EFFICIENCY IMPROVEMENT OF SMALL GAS BEARING TURBINES - IMPACT ON STANDARD HELIUM LIQUEFIER PERFORMANCE

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ABSTRACT

Radial turbine design is dictated by criteria like specific speed and/or velocity ratios. For small capacity plants the size of the turbine wheel needs to be reduced and thus the rotational speed increased in order to reach a high efficiency. The design of a small turbine taking mechanical and manufacturing criteria into account will be presented as well as the impact of reducing size (lower Reynolds number, bigger wheel relative clearance and blockage, etc.) An overview of the technical and operational improvements is presented as well as the first test results. The impact of the new turbine design on the capacity of standard liquefiers is calculated and analyzed. Additional tests on a standard plant with a liquefaction capacity of 65l/h are being performed to confirm the calculated results.

INTRODUCTION - GETTING SMALLER

It is commonly acknowledged that scale down of expansion turbines to very small mass flow rates and thus tiny dimensions will encounter difficulties for many reasons. This is specifically the case, if the turbine is to be designed for a gas with a low molecular weight as helium. The key reasons for this are:

- Both the diameter and the blade height are reduced. Thus a very small geometry is to be machined with very high accuracy and smooth surfaces
- The clearance between wheel, its cover and its inlet geometry have to be decreased in order to keep clearance losses small
- While the turbine wheel diameter decreases, the turbine speed must be increased to maintain the required high tip speed
- With decreasing turbine mass flow, heat leaks along the turbine housing and the rotor have an increasing effect on its efficiency.

While the above arguments are rather discouraging, other features, such as the low maintenance requirements, the exceptional reliability and the low vibration levels of turbines are highly in favour of using them also for small helium plants. Further development work focussing on efficiency seemed therefore to be justified.

DESIGN

Standard process and machine design are based on physic laws and on experimental parameters. A characteristic parameter for a turbine is the non-dimensional mass flow rate defined as in equation (1).

$$\mu = \frac{\dot{m}}{\rho_1 \pi \cdot r_1^2 C_s} \tag{1}$$

with:

$$C_s = \sqrt{2\Delta h_{0s}} \tag{2}$$

Turbine characteristics and efficiency curves for small capacity expander ($\mu < 0.045$) are retrieved from measurement, but no retrofit of the basic turbine design (clearance, thermal losses, etc.) was performed until now. As the market for small liquefiers (30-65l/h) has grown, the corresponding turbine design is reconsidered for improvement. In FIGURE 3 a process flow diagram shows a typical components arrangement for a small liquefier.

Current design

The current design is based on the above described efficiency curves and is named case 0. The process values are written down in TABLE 1.

TABLE 1. Typical process values for a small liquefier

Turbine		Tu 1	Tu 2
Inlet pressure	[bara]	12.6	5.57
Outlet pressure	[bara]	5.57	1.37
Inlet temperature	[K]	70	17
Massflow	[g/s]	27	27

New design

Considering above process cases, the turbine geometry is computed based on new one-dimensional correlations, considering the size reduction effects. The results define the design case 1. Geometry differences are illustrated in FIGURES 1 and 2.

Thermodynamic design criteria

The wheel inlet mach number shall have a minimum value in order to satisfy the Euler equation of turbo machines for adequate energy transfer rate. This can be set as design criteria in order to minimize the friction losses at the entrance. [1]

The entrance velocity ratio $U_1/C_s = 0.7$ defines the optimal relation between the wheel velocity (U_1), the turbine inlet temperature and the pressure ratio (summarized into the isentropic spouting velocity, C_s). This relation is both theoretically and practically verified. Some more parameters are being used like the discharge velocity ratio, the specific speed and the loading coefficient. References [1-2] are describing these parameters and their optimal values.

Constructive design improvement

A specific constructive turbine design is initiated for small capacity liquefier. The following topics are handled with the application perspective of small turbines, standard product, large process application range and low costs. The proven robustness of the Linde gas bearing technology shall be kept without exception throughout Linde's product range.

In order to increase the rotational speed, the shaft geometry has to be optimized. The main design criteria are the 1st bending frequency, the local maximal stress at the axial thrust bearing disc and the radial bearing size. Both the material choice and the geometry are evaluated based on the above criteria. A maximal working speed of 5500 Hz has been reached, which corresponds to a 15% higher speed.

As the Linde bearings are dynamic bearings, the maximal allowable axial thrust is strongly dependent on the turbine running conditions (rotational speed, pressure ratio, pressure level, etc) on one side and on the manufacturing and assembling accuracy on the other side. The precision of the axial bearing is improved through appropriate material choice and manufacturing procedure. An important improvement has been reached in the reliability of the bearings due to an appropriate material choice (coal).

Due to the increasing importance of thermal losses by low turbine capacity, the design of the cold housing, the turbine shaft and of the interface isolation is reconsidered. A maximal thermal loss of 25 W at 70K process gas temperature is computed and measured, which corresponds to a 35% thermal loss reduction.

For small turbine dimensions, the effective average clearance can easily reach 25-30% of passage height. Clearance reduction can be reached by considering the manufacturing and assembling procedures as well as the running conditions of the turbine. Considering all these parameters the effective average axial clearance could be reduced to 10 % of the passage height.

Free surface blade definition, splitter blades, smaller blade thickness, etc can be introduced through today's modern manufacturing technology (5 axes machines and high speed milling). A competitive milling company manufactured new free surface high precision wheels with comparable prices to the usual radial shape wheels.

TESTS

A variety of turbine tests are performed under real conditions on an in house test bench. Helium as process gas has been used by operating temperatures between 20 and 70 K (inlet). The test bench is provided with accurate instrument technology and implementation know-how in order to reach a measured efficiency accuracy of ± 2% point. Reproducibility is achieved within a ± 0.7% point range. Temperatures are measured with Rh/Fe sensors. Sensor self-heating is reduced to a minimum. Thermoelectric forces (Seebeck effect) are compensated by appropriated measuring procedure (reversing poles).

RESULTS

Efficiency has been significantly improved in the first design step. TABLE 2 shows efficiency values for the current(0) and new (1) design.

TABLE 2. Turbine efficiency comparison

Design case	0	0	1	1
Turbine position	Tu 1	Tu 2	Tu 1	Tu 2
Isentropic efficiency	61%	67%	77%	77%

Optimal values are checked to be at $U_1/C_s = 0.7$. Optimal rotor incidence flow angles are not uniformly distributed. FIGURE 4 shows the comparison on a non dimensional mass flow diagram. The stage efficiency could be improved up to 20% point for the lowest non dimensional mass flow $\mu = 0.025$.

LIQUEFIER PROCESS ANALYSIS

Theory

The impact of improved expander efficiency on refrigerator/liquefier process is quantified through the exergy analysis which compares the energy transfer quality of the different components and reduces it in a common defined value, the exergy. Equation (3) shows the exergy transfer that occurs in a gas flow expanding from 0 (inlet) to 3 (outlet):

$$e_t = (h_0 - h_3) - T_a(s_0 - s_3) \tag{3}$$

$$(h_0 - h_3) = (h_0 - h_{0s})\eta_s = c_p T_0 \left(1 - \left(\frac{P_3}{P_0}\right)^{\gamma-1/\gamma}\right)\eta_s = f(T_0;\pi;\eta_s) \tag{4}$$

with: π = expansion pressure ratio; η_s = turbine isentropic efficiency.

$$T_a(s_3 - s_0) = T_a \int_0^3 \left(c_p \frac{dT}{T} - r\frac{dP}{P}\right) = T_a\left(c_p \ln\left(\frac{T_3}{T_0}\right) - r\ln\left(\frac{P_3}{P_0}\right)\right) = f(\pi;\eta_s) \tag{5}$$

with T_a= ambient reference temperature = 300 K and:

$$\frac{T_3}{T_0} = 1 - \eta_s\left(1 - \left(\frac{P_3}{P_0}\right)^{\gamma-1/\gamma}\right) = f(\pi;\eta_s) \tag{6}$$

Equation (4) describes the first term of equation (3) for ideal gases and is equal to the effective specific work transfer through the turbine. For a constant pressure ratio π and turbine efficiency η_s the effectively transferred work is proportional to the entrance temperature. The effect of improving the turbine efficiency increases the extracted exergy in mechanical form and increases the temperature difference generated through the expansion.

Equations (5) and (6) describe the second term of equation (3) for ideal gases and is equal to the exergy losses during the expansion. The losses remain constant for any inlet temperature level and are only depending on the pressure ratio π and the turbine efficiency η_s. FIGURE 5 shows a numerical example for the evolution of the losses as the efficiency grows from 70 to 100 % both for turbine inlet temperature 16 and 70 K. The slope of the curves shows that for both turbine inlet temperatures, the temperature difference gain or mechanical work gain across the turbine is relatively small in comparison to the exergy losses reduction.

Computed results

Based on the new efficiency values the total exergy loss reduction is computed from equations (3) for each turbine in both cases 0 and 1. The turbine losses relative to the cold box input power are reduced of 9% point, from 36 % down to 27%. This saved exergy (9% point) is in the range of the formally exergy flow provided as liquid Helium (12%). As the saved exergy is not at the process end stage, part of it is dissipated due to the inefficiencies of the downstream components (heat exchangers, process valves, etc).

Based on the effective performance of a cold box with current turbines, a process calculation is performed in order to validate the computation and the approach. The same approach is then used for a process calculation with improved turbines. The results in TABLE 3 show that the liquefier capacity is increased by 45%.

TABLE 3. Influence of turbine efficiency on liquefaction capacity for a given plant

Case	Compressor input power	Turbine efficiency		Liquefier capacity	Relative capacity
[-]	[kW]	Tu1	Tu2	[l/h]	[%]
0	110	61%	67%	33.6	100%
1	110	77%	77%	48.7	145%

CONCLUSION & PERSPECTIVES

By considering the downscaling effects in a turbine retrofit the expansion stage efficiency is increased by 16 % point for Turbine 1 and 10% point for Turbine 2. The total capacity of a standard liquefier fitted with these improved turbines can be increased to a 145% rated capacity. A plant equipped with new designed turbines is presently being commissioned to confirm the expected values.

The development work done on small scale turbines confirmed that scale down of expansion turbine is possible with only moderate reduction of efficiencies. Furthermore the work on the small turbines has resulted on valuable additional knowledge also applicable to larger turbines.

FIGURES

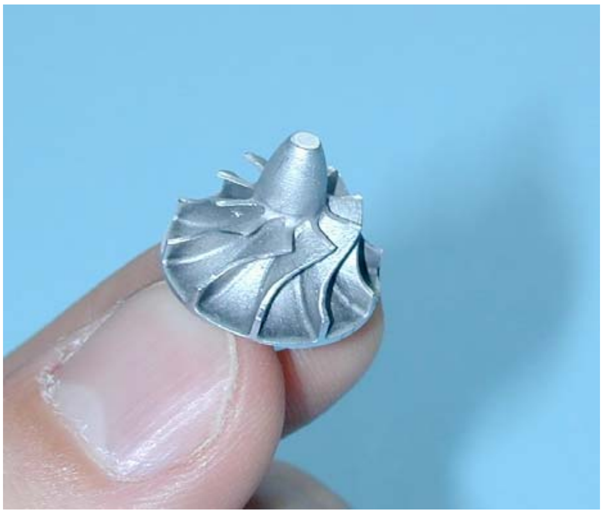

FIGURE 1 Turbine wheel geometry, Case 0

FIGURE 2 Turbine wheel geometry, Case 1

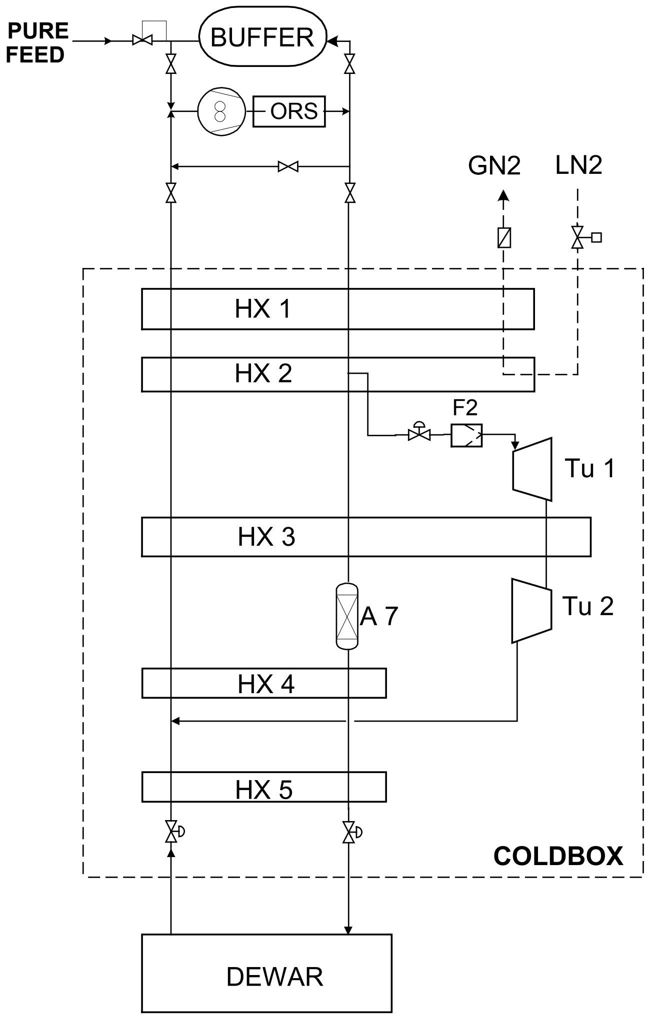

FIGURE 3 Process flow diagram of a typical small size liquefaction plant.

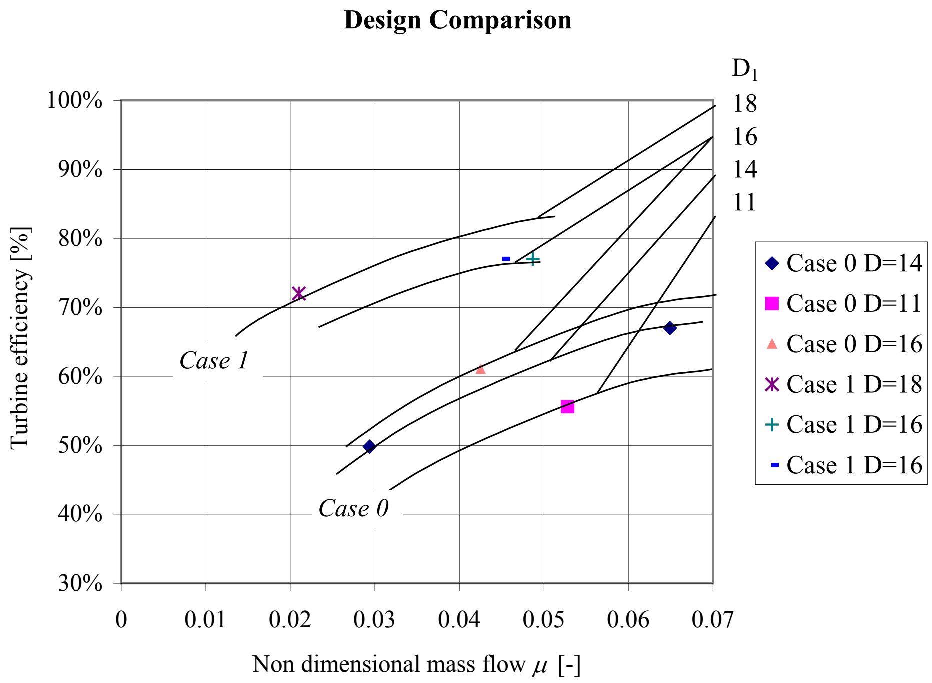

FIGURE 4. Comparison of turbine efficiency for current and new turbine design with wheel entrance diameters (D_1) between 11- 18 mm.

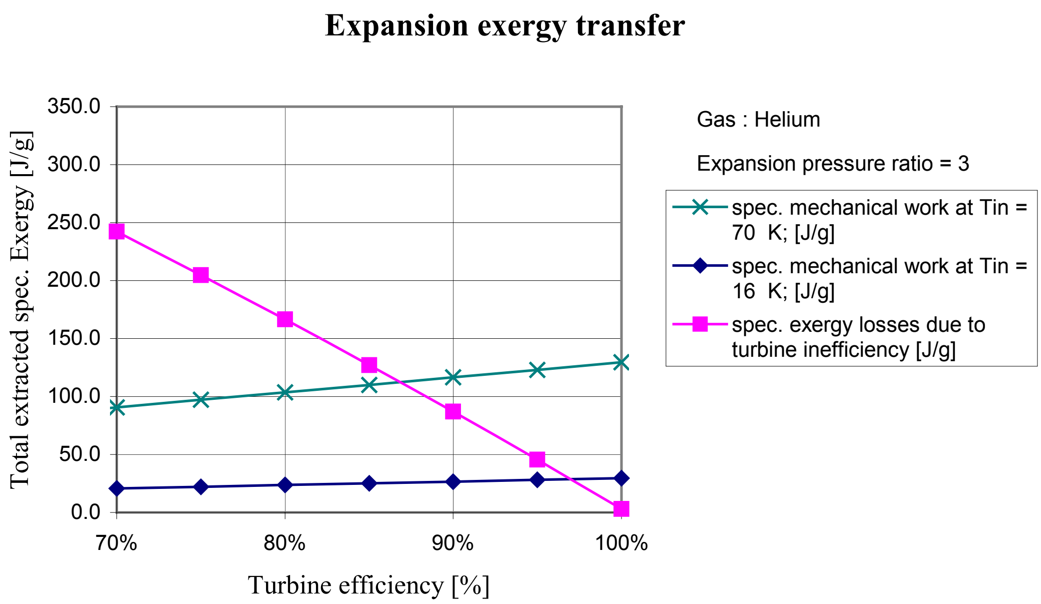

FIGURE 5. Evolution of turbine exergetic losses as the efficiency grows from 70 to 100 % and for turbine inlet temperature 16 or 70K.

NOTATION

h	enthalpy [J/KgK]
T	temperature [K]
P	pressure [bar]
e	specific exergy [J/g]
r	gas constant [J/kgK], radius [m]
$\dot{m}$	mass flow rate [kg/s]
C	absolute velocity [m/s]
U	wheel tip velocity [m/s]
C_s	isentropic spouting velocity [m/s]
C_p	specific heat at constant pressure [W/mK]
γ	specific heat ratio [-]
π	pressure ratio [-]
μ	non dimensional mass flow rate
ρ	density [kg/m^3]
η_s	isentropic efficiency [-]

Indices

t	turbine
s	isentropic transformation
a	ambient
0	turbine stage inlet
1	turbine wheel entrance
2	turbine wheel outlet
3	turbine stage outlet

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3. Ziegler, B. O. "Second law Analysis of the helium refrigerators for the HERA Proton magnet ring", in *Advances in Cryogenic Engineering* 31, Plenum Press, New York, 1986, pp. 693-698.