Employment of a centrifugal turbine for the utilization of the natural gas energy within the process of pressure reduction

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Abstract. There is no doubt that turboexpander generation technology has considerable benefits such as reduction of CO2 emission and maintaining autonomous power supply for gas letdown stations and some technological processes. In spite of the abovementioned advantages of this technology its further development faces challenges due to: 1. long payback period of plants with conventional turbines; 2. significant annual fluctuations of gas inlet parameters; 3. high demands of conventional turbines for working fluid cleanliness. In current study the bladeless centrifugal reaction turbine was considered in order to struggle with abovementioned constraints. The investigated turbine is a variation of so-called Segner wheel known for centuries. Its efficiency is lower comparing with the conventional rotary machines, but remains beneficial within the specific conditions of turboexpander application. A comparison of the proposed turbine and axial transonic one was provided within a wide range of pressure ratios as typical operating conditions. The advantages of the investigated centrifugal turbine such as ability to generate comparable power as conventional ones while having lower cost and higher mud and erosion resistance has been highlighted.

1. Introduction

One of the possible applications of turboexpanders is electrical generation with using pressurized natural gas at gas letdown stations. In this case the pressure reduction occurs in a turbine with some energy output while routine procedure of throttling is not able to produce work. A potential of electricity generation via turboexpanders is immense all over the world. An interesting fact is that both gas exporting and importing countries may have significant generation potential. Rasmussen [1] in the framework of ZEGEX+ project showed that the net annual electricity production of 80 GWh is possible in Danish gas distribution system. Lehman et al. [2] stated that the theoretical maximum of annual energy generation by expansion turbines in USA gas transport system is 21TWh, which is an equivalent of about 2500 MW power. Cleveland et al. [3] referred to the previous investigations which had assessed the British gas distribution system power capability as 390 MW excluding small letdown stations with less than 100 kW available output. Korean scientists [4] evaluated the generation potential in their country to be in the range 100-200 MW taking into account seasonal oscillations of gas consumption. As reported by Environics advisory company in cooperation with dr. Mohamed Salah Elsobki [5], Egyptian gas distribution system may provide up to 90 MW electricity via turboexpander generation. In Pakistan Unar et al. [6] assessed the maximal installed power as 6.32 MW.

Currently, despite the high potential of energy recovery just a limited number of turboexpander generation plants exist. Analysis prepared by Energy and Environmental Analysis, Inc. [7] established
that by 2008 there were not known commercial turboexpander installations generating electricity at city gates in the U.S. pipeline system. Several plants have been established in Netherlands and Japan in 1990-s as mentioned by Lehman et al. [2], but they have power range over 1 MW. Up to 10 plants were installed in Russia and CIS over the last 20 years within the same power range. The overall world generation potential of gas transport systems is shown in the figure 1. The three main reasons that inhibit the further development of turboexpander market can be outlined:

1. High plant capital cost and therefore extended payback period;
2. Significant annual fluctuations of gas inlet parameters, which cause low efficiency and reduced power output;
3. High demands of conventional turbines for working fluid cleanliness.

Figure 1. The world generation potential of gas transport systems.

2. A centrifugal reaction turbine

2.1. Turbine design features
In contrast to conventional turbines which comprise several nozzle vanes and one impeller, the centrifugal reaction turbine has an impeller with several passages placed in a hood as shown in figure 2. A working fluid enters the impeller at the center, turns to the radial direction getting into the passages and moves radially outwards. The passage is shaped so as to meet the working fluid compression due to centrifugal forces. Each passage terminates in the nozzle where the flow accelerates reaching high Mach numbers since the overall gas potential is captured by impeller. Due to transonic operation convergent-divergent nozzle has to be applied. In order to simplify the design the divergent part is skipped in favor of fluid expansion in the axial gap.

The baseline turbine designed so as to provide 32 kW of mechanical power at 3000 rpm and the pressure ratio of 3.167. The design parameters are provided in table 1.
Table 1. Design parameters of the baseline turbine

| Parameter   | Dimensions | Value  |
|-------------|------------|--------|
| Throat shape| -          | Rectangular |
| \(D_m\)    | mm         | 398.5  |
| \(\varepsilon\) | -       | 0.42   |
| \(a_{th}\) | mm         | 10     |
| \(l_{th}\) | mm         | 12.1   |
| \(\beta^{*}_2\) | deg.   | 5      |
| \(Z\)      | -          | 4      |

Figure 2. The centrifugal turbine outline.

2.2. Mechanical design, manufacturability and erosion resistance

The impeller consists of two parts. A primary disk contains channels and nozzles, which are milled out on it, and an inlet spinner cone. A cover disk finally shapes the flow part and also bears a radial labyrinth seal.

Passages are straightforward to manufacture due to a rectangular-shaped cross section. Outlet nozzles require 5-axis milling. Due to small turbine outlet angles the number of passages is several times smaller than in conventional turbines within similar mass flow rates. The abovementioned factors and the absence of stator vanes define a centrifugal turbine’s manufacturability and, therefore, a lower production cost.

The exhaust hood is designed to be straightforward to manufacture. It consists of a hollow cylindrical bottom part, where the impeller is located, and a prismatic top part (collector) with a simple flow separator.

Since the impeller’s flow path is arranged with channels instead of blade passages its erosion resistance is obviously higher. Rotating passages in the same time prevent mud that natural gas contains from deposition in the flow path.

3. A centrifugal reaction turbine

As reported in the introduction pressure and gas flow variation is typical for a gas letdown station operation. Therefore ability of an expander unit to provide efficient and stable operation within a wide range of inlet conditions is of particular importance. This chapter aimed at comparison of a centrifugal reaction turbine and axial impulse turbine performance within a wide range of pressure ratios. A centrifugal turbine preliminary design has been implemented via 1D in-house tool described above, as well as the turbine’s performance lines. An axial turbine from the doctoral thesis of Sebelev [8] is considered. 1D meanline model described in the said thesis is used for the axial turbine simulation. Both turbines have 28 kW of output power at the design point of \(p_2 = 1\) atm, \(\pi = 5.0\), \(T_0^* = 343\)K and are driven by air. A pressure ratio \(1.5 < \pi < 8.0\) is considered. An optimal \(u/C_0\) value (the value that provides the maximal total-to-static efficiency) is chosen for each pressure ratio. Charts for total-to-static efficiency, normalized mass flow rate (a ratio between a current mass flow rate and the mass flow rate through the axial turbine at the design point) and normalized power (a power at different pressure ratios divided by the nominal power) are given in figure 3.
Figure 3. Comparison of centrifugal bladeless and axial impulse turbine (1D model results).

Total-to-static efficiency curve illustrates typical axial turbine behaviour: efficiency sees a decrease while the pressure ratio $\pi$ varies from the design point, whereas the pressure ratios below the design one are evidently more adverse. The latter was also shown by Natalevich [9]. On the contrary, centrifugal turbine exhibits flat efficiency curve within the whole range of pressure ratios. In fact, higher values could be obtained if rotational velocity limit due to strength safety were not imposed.

In spite of significantly different efficiency both turbines provide comparable power within the range under consideration. This is owing to an increasing mass flow rate of the centrifugal turbine affected by its rotational velocity as described above. Indeed, centrifugal turbine requires higher mass flow rates to generate the same power due to a lower efficiency. In periods of peak power consumption it is supplemented by conventional throttling values. Energy within the excessive flow is therefore lost in case of conventional turbine, but is captured by centrifugal turbine resulting in almost the same power output as shown in figure 3. In general, normalized mass flow rate plots show that a centrifugal turbine is able to operate within a wider range of mass flow rates than a conventional axial turbine.

It should be noted that due to its expected lower cost a set of two centrifugal turbines installed in parallel may be introduced to avoid deep part load operation. In case of conventional turbine units such measures are usually not economically viable. In general, more extensive economical evaluation is required to obtain secure predictions of the proposed turbine applicability. On this account technical-economic model is a subject of further studies.

4. Conclusions
A brief overview on turboexpander generation potential worldwide has been highlighted in current paper. As was shown the turboexpander power generation is a promising source of electrical energy production in the future. Nevertheless, the development of gas turboexpander market is inhibited by the several significant restrictions.

A comparison study of a centrifugal turbine and transonic axial turbine was carried out for a wide range of non-design points. The following major outcomes should be noted:

1. The ability of a centrifugal turbine to process higher range of mass flows is in line with gas letdown stations application specificities.
2. A centrifugal turbine provides more stable efficiency within a wide range of pressure ratios while an axial one has a pronounced peak. In spite of axial turbine higher efficiency, both machines exhibit close power output as a result of the abovementioned performance particularities.
3. The cost of a centrifugal bladeless turbine is expected to be lower than of conventional ones due to technological simplicity, absence of blades and a nozzle block. Bladeless design is also favourable in terms of mud and erosion resistance.
4. Relatively low efficiency of the centrifugal turbine prevents the working fluid temperature from a significant drop within the expansion process. As a result, the amount of the thermal energy for the natural gas preheating is reduced, which is also a positive aspect.
5. The centrifugal turbine has some particularities of the behaviour (non-linear turbine torque, dependency of the mass flow from the rotational speed). The said ones are described in detail by Smirnov [10].

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