The development of a high efficiency microturbine generator for a vehicle

R Iakunin\(^1\), A Kostyukov\(^1\), A Kozlov\(^1\), G Nadareyshvili\(^1\) and A Terenchenko\(^1\)

\(^1\) NAMI Russian State Scientific Research Center, Moscow, Russia
E-mail: r.yakunin@nami.ru

Abstract. Currently, microturbines are becoming increasingly common in power generation. There is a desire to use them as an energy source for the hybrid cars with the wheels driven by an electric motor. However, stationary microturbines are not well adapted for use on the automobiles. The main differences are frequent engine stops and lack of space for large heat exchangers. These differences lead to a decrease in the engine efficiency. The article discusses the technical solutions for the design of microturbines with high efficiency, small size of the heat exchanger and the bearings with reduced friction losses. The calculation studies have shown a possibility of achieving high efficiency for microturbines in a car. Currently, tests of real engine components to confirm the calculations have ended. Based on the test data, a microturbine is being designed. This microturbine will have been made by the end of this year.

Nomenclature

Latin alphabet
- T temperature, K
- N power, kW

Greek alphabet
- \(\sigma\) efficiency of the heat exchange
- \(\eta\) efficiency
- \(\pi\) pressure ratio

Abbreviations
- CNG Compressed Natural Gas
- ICE internal combustion engine

1. Introduction
Currently, microturbines are becoming increasingly common in power generation. A promising trend is the application of a microturbine as a component of an electric hybrid vehicle \([1, 2, 3, 4]\).
In this work, the development of a prototype model of a 40-60 kW multi-purpose gas turbine engine for power plants of various purposes, including those for vehicles, is shown.
A Gas turbine engine have several problems, which are especially important on vehicles:
- engine cost;
- high fuel consumption (low efficiency), especially at partial load;
- large volume of hot exhaust gases to be removed;
- noise level.
The solution of these problems has been made possible by using a heat exchanger (regenerator). The heat exchanger uses the lost energy of the exhaust gases to preheat the air entering the combustion chamber.
It solves these problems as follows:
the cost of the engine is reduced, because to ensure the high efficiency, you can use only one radial compressor and turbine stage instead of several axial stages with a high degree of the pressure increase;
- the fuel consumption is reduced, especially at partial loads;
- the exhaust gases are not so hot;
- a side effect of the heat exchanger is noise damping.
A disadvantage of a heat exchanger – an increase in the engine size.
Several vehicles obtaining serial microturbines as range extenders are shown in Figure 1.

![Figure 1. Vehicles with serial microturbines. On the top: Kenworth hybrid electric truck with Capstone C65 microturbine; Mack hybrid electric truck with Wrightspeed microturbine. Below: Russian hybrid electric bus "Trolza-5250 Ecobus" with Capstone C65 microturbine.](image)

In Russia, in 2010, a pilot batch of environmentally friendly Trolza-5250 “Ecobus” buses, equipped with Capstone C65 microturbines, was manufactured. In 2011, these buses were successfully tested in Moscow and Sochi, and are currently operated in the Krasnodar region. However, serial microturbines have poorly adapted bearings for frequent starts, and their heat exchangers take up a lot of space. Thus, it has become necessary to develop a microturbine, which can be successfully used both in transport and in a stationary application.
The engine being under development is multi-fueled and of high efficiency. It can also use CNG, which allows low environmental pollution. A car with such a power plant meets the environmental requirements of the state of California (the toughest in the world) without the use of a catalytic converter, diesel particulate filter and the use of AdBlue [5].
2. Calculations

The engine must be developed with a high efficiency. Currently, it is not possible to significantly improve the efficiency of the compressor and turbine. Given the efficiency of the compressor and the turbine, the effective specific fuel consumption improves when the temperature at the turbine inlet rises. We can achieve this by increasing the degree of pressure increase, but this requires the use of expensive multistage axial compressors and turbines. At small sizes, their efficiency considerably worsens. Therefore, for microturbines it is advisable to use a heat exchanger with a radial compressor and a turbine.

The heat exchanger (regenerator) provides the Brighton cycle with regeneration. The temperature at the turbine outlet (\(T_{out}\)) is higher than that of the air leaving the compressor. Ideally, with the infinite heat transfer area, the air from the compressor can be heated to the temperature \(T_{out}\) before it is supplied to the combustion chamber. This allows burning less fuel to heat up to the required temperature. In a real heat exchanger, the heat cannot be transferred completely; moreover, some of the energy is required for the air passage through the heat exchanger. Therefore, the heat exchanger must be carefully designed. A poorly designed heat exchanger may not only be useless, but also decrease the efficiency.

The effectiveness of the heat exchanger may be expressed as

\[
\sigma = \frac{T_{cc} - T_{com}}{T_{out} - T_{com}}
\]

Where:

- \(T_{out}\) – the temperature at the turbine outlet;
- \(T_{com}\) – the temperature after the compressor;
- \(T_{cc}\) – the temperature before the combustion chamber.

The efficiency of the heat exchangers in the modern microturbines is within the range of 80-86%. These indicators are essentially the limits for the fixed plate heat exchangers. Attempts to increase their efficiency above these limits lead to an unacceptable increase in their size. Nevertheless, a further increase in the efficiency of the heat exchanger is highly desirable. Increasing the degree of regeneration of the heat exchanger from 86% to 95% provides an increase in the efficiency of the microturbine 50 kW from 32% to 38% (Figure 2).

![Figure 2](image_url). Effect of recuperation degree on the 50 kW microturbine efficiency (gas temperature after the combustion chamber 12230K)
High regeneration rate is achievable in a rotary regenerator. For example, in a ceramic rotary heat exchanger, a regeneration rate of 95–97% was achieved [6]. By using a less exotic material, it is also possible to achieve the regeneration rate up to 95%, which provides competitiveness of microturbines in terms of the fuel efficiency comparable with a reciprocating ICE [7]. The developed microturbine with a power of 50 kW should have an efficiency of 37.8%. According to the results of the calculations, to ensure the obtained calculated efficiency of the microturbine, parts of the microturbine must meet the following requirements:

The compressor
- air flow at the compressor inlet – 0.41 kg/s
- pressure increase degree in the compressor – 3.5.
- adiabatic efficiency – not less than 0.8;

The turbine
- gas temperature before the turbine – 1223K; modern alloys used in gas turbine technologies can operate at temperatures up to 1223 K without cooling and at the same time provide a resource of up to 60 thousand hours.
- adiabatic efficiency – not less than 0.86
- gas temperature at turbine outlet – 967.5 K
- gas pressure at the turbine inlet – 327200 Pa;
- gas flow through the turbine – 0.40 kg / s

The heat exchanger
- recovery rate σ – not less than 0.95;
- total pressure loss from the compressor outlet to the combustion chamber inlet must not exceed 2%;
- total pressure loss from the turbine outlet to the gas exhaust device inlet must not exceed 8%;
- gas temperature at the regenerator inlet – 967.5 K;
- air temperature at the regenerator inlet – 442.9 K;
- regenerator inlet air flow - 0.41 kg / s
- regenerator inlet air pressure - 347800 Pa

The combustion chamber
- the combustion chamber must ensure the fuel combustion with minimal formation of harmful substances (no more than 10 ppm);
- air flow rate at the combustion chamber inlet – 0.41 kg / s;
- air temperature at the combustion chamber inlet – 941.3 K;
- gas temperature at the combustion chamber outlet – 1223K;
- air pressure at the combustion chamber inlet – 340800 Pa;
- total pressure losses from the combustion chamber inlet to the turbine inlet must not exceed 2%.

Other
- air loss from the compressor to the turbine is not more than 3%;
- total pressure loss in the flow part of the microturbine is not more than 15%;

3. Design development
The rotary heat exchanger
The main problem of the rotary regenerators is their seals. The disc frame regenerators have minimum leakage in seals. In them, the seals do not work on a porous heat transfer matrix, but on flat surfaces. To reduce the frame thermal deformations and, accordingly, to reduce leaks, as well as to enable using
of graphite seals (maximum temperature 450-4700°C) the frame is cooled. The cooling system of the frame has been patented. The most promising heat transfer element, as shown by a computational analysis, are slotted elements (Figure 3). They have been successfully tested.

![Figure 3. The frame of the heat exchanger and the cylindrical slotted heat transfer element.](image)

The combustion chamber. For a microturbine with a rotary disc heat exchanger, a tube-type combustion chamber is preferable for the layout reasons. In addition, the tube-type combustion chamber has only one nozzle, and, therefore, has no problem with uneven fuel supply in the different injectors. To reduce emissions, computational studies have been carried out. Based on their results, a combustion chamber with calculated NOx emissions [ppm] 10 has been developed.

The turbine.
A computational analysis of the diffuser efficiency after the turbine has shown a significant advantage (in terms of the microturbine efficiency) of the elongated diffuser. The overall layout of the microturbine with an elongated diffuser and a rotating heat exchanger is shown in Figure 4.

![Figure 4. The overall layout.](image)

This layout has the following advantages:
- low length of the heat exchanger seal;
- design simplicity;
- efficiency of the diffuser;
- modularity design.
Bearings.
The most problematic issue for a turbocharger is the high-temperature heat flow to the bearings, especially after the microturbine has stopped. The removal of this heat is very energy-intensive and complicates the operation of the microturbine power plant. To isolate the bearings from the turbine, the layout with a console mounting of the turbine wheel has been adopted.
It was decided to move away from the air bearings, despite their low friction losses. During the launch of the microturbine, the rotor of the turbocharger rests on elastic petal elements, which leads to a significant reduction in the life of the bearings. The developed turbine is to be used in road transport and, therefore, it will often be launched during operation. In addition, when operating in vehicles, shock loads may occur. Obviously, in these cases, the use of air bearings is hardly advisable.
Currently, liquid bearings are also used in microturbines. The rotor in these microturbines is mounted on one hybrid ball bearing and on one radial liquid bearing. The installation of the oil system does not lead to any significant increase in the cost of the microturbine, because the oil system is useful in any case. In our layout, bearing has been moved to the cold zone, and the oil is not coxed during operation.
The disadvantage of the fluid bearings is large friction losses. Friction in fluid bearings depends on the surface area of the bearings. At high rotary speed, which is a characteristic of the gas turbine engines, it would be a quite small bearing area. However, before achieving the required speed of rotary in the bearing, there is a mode of boundary friction. In this mode, we must provide a small pressure on the bearing material to avoid material wear. For this reason, the area of fluid bearings in turbo compressors is excessively large, which means that the friction losses are large. To reduce wear during boundary friction, the bearing is made of a silicon carbide-based composite. This has made it possible to make the area of the bearing as minimum as necessary to create an oil wedge at operating speed.
For this reason, the friction losses in the liquid bearing at the operating speed have been significantly reduced. The exact value of the possible reduction in bearing area will be obtained from the test results on the bench.

4. Conclusion
1) The selection and justification of the main target parameters of the microturbine has been carried out:
   - fuel consumption – 10.95 kg / hour;
   - microturbine power – 50 kW;
   - microturbine efficiency – 37.8%;
   - degree of pressure increase – 3.5;
   - heat exchanger efficiency 95%.
2) The selection and justification of the optimal layout of the microturbine has been carried out:
   - liquid composite bearings;
   - with a rotary regenerator;
   - with a tube-type low-toxic combustion chamber;
   - with an extended diffuser.
3) 3D microturbine model has been developed
4) Mathematical modeling has been carried out:
   - flows in the gas-air pipes of the heat exchanger with an ultra-high degree of regeneration;
   - gas-air flows in the pipe connecting the combustion chamber and the microturbine,
   - gas-air flows in the diffuser;
   - thermal state of the microturbine composite bearings;
   - thermal stress state of the microturbine body elements;
   - gas-air flows in the compressor and the turbine.
   The optimization calculations of the efficiency and service life of the compressor and turbine for the heat-stressed state: the efficiency values have been obtained for the full parameters 0.801 and 0.901 for the compressor turbine stages. The impeller meets the requirements of the static strength (safety factor $n > 2.5$), and durability when operating at the nominal conditions for 40,000 hours.
5) Individual engine parts have been manufactured and tested:
   - bearings;
- heat transfer element.

5. References

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