combined with the engine in the shut-down mode of a cylinder bank.

1. Introduction
The problem of introduction of gaseous motor fuel to locomotives becomes particularly relevant. When diesel locomotives are converted to gas, the low level of diesel substitution by gas becomes a problem, which crucially reduces efficiency of this conversion especially for shunting locomotives. It fails to substitute diesel fuel for more than 40-50\% by straight switch to gas fuel during gas-diesel cycle [2-4]. The main challenge is that no stable ignition of the spark fuel and gas-air mixture can be reached in light-load and idle modes. Current article highlights the required parameters of gas-air mixture that are necessary in these work modes.

The required stoichiometric ratio of gas-fuel mixture in these modes can be mainly achieved by air throttling at the diesel inlet. By doing so may cause a need to cut the number of the operating cylinders. However, it is not clear, what temperature of the gas-air mixture will be at the beginning of compression; how much will the temperature change due to inverted scavenging of exhausts into the cylinder; what stoichiometric ratio should be maintained for a consistent ignition from spark fuel? To date, there is no mathematical model, which is able to answer the questions when simulating diesel work modes under operation. This paper is an attempt to tackle some of the questions through mathematical modeling.

2. Materials and methods
The goal of the mathematical modeling for the process in gas-air duct of the diesel generator 1-PDG4, during air blow-off in compressor and throttling at the diesel inlet, is to assess technical and economic indicators of a locomotive under operation.

The program is based on the software for calculation of work indicators of a locomotive diesel generator in steady modes and transient processes that was developed in Russian University of

\begin{center}
\includegraphics[scale=0.1]{figure.png}
\end{center}
The need for some refinement of the program is due to the fact that when converting diesel locomotive engines to gaseous fuel, it is necessary to maintain a given stoichiometric ratio of air and gas mixture at all operating modes. Operating at idle speed, it can only be met if air throttling processes at the engine inlet are combined with the engine in the shut-down mode of a cylinder bank. These processes are poorly known and demand further development.

A specific feature of the methodology developed in MIIT [5] is that for a system of differential equations describing the movement of a train and a temporary change in the state of the power plant of a diesel locomotive (rotational speeds of the crankshaft and turbocharger rotor, changes in the temperature of parts and other time-dependent indicators), the parameters of the state of the working fluid of the engine in its gas-air path and related units are determined at each integration step. The concept dwells on the dependencies of parameters of the working fluid (air and exhaust gases) on its consumption in the components of gas-air duct and engine cylinders, which are found at the intervals of integration of differential equations, considering constant variables, which change in time. Unlike standard practices [6,7], we consider processes of filling and exhaust independently, as separate processes.

Design circuit of engine equipped with systems of compressor blow-off and air throttling at the engine inlet is shown in Figure 1.

![Figure 1](image-url)

**Figure 1.** Design circuit of gas-air duct of diesel PD1M 1- filter; 2 - turbo-charge compressor; 3 - air bleed valve; 4 - charge air cooler; 5 - butterfly throttle; 6 - inlet manifold; 7 - admission valve; 8 - cylinder volume at the end of filling; 9 - cylinder volume at the beginning of exhaust; 10 - exhaust valve; 11 - exhaust manifold; 12 - exhaust manifold after nominal diaphragm; 13 - turbine; 14 - exhaust muffler.
In our mathematical model, the system of algebraic equations describing the initial conditions for solving differential equations is supplemented by equations of the state of working fluid in newly introduced devices and equations describing the filling and exhaust processes.

Some authors [8,9] assume that compressor air bleed is necessary to prevent surging mode. After air enters the filter 1 \((P_1, T_1)\) with the flow rate \(C_v\), the flow rate \(G_3\) from the air bleed after the compressor through valve 3 is added to the flow rate at the inlet to the compressor \(G_2\). The compressor inlet temperature rises in accordance with the heat balance:

\[
T_1 = \frac{(G_v \cdot T_0 + G_3 \cdot T_2)}{(G_v + G_3)},
\]

(1)

Due to small temperature changes, air heat capacity is deemed constant. We ignore the pressure changes at the compressor inlet. Let us find the air flow rate when bleeding:

\[
 G_3 = F_3 \cdot m_3 \cdot \sqrt{\frac{2 \cdot R_v \cdot T_2 \cdot \frac{K_v}{(K_v - 1)} \cdot P_2}{P_0}}
\]

(2)

\[
 G_2 = G_v + G_3
\]

\[
 T_1 = \frac{(G_v \cdot T_0 + G_3 \cdot T_2)}{G_2}
\]

Where \(F_3 \cdot m_3\) is effective section of the air bleed device 3;

\(T_0, T_1, T_2\) are temperatures of the working fluid at the inlet of compressor and filter in the air bleed device 3.

Calculation of the work indicators and air parameters after compressor remained the same, comparing to the previous program.

Throttle 5 is added to the design circuit (Figure 1). The working fluid goes to the inlet manifold 6 after throttle 5. Let us assume that the air temperature remains constant when throttle pressure changes. Pressure ratio in throttle:

\[
P_{15} = \frac{1}{\left(\frac{R_v \cdot T_4 \cdot G_v^2}{m_{f_5}^2 \cdot P_{f_5}^2 + P_4^2} + 1\right)}
\]

(3)

\[
P_5 = P_4 \cdot P_{15}
\]

\[
T_5 = T_4
\]

Where \(P_4, P_5\) are pressures after air cooler 4 and the throttle;

\(T_4, T_5\) are temperatures after air cooler and the throttle;

\(m_{f_5}, F_5\) is effective section of the throttle.

Air scavenging processes (inlet and exhaust valves are open) and filling.

When considering gas exchange processes (inlet and exhaust), the following assumptions are made:

- heat exchange processes between cylinder walls and working fluid are ignored;
- difference in the heat capacities of the working fluid due to changes in temperature and composition of the mixture of air with exhaust gases is ignored;
- piston movement during overlapping of inlet and exhaust valves is not taken into account;
- it is accepted that the final pressure in the compression chamber corresponds to the pressure \(P_6\) in the inlet manifold during direct scavenging and to the pressure \(P_{11}\) in the exhaust manifold during inverted scavenging;
- it is assumed that the working fluid is completely mixed during the air scavenging of the cylinder;
- mass of the working fluid, which goes to the exhaust manifold during direct scavenging, is assumed to be equal to the scavenging mass deducted by the value of the mass change of the working fluid in cylinder.
Depending on the operating mode, there can be a “direct” ($P_6 > P_{11}$) and an “inverted” ($P_6 < P_{11}$) scavenging. Consider both options.

Direct air scavenging ($P_6 > P_{11}$)

Let us accept that the amount of the working fluid and its temperature at the beginning of the filling process correspond to the amount and temperature of the working fluid calculated at the end of the release in the previous integration step. In accordance with the pressure difference in the inlet and exhaust manifolds, the working fluid flows from the inlet manifold into the cylinder, mixes with residual gases, and part of the resulting mixture goes into the exhaust manifold 11. Scavenging flow rate is determined by the effective section of the valves. It complies with the parameters of the fluid in the inlet manifold and calculated by the pressure ratio $P_6/P_{11}$.

$$G_{\text{prod}} = m_f F_7 \sqrt{2 R_v T_5 \frac{K_v}{K_v - 1} \frac{P_6}{P_{11}}}$$  \hspace{1cm} (4)

where $m_f F_7$ is the average effective section of inlet and exhaust valves when scavenging.

The working fluid mass flowing into the cylinder and exhaust manifold during the scavenging will be:

$$M_{\text{prod}} = G_{\text{prod}} f_p \cdot \frac{2 \pi_i}{W_d \cdot 720}$$

where $f_p$ is angle of simultaneous opening of the valves in the crankshaft rotation degrees.

The temperature of the working fluid in the cylinder at the end of the scavenging and at the beginning of the filling shall be determined from the heat balance, considering the mixing of air and residual gases to be sufficiently complete.

$$T_7 = \frac{(M_w C_p T_{11} + M_{\text{prod}} C_p T_6)}{(M_w C_p + M_{\text{prod}} C_p)}$$  \hspace{1cm} (5)

Determine the amount of the working fluid when filling begins:

$$M_7 = \frac{V}{\varepsilon \cdot R} \cdot \frac{P_6}{T_7}$$

where $V$ is cylinder capacity (actual displacement); $\varepsilon$ is compression ratio.

Let us determine mass of the working fluid, coming during a cylinder scavenging to exhaust manifold, and the flow ration through all cylinders when scavenging:

$$M_p = M_{\text{prod}} (M_7 - M_9)$$

$$G_{\text{prod}} = \frac{M_p N_n W_d}{2 \pi_i \tau}$$

where $W_d$ is the crankshaft rotation velocity in rad/s, $N_n$ is the number of engine cylinders.

Inverted air scavenging $P_6 < P_{11}$

Assume that in the event of inverted air scavenging, the working fluid enters the inlet manifold 6 from both the throttle 5 and the exhaust manifold. Temperature of the working fluid in the inlet manifold is determined by its flow rate through the throttle 5 and cylinder when scavenging and by the heat balance, assuming the temperature value in the exhaust manifold at the previous step to be known.

Amount and parameters of the working fluid in the cylinder when scavenging begins are equal to its parameters at the end of exhaust:

$$M_7 = M_9$$

$$G_{\text{prod}} = m_f F_7 \sqrt{2 R_g T_{11} \frac{K_g}{K_g - 1} \frac{P_{11}}{P_6}}$$  \hspace{1cm} (6)
where $T_{11}$ and $P_{11}$ are parameters of working fluid in exhaust manifold. Mass and flow rate of the fluid coming from the exhaust manifold during scavenging will be:

$$M_{pr} = G_{prod} f_p \frac{2 \pi_i}{W_d} \frac{20}{720}$$

$$G_{pro} = \frac{G_{prod} f_p N_o}{720}$$

Temperature of the working fluid in cylinder is taken even to the one in exhaust manifold $T_7 = T_{11}$. Temperature in the inlet manifold is calculated from the heat balance.

$$T_6 = \frac{G_v T_5 + G_{pro} T_{11}}{G_v + G_{pro}}.$$

Determine the amount of the working fluid in compression chamber when intake begins:

$$M_7 = \frac{V}{\varepsilon - 1} \frac{P_{11}}{R_v T_7}.$$

Filling

The filling process is outlined in Figure 2.

**Figure 2.** Design circuit for filling process during direct (a) and inverted (b) scavenging.

$P_2$ is the pressure after the compressor; $P_5$, $P_6$ are pressures after the throttle and in the receiver; $P_7$ is the pressure in the cylinder; $P_{11}$ is the pressure in the exhaust manifold; $V/(\varepsilon - 1)$ is the volume of the compression chamber; $V'(1-1/\kappa)$ is conditional volume that working fluid will take after cooling due to the filling work.

Let us assume that working fluid enters the cylinder from the exhaust manifold at the filling stroke during direct scavenging (Figure 2a) and inverted scavenging (Figure 2b) with temperature $T_6$ and pressure $P_7$, which is reduced by valve section losses in comparison to $P_6$:

$$P_{16} = \frac{R_v T_6 G_v^2 P_6^2}{\omega \epsilon^2 - F_6^2} + 1;$$

$$P_7 = P_6 P_{16}.$$
Within the piston movement from TDC to BDC, the ingoing fluid performs work when filling, which is related to the piston displacement at constant pressure. Let us calculate the mass of the working fluid, performing this work, and its temperature after completing the work:

\[ M_{81} = \frac{V \cdot P_0}{R_c \cdot T_7} \]
\[ T_{81} = T_7 - \frac{P_0 \cdot V}{C_{pv} \cdot M_{81}} \]

Since \( M_{81} = \frac{P_2 - P_1}{k} \) and \( C_{pv} = k/(k-1) \cdot R \), then:
\[ T_{81} = \frac{1}{k} \cdot T_7. \] (10)

Volume, which mass \( M_{81} \) will take up after work completion, will be:
\[ V_{81} = V \cdot \frac{T_{81}}{T_7} = V \cdot \frac{1}{k} \]

The freed volume when «charging up» will be filled with new charge:
\[ M_d = \left( 1 - \frac{1}{k} \right) \frac{V \cdot P_2}{R_c \cdot T_7} \] (11)

and the mass of the working fluid in cylinder at the end of filling will be:
\[ M_a = M_f + M_{dd} + M_d \]

Ignoring the heat transfer from cylinder walls to incoming charge, let us find the fluid temperature at start of the compression in compliance with the heat balance:
\[ T_a = \frac{M_f \cdot T_7 + M_d \cdot T_5 + M_{81} \cdot T_{81}}{M_a}. \] (12)

2.1. Free and forced exhausts

Processes of free and forced exhausts are outlined in Figure 3.

The real process is drawn with a dashed line. Solid lines are design process. Line AB represents free exhaust and line BC - forced one, if the pressure at the end of expansion in the operating cycle is greater than the pressure in exhaust manifold (Figure 3a). It is taken that pressure of the working fluid in AB go down and its temperature does not change. According to the pressure change, a portion of the mass of the working fluid flows through the manifold 11 to the turbine 12 in the form of a short pulse \( M_9 \) (Figure 1). It is assumed that duration of the pulse corresponds to the time of initial opening of the exhaust valve. In process BC piston 9 forces the fluid out, performing the work at constant pressure value.

Part of the working fluid will flow from the exhaust manifold into the cylinder when the pressure of the working fluid at the end of the expansion is lower than the pressure in the exhaust manifold (with the cylinder turned off or with deep throttling) and when the exhaust valve is opened (Figure 3b). This process was not taken into account in our calculations and the cylinder pressure was assumed equal to the pressure value at the end of compression. In process BC piston 9 forces the fluid out, performing the work at constant pressure value and the work of adiabatic compression to pressure level in the exhaust manifold.

Determine the mass of the working fluid at the beginning of exhaust valve opening:
\[ M_9 = M_d + q_s \]

where \( q_s \) is fuel delivery per stroke.

Calculate temperature \( T_v \) and pressure \( P_9 \) in the cylinder at the end of expansion in operating mode (A, Figure 3):
\[ T_g = T_a + \frac{H_u q_z (1-N_{kpdi} - et - cnt) \varepsilon}{C_p \varepsilon M_a (\varepsilon - 1)} \]  

where \( H_u \) is fuel lower calorific value, 
\( q_z \) is fuel delivery per stroke, 
\( N_{kpdi} \) is indicated efficiency, 
et is heat availability factor, 
cnt is factor for unburnt fuel.

\[ P_g = M_g R_g \frac{T_g (\varepsilon - 1)}{V_e} \]  

Exhaust process will be different depending on the cylinder pressure ratio.

**Figure 3.** Design circuit for the exhaust processes at the end of expansion (A) when the pressure is greater in the exhaust manifold (C) (a) and when it is less (b). A - pressure at the end of expansion; B - cylinder pressure; \( P_0 \) - atmospheric pressure; C - pressure in the exhaust manifold.

At the cylinder pressure above the manifold pressure, which is known from the previous calculation, determine the mass of the pulse of the fluid at one-cylinder free exhaust.

\[ M_{0i} = M_g P_{10} / P_g \]

The flow rate of the working fluid in the pulses of all cylinders, assigned to the entire working cycle, will be:

\[ G_i = M_{0i} \frac{W_0 N_o}{2 \pi e + D_{imp}} \]  

where \( N_o \) is a number of operating cylinders;
\( D_{imp} \) is relative pulse duration.
We take \( D_{imp} = 0.07 \).

Determine the mass of the working fluid at forced exhaust

\[ M_{0f} = M_g \frac{e - 1}{\varepsilon} \cdot M_{0i} = M_g \frac{e - 1}{\varepsilon} \cdot (1 - \frac{P_{10}}{P_g}) \]  

Determine the change in the enthalpy of the working fluid during forced exhaust due to the forcing out work:
Let us find the manifold temperature through the heat balance considering the scavenging direction. We ignore the pulse influence on temperature in the manifold:

$$T_{10} = T_9 + \frac{L_9}{c_{pg}}$$

when \(P_p > P_{11}T_{11} = \frac{M_9 T_{10} M_p T_6}{M_p + M_9}$$

when \(P_p < P_{11}T_{11} = T_{10};$$

If the pressure in the cylinder at the end of the expansion is lower than the pressure in the exhaust manifold, the pressure in the cylinder shall be taken to be the same pressure at the end of the expansion. Then, mass of the working fluid, during forcing out is:

$$M_9 = (M_a + \frac{P_9}{M_9})$$.

Forcing-out work and adiabatic compression during process AC (Figure 3b), as well as the increase in the temperature of the working fluid entering the exhaust manifold, will be:

$$L_9 = \frac{P_9 V_{e}}{M_9 (e-1)} + C_{pg} T_9 (\frac{P_{11}}{P_9} \frac{k_p}{k_e} - 1),$$

$$T_{10} = T_9 + \frac{L_9}{C_{pg}}$$

Determination of pressures throughout the diesel duct is carried out in the same way as [6]. The effect of the pulse on the pressure distribution along the path is ignored. In the calculation, we take into account only work of the pulse in the turbine of the turbocharger.

Assume that the average pulse pressure is found by the following equation:

$$P_{9i} = 0.5(P_9 + P_{10}).$$

It was assumed that when the pulse moves from the cylinder to the turbine, the temperature of the working fluid in it does not change, and the pressure changes in local resistances. The pressure loss in the exhaust valve and manifold is determined for the pulse in the same way as for the main flow.

**Table 1.** Influence of throttling on the operating indicators of diesel engine 1PD4 in idle mode.

| № number | Indicator                                      | Dimension | Throttle section, m² | 0.008  | 0.002  | 0.0009 | 0.0005 |
|----------|-----------------------------------------------|-----------|----------------------|--------|--------|--------|--------|
| 1        | Crankshaft rotation velocity                  | rad/s     | 26                   | 26     | 26     | 26     | 26     |
| 2        | Fuel flow rate                                | kg/h      | 6.41                 | 6.25   | 6.15   | 6.05   |        |
| 3        | Air flow rate                                 | kg/s      | 0.41                 | 0.34   | 0.25   | 0.185  |        |
| 4        | Throttle pressure                             | Pa        | 98850                | 79521  | 58507  | 44156  |        |
| 5        | Flow rate during scavenging of working fluid  | kg/s      | 0.001                | 0.00113| 0.00133| 0.00165|        |
| 6        | Temperature of working fluid in receiver      | K         | 332                  | 323    | 326    | 334    |        |
| 7        | Mass of working fluid involved in work        | kg        | 0.0272               | 0.0218 | 0.0158 | 0.0165 |        |
| 8        | Temperature of the mass (7) due to performed work | K     | 230                   | 231    | 233    | 239    |        |
| 9        | Mass of working fluid at start of compression | kg        | 0.037                | 0.03   | 0.022  | 0.0165 |        |
| 10       | Temperature of working fluid at start of compression | K  | 258                  | 261    | 268    | 281    |        |
| 11       | Pressure at end of compression                | Pa        | 116000               | 9850   | 78750  | 65200  |        |
| 12       | Temperature at end of compression             | K         | 308.6                | 324    | 350    | 386    |        |
| 13       | Working fluid flow rate at free exhaust       | kg/s      | 0.372                |        |        | -       | -      |
| 14       | Temperature after forced exhaust              | K         | 389                  | 409    | 442    | 488    |        |
The pressure drop in the turbine of the turbocharger is calculated for both the main flow and the pulse. Based on the pressure drop, we calculate the available head and turbine power for the main flow and for the pulse in accordance with its duration.

Some calculation results.

The above provisions of the method are implemented to calculate the performance of diesel engines 1PD4 and D49 in idle mode in the program written on BASIC. As an example of calculation studies with the use of the program, Table 1 shows the indicators of the 1PD4 in idle mode depending on the degree of throttling of air at the inlet to the receiver. The calculations were carried out at a constant crankshaft speed of 250 min\(^{-1}\) with a change in the throttle section 5 (see Figure 1) from 0.008 to 0.0005 m\(^2\) at an outdoor temperature and pressure of 293 K and 101.3 kPa, respectively. Temperature of the water at the inlet of the air cooler is 47 °C (320K).

### Table 1: Performance Indicators of 1PD4 in Idle Mode

| No. | Mass of working fluid at forced exhaust (kg) | 0.0311 | 0.0278 | 0.0206 | 0.0154 |
|-----|--------------------------------------------|--------|--------|--------|--------|
| 15  | Turbine power of turbocharger (W)           | 693    | 301    | 88     | 32     |
| 16  | Power loss due to friction (W)              | 17300  | 18400  | 20340  | 21600  |
| 17  | Excess air factor                           | 17.0   | 14.0   | 10.4   | 7.8    |

### 3. Results and discussion:

Inverted scavenging is conducted in all mentioned operating modes due to pressure in receiver is lower than in exhaust manifold;

In the result of the work performed by the fluid at the filling stroke, it cools for a value T/\(k\) and «charging up» (additional filling) of the cylinder happens because of the volume drop of the working fluid before and after its cooling.

With an increase in the throttling degree, the pressure at the end of the expansion can decrease below the pressure in the exhaust manifold, and then instead of the free exhaust in the cylinder, in addition to forced exhaust, an extra compression work is performed from the pressure at the end of the expansion to the pressure in the exhaust manifold, which leads to an increase in the temperature of the working fluid and gas exchange.

### 4. Conclusion

The results are preliminary, they do not take into account the heat exchange processes between the working fluid and the cylinder walls and will be refined in further studies, but they allow us to draw certain conclusions:

- when throttling the air at the inlet to the diesel engine, the stoichiometric ratio cannot be reduced to the value required by standard [5]. Apparently, it is necessary to apply cylinder shutdown in addition to throttling;
- when performing air throttling at the engine inlet, there is no need to use air bleed in the compressor, since the turbine power drops and there is no threat of surging mode;
- if there is a charged air cooler and a high coolant temperature in all operating modes, the temperature necessary for self-ignition of the spark fuel is provided at the end of compression.

### References

[1] Gapanovich V A 2017 Vnedreniye gazomotornyh lokomotivov v OAO «RZHD» // Zheleznodorozhnnyy transport - №9 S 35-38
[2] Ekspluatatsionnyye ispytaniya gazoteplovozov TEM18G Nauchno-issledovatel'skiy institut zheleznodorozhnogo transporta (VNIIZHT)/ Otchet o nauchno-issledovatel'skoy rabote Gosregistratsiya № 01200115547 2001
[3] Nosyrev D YA, Balakin A Yu, Petukhov S A, Kurmanova L S 2016 Otsenka vliyaniya sootnosheniya uglera k vodorodu na teplofizicheskiye svoystva kompozitnykh topliv dlya raboty teplovoznih dizeley Vestnik transporta Povolzh'ya № 2(56) - S 33-38
[4] Asabin V, Roslyakov A, Kurmanova L, Petukhov S, Erzamaev M P 2020 Conversion of diesel locomotive engines to operation on natural gas motor fuel V sbornike: E3S Web of Conferences Key Trends in Transportation Innovation S 01003
[5] GOST R56286-2014 Lokomotivy manevrovyye, rabotayushchiya na szhizhennom priродnom gaze Obshchie tehnicheskie trebovaniya (s Popravkoy) TK 45 «Zheleznodorozhnyy transport» – 2015g
[6] Ilyukhin A V, et al 2019 IOP Conference Series: Materials Science and Engineering 643 012102. doi:10.1088/1757-899x/643/1/012102
[7] Pershakov V, Bieliatynskiy A, and Akmaidinova O 2020 Advances in Intelligent Systems and Computing 94–103 doi:10.1007/978-3-030-57450-5_9
[8] Yulin H, Bieliatynskiy A, Sytnichenko N, and Borodina N 2019 Journal of Environmental Management & Tourism 10(7 (39)) 1532-1538.
[9] Barabanshchikov Y, Belkina T, Muratova A, and Bieliatynskyi A 2016 Materials Science Forum 871 9–15 doi:10.4028/www.scientific.net/msf.871.9
[10] Beliatynskij A, Prentkovskis O, and Krivenko J 2010 Transport 25(4) 394–402 doi:10.3846/transport.2010.49
[11] Alpatov V 2019 IOP Conference Series: Materials Science and Engineering 661 012021 doi:10.1088/1757-899x/661/1/012021
[12] Petrov S, Alpatov V 2017 MATEC Web of Conferences 106 04002 doi:10.1051/matecconf/201710604002
[13] Kozlov G, et al 2020 E3S Web of Conferences 175 12015 do:10.1051/e3sconf/202017512015
[14] Abiev R S, et al 2012 CHISA 2012 - 20th International Congress of Chemical and Process Engineering and PRES 2012 - 15th Conference PRES
[15] Kozlov G, et al 2020 Advances in Intelligent Systems and Computing 676–684 doi:10.1007/978-3-030-57453-6_64