The results of pre-design studies on the development of a new design of gas turbine compressor package of GPA-C-16 type

A V Smirnov¹, V M Chobenko², O M Shcherbakov¹, S M Ushakov¹, V P Parafiyynyk¹, R M Sereda¹

¹ Special Design Bureau, Sumy NPO, 58, Gorkogo Str., Sumy, 40004, Sumy, Ukraine
² Gas Turbine Research and Production Complex Zorya-Mashproekt, 42a, Zhovtnevy str., Mykolaiv, 54018, Ukraine

E-mail: ang8l@yandex.ru

Abstract. The article summarizes the results of analysis of data concerning the operation of turbocompressor packages at compressor stations for the natural gas transmission system of Ukraine. The basic requirements for gas turbine compressor packages used for modernization and reconstruction of compressor stations are considered. Using a 16 MW gas turbine package GPA-C-16S/76-1,44M1 as an example, the results of pre-design studies and some technical solutions that improve the energy efficiency of gas turbine compressor packages and their reliability, as well as its environmental performance are given. In particular, the article deals with the matching of performance characteristics of a centrifugal compressor (hereinafter compressor) and gas turbine drive to reduce fuel gas consumption; as well as application of energy efficient technologies, in particular, exhaust gas heat recovery units and gas-oil heat exchangers in turbocompressor packages oil system; as well as reducing emissions of carbon monoxide into the atmosphere using a catalytic exhaust system. Described technical solutions can be used for development of other types of gas turbine compressor packages.

1. Introduction
The Ukrainian gas transmission system (GTS) is one of the largest in Europe. Its total length is 38.6 thousand kilometers including 22.2 thousand kilometers of main gas pipelines. The transmission system includes 72 compressor stations (CS) with 702 turbocompressor packages with the total power of 5.44 million kW.

Industrial gas turbines, aero- and marine-derivative gas turbines, electric motor drives and gas reciprocating engines are used as mechanical drives for pipeline compressors at stations. Gas turbine drives are the dominant option; with power share approximately 82 % of the total power of drive units on the GTS [1].

As of today, a lot of gas turbine compressor packages (GTCP) have reached the end of their first design service life. The efficiency of the prior-art-type drive is 23 to 25 % [1]. Further operation of such gas turbines leads to unreasonable operating costs. In addition, compressors of turbocompressor packages installed at many CS require replacement and modernization due to changes in the operation conditions of gas transmission pipelines.

Therefore, modernization and reconstruction of the Ukrainian GTS is a pressing issue. We should proceed from the fact that, in the short term future, gas turbines will remain the main and most
The most important requirements for GTCP are independence of operation, reliability, efficiency, high level of automation, on-site maintainability, acceptable life-cycle cost, and compliance with current environmental requirements.

Sumy NPO has more than 40 years of experience in design and manufacture of equipment for natural gas transmission CS. GTCPs manufactured by Sumy NPO are widely used in oil and gas industry of Ukraine, Russia, Iran, Uzbekistan and other countries. Today, the Company has designed, manufactured and delivered over 2,350 turbocompressor packages of more than 100 modifications ranging from 4 to 25 MW, designed on the basis of 19 modifications of derivative gas turbines of different types, and 2 types of electric motor drives. Depending on the desired increase in pressure in the packages single-body or multi-body compressors can be used. Fluid or electromagnetic journal and thrust bearings, oil shaft-end or dry gas seals can be used. GTCP equipment has the following installation options: modular type, sandwich panel steel construction, and hard wall compressor shop building.

The article gives the results of GTCP pre-design studies for reconstruction of one of the pipeline CS of the Ukrainian GTS using GPA-C-16S/76-1,44M1 of modular type design as an example. Some technical solutions, which allow improving GTCP efficiency and reliability, and ensuring improved environmental performance by reducing carbon monoxide emissions into the environment, were considered.

2. General characteristics of GPA-C-16S/76-1,44M1
Gas turbine compressor package GPA-C-16S/76-1,44M1 (Figure 1) has the following technical characteristics at the design operating conditions: capacity reduced to a temperature of 293 K and pressure of 0.101 MPa V = 29.5 MCM/PD; discharge pressure (absolute) \( p_d = 7.45 \) MPa; pressure ratio \( \pi = 1.44 \); polytropic efficiency of a compressor \( \eta_p = 87.1 \% \); compressor rotor speed \( n = 5,100 \) rpm; absorbed power \( N = 14.4 \) MW.

Figure 1. General view of GPA-C-16S/76-1,44M1 of modular type: 1 – power unit; 2 – compressor unit; 3 – turbine air inlet system; 4 – turbine exhaust system; 5 – gas turbine oil cooler unit; 6 – supply systems unit; 7 – power unit cooling system; 8 – compressor oil cooler unit; 9 – automation unit; 10 – fire suppression unit.

GTCP main and auxiliary equipment is supplied to the CS as complete and ready-to-operate modules. On-site installation procedure includes their installation on the foundation, interconnection and connection to external communications.

The main GTCP module is a turbine unit with a compressor and gas turbine equipped with auxiliary system elements installed in separate compartments (Figure 2).
Figure 2. Turbine unit of GPA-C-16S/76-1,44M1 of modular type: 1 – frame; 2 – acoustic enclosure; 3 – gas turbine; 4 – ventilation unit; 5 – inlet ventilation silencer; 6 – turbine air inlet duct; 7 – exhaust collector; 8 – protective screen; 9 – compressor; 10 – coupling guard; 11 – coupling; 12 – exhaust duct transition; 13 – gas turbine oil system elements; 14 – fuel gas pipeline; 15 – turbine air inlet tube; 16 – exhaust ventilation duct; 17 – ventilation air distribution system.

The compressor is the main functional system that determines operating conditions of GTCP at the CS. The given GTCP uses a 3-stage barrel compressor with dry gas seals and tilting pad bearings (Figure 3).

Figure 3. General view of centrifugal compressor: 1 – outer casing; 2 – front cover; 3 – rear cover; 4 – journal bearing; 5 – journal-thrust bearing; 6 – dry gas seal; 7 – inner casing; 8 – rotor; 9, 10 – enclosure; 11 – pipeline; 12 – oil pump.

The drive is designed on the base of 16,7 MW marine-derivative engine UGT 16000 with efficiency $\eta_e=35\%$ (ISO 2314) manufactured by Gas Turbine Research and Production Complex Zorya-Mashproekt (Mykolaiv, Ukraine). It should be noted that in this article gas turbine drive (GTD) means a drive unit consisting of a gas turbine supplied by gas turbine manufacturers and auxiliary systems (e.g. turbine air inlet and exhaust system, lubricating oil and fuel supply, etc.) designed and manufactured by Sumy NPO, which provide its effective and reliable operation as a part of the GTCP.

To reduce noise and protect the station staff and equipment against radiant heat, which occurs during gas turbine operation, the engine together with its exhaust collector, lubrication system
elements, fuel gas pipelines, instrumentation and wires and cables is installed in the special acoustic enclosure that is a part of the power unit. To ensure acceptable operating temperature conditions for the engine and auxiliary equipment of the power unit, a mechanical ventilation system is used. The design feature of power unit designed by Sumy NPO is the special ventilation air distribution system inside the acoustic enclosure [3].

The ventilation system has been designed on the base of numerical simulation results [3, 4] and test-based data obtained in collaboration with the experts of the N. E. Zhukovsky National Aerospace University Kharkiv Aviation Institute during field tests of power unit thermal conditions at the CS [5].

3. Matching of characteristics of a compressor and a gas turbine

The most important issues when designing highly efficient GTCP are selection of the compressor flow path and matching of its characteristics with gas turbine characteristics when operating as a one unit.

Matching of characteristics is based on the formation and analysis of the GTCP system or integral characteristic [6-9]. In this case, matching of compressor design (optimum) mode and GTCP optimum mode should be reached. At the same time, the compressor optimum conditions mean those operating conditions of the drive unit, at which the maximum exergy or integral efficiency value of the package, determined at the stage of thermodynamic analysis of GTCP scheme [6, 7], can be reached. This mode corresponds to the minimum flow rate of fuel gas in the gas turbine.

Taking into account the nature of different forms of energy conversion during GTCP operation, as a criterion of its effective operation, it is more correct to use GTCP exergy efficiency in the form of:

$$\eta_{ex}^{GTD} = \eta_{ex}^{CC} \cdot \eta_{ex}^{GTD} \quad (1)$$

where $\eta_{ex}^{CC}$ and $\eta_{ex}^{GTD}$ – compressor exergy efficiency and GTD exergy efficiency, respectively.

However, its use in the initial design stage poses a number of difficulties, so when selecting the compressor flow path, it is permissible to use as a criterion for the GTCP efficiency the integral efficiency [8] in the form of:

$$\eta_{\Sigma}^{GTD} = \eta_{e} \cdot \eta_{p} \quad (2)$$

where

$\eta_{e}$ – GTD efficiency;
$\eta_{p}$ – compressor polytropic efficiency.

Fuel consumption ratio, which is calculated based on the basic parameters of the compressor and GTCP, can be used as a criterion of the package fuel efficiency [9]:

$$K_{f}^{GTD} = \frac{G_{f}}{G_{CC} \cdot \psi_{p}} \quad (3)$$

where

$G_{f}$ – fuel gas mass flow rate, kg/s;
$G_{CC}$ – process (compressed) gas mass flow rate, kg/s;
$\psi_{p} = \psi_{p} / (\psi_{p})_{des}$ – compressor relative polytropic head coefficient, determined at the current $\psi_{p}$ and design $(\psi_{p})_{des}$ operation conditions.

Matching of characteristics and operation conditions of the GTD and compressor reduces to the system analysis of GTCP process with the implementation of the following sequence of calculation tasks [6-11]:

- Selection of the flowing path, determination of compressor gas-dynamic and power characteristics based on customer’s technical requirements (Figure 4, $\eta_{p} = f(N_{PT})$, where $N_{PT}$ – power on the shaft of engine power turbine is equal to $N_{CC}$ including mechanical losses in the rotor system).
• Analysis of gas turbine characteristics in the required power and speed ranges and determination of gas turbine effectiveness including external losses (Figure 4, $\eta_f= f(N_{pr})$).
• Determination of GTCP integral ($\eta_{GTCP}^\Sigma = f(N_{pr})$) or system ($\eta_{GTCP}^e = f(N_{pr})$) characteristics.
• Selection of the compressor/turbine drive operating modes to match the compressor operating mode with $(\eta_{ex}^{CC})_{\text{max}}$ and GTCP operating mode with $(\eta_{es}^{GTCP})_{\text{max}}$. This mode corresponds to the minimum fuel gas flow rate.

Matching of characteristics is determined by behavior of the fuel consumption ratio as a function $K_f^{GTCP} = f(N_{pr})$ [9].

Matching of the package rated operating mode and the modes corresponding to $(\eta_{es}^{GTCP})_{\text{max}}$ and $(K_f^{GTCP})_{\text{max}}$ confirms the accuracy in flowing path selection. Achieving the minimum value of $K_f^{GTCP}$ proves the best efficient mode of GTCP operation.

When selecting the compressor flowing path geometry for GPA-C-16S/76-1,44M1, some kind of studies were performed to match compressor/turbine characteristics using a GTCP efficiency $\eta_{GTCP}^\Sigma$. Figure 4 shows that at the GTCP rated operating mode $N_{pr}=14.4 \text{ MW}$ matching of the compressor design operating mode ($\eta_f=0.87$) and GTCP optimum operating mode ($\eta_{GTCP}^\Sigma =0.283$) is provided. It corresponds to the GTCP best efficient operating mode. It is also confirmed by the corresponding minimum value $K_f^{GTCP}$ (Figure 4 (b)).

![Figure 4. Efficiency of compressor, GTD and GTCP (a) and fuel consumption ratio (b) as a function of power at $n_{ct}=5,100$ rpm.](image)

As the study results [8] show, using low- and medium-head impellers ensures optimization of the compressor and turbine drive operating modes. In this case, consumption of UGT 15000 fuel gas can be reduced by 5.4…7.6% (3...3.5 million nm$^3$/year) [8].

4. Improvement of GTCP energy efficiency using heat recovery systems

GTCPs are significant sources of thermal pollution to the environment. Most heat losses are caused by emission of exhaust gases into the atmosphere.

Heat recovery from the GTCP exhaust gases can be used for increasing gas turbine efficiency using regenerative or water (steam) injection cycles, as well as being used to produce heat, electric power or cold for needs of the CS or the external consumers.

Considering the high power consumption of the economy of Ukraine, development of combined power, heat and cold generation facilities on the basis of the CS is a prospective line of GTS.
development. This will reduce the costs of natural gas transmission and can also become the basis for development of effective regional power production and infrastructure of new agricultural and industrial enterprises [12, 13]. It should be noted that for existing simple-cycle gas turbines with the efficiencies of 30 to 40% (for 6.3 to 32 MW power respectively), complete recovery of thermal resources of exhaust gases provides the increase of thermal efficiency of the system up to 0.8 to 0.85.

However, the implementation of this approach is associated with high capital costs (especially in the context of combined cycles), the need to improve reliability of equipment, and also requires the development of the appropriate legislative and regulatory framework for the practical solution of the issues connected with sale of electric power, heat and cold produced by heat recovery facilities operating as a part of the CS [13, 14].

Therefore, currently waste heat of GTCP exhaust gases is mostly used for heating of CS, residential settlements or greenhouse facilities near the station, etc. The disadvantage of this solution is inability to use all turbine exhaust gases thermal resources due to lack of the regular consumers of generated heat at the CS.

To produce utility heat GTCP of GPA-C-16S type are equipped with heat recovery units, of between 3.5 to 9 MW which are designed and manufactured by Sumy NPO on the basis of finned tube heat exchangers (Figure 5). Their application allows achievement of thermal efficiencies of 0.36 to 0.46. The heat recovery units can contain several separate heat exchangers. Heat power control of the heat recovery units is carried out by heat exchangers on-off and by controlling the flow rate of exhaust gases through them. The design of the heat recovery units allows filling heat exchangers with water without shut-down of the gas turbine. To avoid high thermal stresses heat exchangers are cooled with atmospheric air supplied by fan of the heat recovery unit cooling system.

One of the promising ways to improve GTCP efficiency is also the recovery of low-grade heat, which is obtained with oil from the compressor and gas turbine bearings. This heat can be used for fuel gas heating in special gas-oil heat exchangers [15-19]. The use of gas-oil heat exchangers allows reduction of fuel gas consumption and decrease of the electric power consumption to cool the oil. A general view of the gas-oil heat exchanger designed for use in the gas turbine lubrication systems for GPA-C-16S type packages is shown in Figure 6. Its main technical characteristics are given in Table 1.

According to the results of calculations [19], use of the gas-oil heat exchanger in gas turbine lubrication system of GPA-C-16S type package allows reducing the fuel gas consumption by 42,000 m³/year and decreasing the electric power consumption of oil coolers fans by 23,000 kWh/year.
**Figure 6.** General view of the heat recovery gas-oil exchanger:
1 – inlet-outlet chamber; 2 – intermediate chamber; 3 – shell; 4 – bimetal finned tube; 5 – leakage path; 6 – leakage safety chamber.

**Table 1.** Main technical characteristics of the gas-oil heat exchanger.

| Parameter                      | Value   |
|--------------------------------|---------|
| Medium                         | Oil     | Natural gas         |
| Rated flow rate, kg/h          | 8,700   | 3,500               |
| Inlet temperature, °C          | 100     | 20…40               |
| Outlet temperature, °C         | 89.5    | 40…60               |
| Operating pressure at inlet, MPa | 0.44    | 3.0                  |

5. **Reduction of carbon monoxide emissions using the catalytic exhaust system (CES)**

One of the most important objectives of the CS reconstruction and modernization is reduction of emissions of pollutants, primarily nitrogen oxides (NOx) and carbon monoxide (CO) in the atmosphere.

Thanks to the implementation in Ukraine of Directive 2010/75/EU of the European Parliament and of the Council of 24 November 2010 on industrial emissions, the concentration of NOx and CO in GTCP exhaust gases (at loads more than 70% of the gas turbine rated power) should not exceed 75 and 100 mg/nm³ respectively. Currently, Gas Turbine Research and Production Complex Zorya-Mashproekt has designed and tested a low-emission combustion chamber for UGT15000 gas turbine meeting the requirements of the Directive. However, according to the Terms of Reference for packages intended for reconstruction of the CS of Ukrtransgaz, the CO concentration in exhaust gases should not exceed 100 mg/nm³ within the whole range of gas transmission modes. It corresponds to the load range of 45 to 90% of the gas turbine rated power. Ensuring the acceptable environmental performance of the gas turbine in such a wide range of capacities is a complex scientific and technical problem [14, 20, 21].

The main methods for reducing the pollutant emissions with products of fuel combustion in the gas turbines are as follows [14, 20, 21]:

- improving the methods of fuel combustion (so-called "dry" methods DLN, DLE; catalytic combustion chambers);
- injection of water (steam) into the gas turbine combustion chamber;
- using special systems for combustion product purification in the exhaust systems.

The first two methods require a significant change in a gas turbine design. In this regard, to meet the environmental requirements specified by the customer, the package design is provided for use of CO removal catalytic system installed in the exhaust system of GTCP. The principle of CES operation
is based on the oxidation of carbon monoxide to carbon dioxide under the action of the catalyst layer at a certain temperature.

Currently, catalytic purification of combustion products is mainly used in power engineering at thermal power plants, boilers, combined cycle and gas turbine plants. At the same time, they are not widely used at CS because of relatively high capital costs [14].

There is relatively small number of CES international manufacturers for gas turbines, among which are BASF (Germany), Haldor Topsoe (Denmark), ECAT (RF).

Traditionally the CES consists of units that are installed normal to the direction of exhaust gases flow. The units are made on the basis of highly porous ceramic or metal honeycomb structures coated with catalysts.

Use of CES in GTCP exhaust systems is associated with some problems. To ensure effective chemical oxidation of carbon monoxide, it is required to meet the following conditions: the temperature of the exhaust gases should be within the certain limits (200 to 450°C depending on the type of the used catalyst); the CES inlet flow should be uniform.

As was mentioned above, to recover waste heat of gas turbine exhaust gasses, the exhaust system of GPA-C-16S/76-1,44M1 package is equipped with a heat recovery unit. Therefore, to ensure the required exhaust gas temperature at the CES inlet the catalyst should be installed before the heat recovery unit. To select the CES location, a series of numerical simulations have been performed in the exhaust system of the existing design (Figure 7(a)). Figure 7(b) shows the streamlines in the existing exhaust system of GPA-C-16S/76-1,44M1 package at the turbine rated operation conditions.

![Figure 7(a) and streamlines (b)](image)

**Figure 7.** Structural scheme (a) and streamlines (b) in the existing exhaust system of GPA-C-16S type package: 1 – exhaust collector; 2 – exhaust duct transition; 3 – expansion joint; 4 – the 1st stage diffuser; 5 – the 2nd stage diffuser; 6 – heat recovery unit; 7 – transition piece; 8 – silencer; 9 – confuser; 10 – exhaust stack; 11 – hood.

Figure 7b shows that flow in the exhaust system has a complex structure with multiple vortex zones and reverse flows. The total pressure loss in the exhaust system without catalyst is 2,550 Pa. The maximum deviation of velocity from the mass flow averaged value in the proposed cross-section of CES (between the diffusers of the first and second stages) is 400 %, RMS is 128 %. Such non-uniform flow structure in the
exhaust system is associated primarily with features of the flow at the exhaust collector outlet. The flow at the exhaust collector outlet has symmetrical swirling, which occurs as a result of stream rotation flowing on the plenum of the exhaust collector after coming out the axial-radial diffuser. This swirl remains until the outlet section of the exhaust collector and extends to the silencer.

According to the CES designers at such cross-sectional area of the catalyst and the degree of non-uniformity of the inlet flow, the total pressure loss in CES at the rated operating mode of the turbine exceeds 5,000 Pa, which is unacceptable. To ensure the guaranteed parameters of the gas turbine operation, the total pressure loss in the exhaust system from the exhaust collector inlet to the outlet to the atmosphere should not exceed 4,905 Pa.

To achieve the goal, a series of numerical simulations have been done to develop a new design of the exhaust system using various types of flow straighteners. According to the results of the study, the exhaust duct design with tubular straightener (Figure 8(a)) has been selected.

According to the results of calculations, the total pressure loss in the exhaust system of the new design is 4,630 Pa, maximum velocity deviation from the mass flow averaged value at the CES inlet is 65 % and RMS deviation is 18 %.

6. Conclusion
Taking into account a large power of GTCPs used at CSs of the Ukrainian main gas pipelines and strategic importance of the GTS for Ukraine and other European countries, further improvement of GTCPs to increase their energy efficiency, reliability and improve environmental performance is of great importance. Tasks in the improvement process are different and knowledge-intensive. The main ones are as follows:

- Improvement of gas turbines based on the design principles of industrial gas turbines and achievements in thermo-gas dynamics, materials technology, dynamics and strength of components and systems used in aircraft and marine engines building.
• Development of new gas turbine designs based on regenerative and combined operation cycles.
• Improvement of design methods for compressor flow paths and other energy-engineering equipment.
• Application of heat recovery technologies; promising direction of further improvement of GTCPs and CSs to increase their energy efficiency is designing heat recovery facilities for power, heat and cold generation on the base of CSs.
• Improvement of auxiliary GTCP systems to provide efficient, reliable and safe operation of the package. Such systems include turbine air inlet and exhaust systems, turbine unit systems, gas turbine and compressor lubrication systems, etc.
• Improvement of GTCP environmental performance, in particular, reduction in pollutant emissions by improvement of gas turbine combustion chambers, as well as due to use of CES.

References
[1] Paton B, Khalatov A, Kostenko D, Bileka B, Pismenny O, Botsula A, Parafiynyk V and Konyakhin V 2008 Bulletin of NAS of Ukraine 4 pp 3-9
[2] Pirani S 2007 Ukraine's gas sector (Oxford: Oxford Institute for Energy Studies) p 115
[3] Smirnov A, Shcherbakov O, Tkachenko D, Parafiynyk V and Slabko Yu 2016 Aerospace technic and technology 7 pp 179-187
[4] Smirnov A, Kostyuk V, Tkachenko D, Kyrylash E and Slabko Yu 2013 Herald of aeroenginebuilding 2 pp 99-107
[5] Shcherbakov O, Tkachenko D, Parafiynyk V, Gurinenko V, Kostyuk V, Skrypka O and Kyrylash O 2015 Eastern-European Journal of Enterprise Technologies 6/7 (78) pp 35-42
[6] Tertyshny I, Prylypko S, Miroshnichenko E and Parafiynyk V 2015 Journal of mechanical engineering 18 (4/1) pp 9-17
[7] Tertyshny I, Prylypko S and Parafiynyk V 2016 Journal of mechanical engineering 19 (2) pp 10-18
[8] Parafiynyk V, Nefedov A, Yevdokymov and V, Tertyshny 2012 Compressors and Pneumatics 2 pp 10-17
[9] Tertyshny I, Parafiynyk V, Nefedov A, Rogalsky S, Kotenko N, Tymoshadchenko D and Mikhaylenko S 2014 Proc. of XVI Int. scientific-technical Conf. on Compressor Manufacturing vol 1 (Saint Petersburg, REP-Holding) pp 328-339
[10] Taher M and Meher-Homji C 2012 Proc. of ASME Turbo Expo 2012: Turbine Technical Conference and Exposition (Copenhagen) pp 51-61
[11] Morosuk T and Tsatsaronis G 2008 Energy 33 pp 890-907
[12] Bileka B 2006 Industrial heat engineering 28 (2) pp 132-149
[13] Smirnov A, Parafiynyk V, Shcherbakov O, Epifanov S, Kostiuk V, Chobenko V and Shevchuk V 2016 Bulletin of National Technical University "KhPI": coll. of sci. papers. Ser: Power and Heat Engineering Processes and Equipment 9 (1181) pp 13-25
[14] Galiullin Z, Salnikov S and Shchurovsky V 2014 Modern Gas Transmission Systems and Technologies ed Shchurovsky V (Moscow: Gazprom VNIIGAZ) p 346
[15] Bodunov D. 2013 Gas Turbo Technology 6 pp 18-19
[16] Yokell S 1973 Chemical Engineering 14 pp 133–136
[17] Kuznetsov L, Kuznetsov Yu, Efremov A, Burakov A and Abramov A 2009 Fuel gas treatment unit Patent RU 92934
[18] Belousov Yu, Vereshchagin N, Pchelintsev V and Golubtsov S 2012 Tubular bundle with buffer chambers of gaseous oil shell-and-tube heat exchanger Patent RU117599
[19] Smirnov A, Sereda R and Borisov M 2016 Bulletin of National Technical University "KhPI": coll. of sci. papers. Ser. : Power and Heat Engineering Processes and Equipment 10 (1182) pp 93-98
[20] Shchurovsky V, Sinisyns Yu and Cheremin A 2004 Reduction in Pollutant Emissions of Gas Turbines at GASPRM Compressor Stations: Report (Moscow)
[21] Varlamov G, Khalatov A, Poznyakov P and Yurashev D 2012 Eastern-European Journal of Enterprise Technologies 10 (57) pp 9-14