# DEVELOPMENT OF HIGH SPEED PELTON TYPE EXPANDER-GENERATOR FOR RECOVERY OF THE THROTTLING LOSS IN CARBON DIOXIDE REFRIGERATION SYSTEM

*Takeshi HIKAWA, Makoto KOJIMA,*
*ENVIRONMENTAL TECHNOLOGY LABORATORY, DAIKIN industries,LTD.,*
*1304 Knaoka-cho, kita-ku, Sakai, Osaka 591-8511 Japan*

**Abstract:** High Speed Pelton Type Expander-Generator was developed for recovery of the throttling loss in carbon dioxide refrigeration system. Many types of efficient positive displacement expanders have been investigated to recover throttling loss. Different from these work, compact and low cost type Pelton expander with high speed generator was chosen and evaluated. Pelton expander consists of nozzle, impeller and generator. The nozzle is an energy conversion device from the pressurized potential energy to the kinetic energy of refrigerant jet flow. Pressure distribution along the nozzle and jet momentum at the exit of the nozzle was measured with various nozzles and highly efficient nozzle design was found. The impeller extracts the kinetic energy of refrigerant jet flow. Various shapes of impellers were tested and highly efficient impeller design was found. The $CO_2$ heat pump cycle applying Pelton Expander-Generator with highly efficient nozzle and impeller designs turned out to improve system performance by eight percent.

**Key Words: Expander, Pelton turbine, Carbon dioxide, Heat pump**

## 1 INTRODUCTION

Extensive research on alternative refrigerant has been conducted to prevent global warming.
One of the promising alternative refrigerants is $CO_2$, since it has low GWP and safer characteristics. However, it has been commercialized so far only in a limited market such as domestic hot water which is called “Eco Cute”, since the cooling performance of $CO_2$ refrigerant is inferior to that of HFC refrigerants. $CO_2$ refrigeration cycle is operated under high-pressure condition. So the energy loss during the throttling process is considerably large. Various types of positive displacement expanders have been investigated to recover this large loss, but they have not been widely accepted to the market owing to their expensiveness.
Therefore, this paper introduces cost–effective compact turbine expander. Its compactness is promising since it allows extreme high speed. There are two major types of turbine expanders. One is a reaction turbine such as Francis type; another is an impulse turbine such as Pelton type or Turgo impulse type. The reaction turbine has a runner, where pressurized fluid flow gives rotational power. The impulse turbine mainly consists of nozzles and an impeller. A nozzle is an energy conversion device from pressurized potential energy into kinetic energy of jet flow. An impeller is an energy conversion device from kinetic energy of jet flow to mechanical rotation energy which generator can convert to electric power. Reaction turbine has a good advantage under the condition of low-pressure difference and relatively high flow rate. On the contrary, Impulse turbine shows advantages under high pressure difference and low flow rate which can be observed in conventional refrigeration cycle. Furthermore, impulse turbine has nozzles, which enables flow rate control like expansion valves in refrigeration cycle. This means that the replacement from expansion valve to impulse turbine expander is easy. The nozzles of Turgo Impulse turbine are set inclined to the impeller rotor body requiring additional space. Therefore, the Pelton turbine

was chosen by the advantages of compactness and suitability for refrigeration cycle with high-pressure difference and low flow rate.
Some investigations on turbine expanders in the refrigeration cycle have been conducted. E.Tøndell investigated Turgo Impulse turbine for $CO_2$ refrigeration cycle. Test expansion machinery was manufactured and its nozzle performance was evaluated. T. HE et. al applied Pelton turbine for HFC refrigeration cycle. Test expansion machinery was manufactured and turbine efficiencies at various test conditions were evaluated. However, there are still very few studies in the detailed turbine performance analysis such as nozzle performance analysis and impeller performance analysis. This paper describes compact Pelton turbine expander design and its experimental evaluation of nozzle efficiencies and impeller efficiencies. Nozzle jet momentum at the outlet of the nozzle was measured with various nozzles, and the nozzle efficiencies were figured out for each nozzle shape. On the other hand, nozzle internal pressure-profile along the flow was observed, and the relation between nozzle geometric specifications and pressure profile was evaluated. Furthermore, parametric experiments on the geometric specifications of the impeller were conducted. As a result, most effective geometric specifications were figured out for the Pelton turbine in the $CO_2$ refrigeration cycle.

## 2 PELTON TURBINE DESIGN

Figure 1 shows P-H Diagram of $CO_2$ refrigeration cycle. The expander replaces conventional expansion valve and recovers throttling loss.
Figure 2 shows a section of the test expander and indicates main components such as nozzle, impeller, generator, connecting shaft and bearings. The height and outer diameter of this test expander are 250mm and 70mm, respectively. High-pressure refrigerant from gas-cooler goes into nozzle, depressurized and accelerated along the nozzle. Accelerated jet flow goes into impeller bucket giving its kinetic energy. Impeller recovers kinetic energy into mechanical rotating energy which generator can convert to electric power. The generator is direct current type with 60mm diameter and is capable of high-speed rotation beyond 30000rpm. The generator is supported by the bearings at each end, and a nozzle and an impeller are located at the lower side of the generator. Depressurized refrigerant flows out from the bottom of the shell.

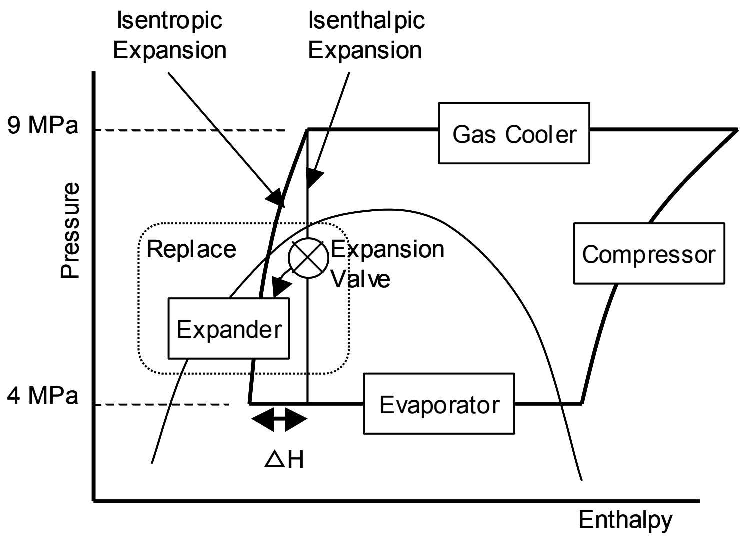



**Figure 1: P-H Diagram with Refrigeration Cycle**

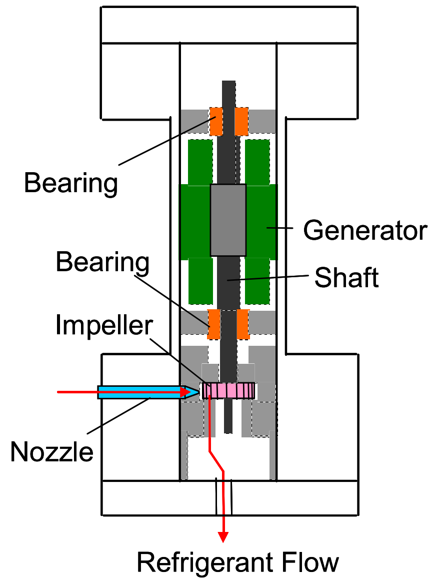



**Figure 2: Section of Pelton Type Expander**

## 2.1 Nozzle Structure

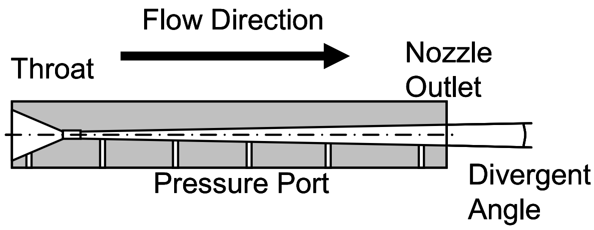



**Figure 3: Section of Nozzle**

The nozzle shown in Figure 3 is an energy conversion device from pressurized potential energy to kinetic energy along the flow to the outlet of the nozzle.
The pressure of the liquid refrigerant going through the throat goes down as it flows along the nozzle. Because the refrigerant starts to evaporate as the pressure goes down, the flow phase changes into two-phase mist flow after the throat. Two-phase mist flow loses pressure accelerating to high speed beyond 100m/sec along the down stream from the throat. Improper geometric shape of the down stream causes a shock wave and reduces nozzle efficiency.
Therefore, the nozzle efficiency is sensitive to down stream geometric specifications. The area ratio between throat area and nozzle outlet area of the test nozzle is varied by divergent angle in order to figure out the optimum one.

## 2.2 Impeller Structure

Figure 4 shows the sketch of the Pelton turbine applied in hydroelectric machinery. Buckets are mounted at the outer rim of rotor body. The jet flow going into the bucket is split by the water-cut edge, push the surface giving the power to the bucket and flow backward to leave from the bucket. Reduction in size keeping same complicated design makes the turbine considerably expensive. Therefore, simple carving twin arc bucket design at outer rim shown in Figure 5 is applied because of its easy manufacturing process, i.e. no fine assembles, cutting only outer radial rim. The impeller diameter is kept 32mm; thickness is 8mm, number of the bucket is twelve among all test impellers.
Nozzle and Impeller are mounted as close as possible so as to make high-speed jet flow goes into buckets before deceleration. The radial gap between shroud and the impeller is designed very small so as to decrease drag loss. The refrigerant leaving the buckets flows along the path between the rotor body side and shroud. This path is designed to promptly lead the refrigerant to the outlet hole at the bottom shell to prevent disturbing impeller rotation.

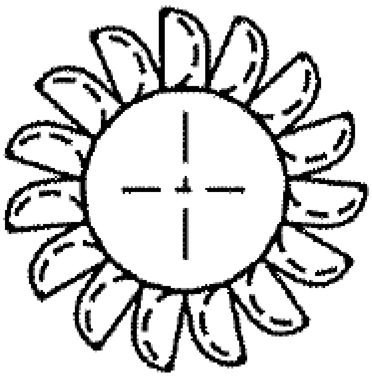


**Figure 4: Sketch of Pelton Turbine**

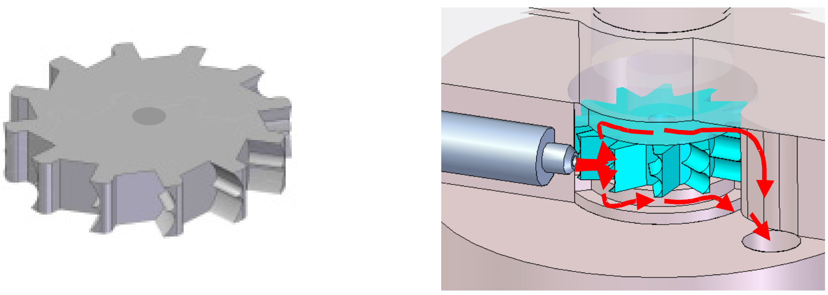
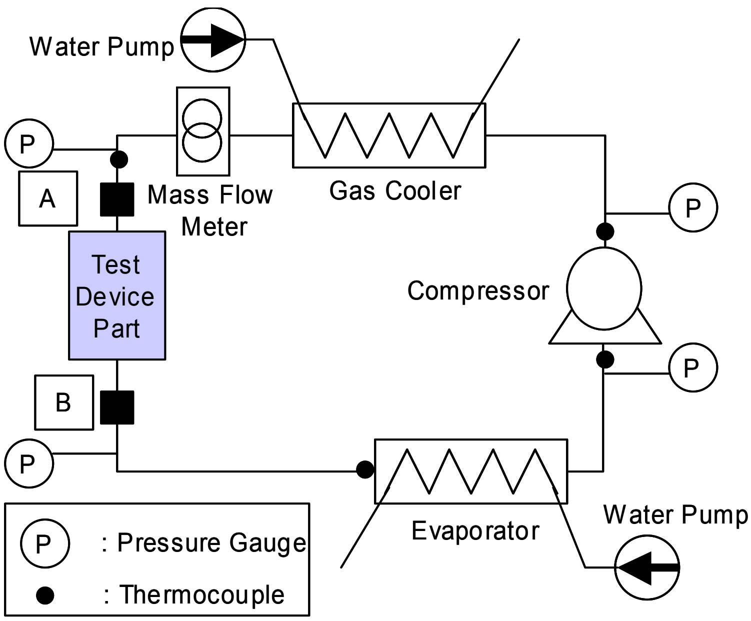


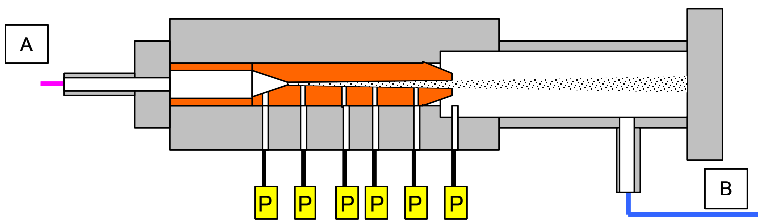
**Figure 5: Impeller Appearance and Impeller's Surrounding Structure**

## 3 EXPERIMENTAL METHODS

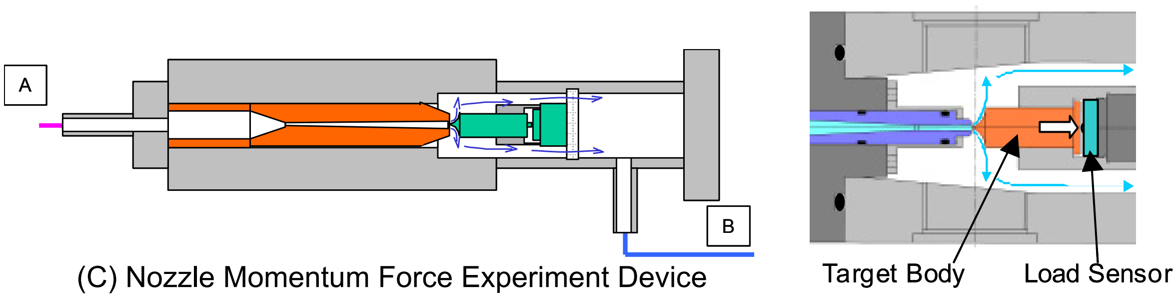
Figure 6 shows the schematic diagram of the refrigeration circuit with the test device part. Expander, pressure profile experiment device and nozzle momentum force measurement device can be connected there. Pressure, temperature and refrigerant mass flow rate are measured. Electric power output of the generator is converted to direct current at the AC-DC converter and direct current wattage was measured.
The experiment condition is High pressure / Low pressure: 9MPa / 4MPa, inlet temperature of the nozzle is kept 40 degree-C. This condition is derived from typical cooling condition of the $CO_2$ refrigeration cycle, because $CO_2$ cycle has inferior cooling performance to other conventional refrigerants.




(A) Schematic Diagram of the Refrigeration Circuit




(B) Nozzle Pressure Profile Experiment Device




(C) Nozzle Momentum Force Experiment Device

**Figure 6: Schematic Diagram of the Refrigeration Cycle and Experiment Devices**

### 3.1 Efficiency Evaluation

$\Delta H$ as shown in Figure 1 is the difference between isentropic expansion process and isenthalpic expansion process. It indicates theoretical expansion loss. The expander efficiency is expressed by the ratio of obtained electric power to theoretical expansion loss as Eq.1.

$$\eta_{\exp} = W / (\Delta H * G_r) \tag{1}$$

Where $\eta_{\exp}$ denotes expander efficiency, $W$ denotes output electric power of the expander, $\Delta H$ denotes expansion loss described above, $G_r$ denotes refrigerant flow rate. Expander efficiency is also expressed using nozzle efficiency and impeller efficiency in Eq.2, assuming that the generator loss is negligible.

$$\eta_{\exp} = \eta_{nozzle} * \eta_{impeller} \tag{2}$$

With this equation, impeller efficiency can be calculated from measured expander efficiency and measured nozzle efficiency.

Figure.6 (B) shows the apparatus of pressure profile experiment device, which is connected to the test device part and the measurement was carried out. Test nozzles have a series of small hole with 0.8mm diameter to measure pressure profile along the nozzles.
The momentum force test device is also shown in Figure.6 (C). It is connected to the test device part and measurement was carried out. The target body is located very close to nozzle outlet in order to catch jet flow before deceleration. The momentum force is transferred to the load sensor located behind the target body. Eq.3 is the conservation low of momentum.

$$F = G_r * (V_{nozzle} - V_{sc}) \tag{3}$$

Where $F$ denotes impact force, $V_{nozzle}$ denotes nozzle outlet velocity, $V_{sc}$ denotes the nozzle flow direction component of the leaving flow velocity from the target. This target body is designed to make the flow scatter to radial direction so as to minimize $V_{sc}$. Therefore, $V_{sc}$ is assumed to be negligible in Eq.3 and the $V_{nozzle}$ is calculated through Eq.4

$$V_{nozzle} = F / G_r \tag{4}$$

The nozzle efficiency is evaluated by the ratio of the kinetic energy to the theoretical expansion loss.

$$\eta_{nozzle} = \left( 1/2 * V_{nozzle}^2 \right) / \Delta H \tag{5}$$

# 4 RESULTS AND DISCUSSION

## 4.1 Nozzle Experiment

Table 1 shows the test nozzle specifications of pressure profile experiment. The divergent angle is varied to evaluate the effect of the area ratio. The diameter of the throat is kept constant among all test nozzles.

**Table 1: Nozzle Specifications**

| | Nozzle A | Nozzle B | Nozzle C | Nozzle D |
|---|---|---|---|---|
| Divergent Angle(degree) | 0.2 | 0.4 | 0.5 | 0.76 |

Figure 7 shows variation of the pressure profile with different nozzle divergent angles. X-axis is the distance from the throat, and Y-axis is the nozzle internal pressure. As shown in the pressure profiles, big over-expansion was observed as the divergent angle increased. It shows the generation of shock wave, which declines nozzle efficiency. On the contrary, as the angle decreased, no over-expansion was observed. Table 2 shows the experiment results of nozzle efficiency. No over-expansion profile turned out to be most effective through this evaluation. The nozzle efficiency of 87.7% was observed for this most effective profile.

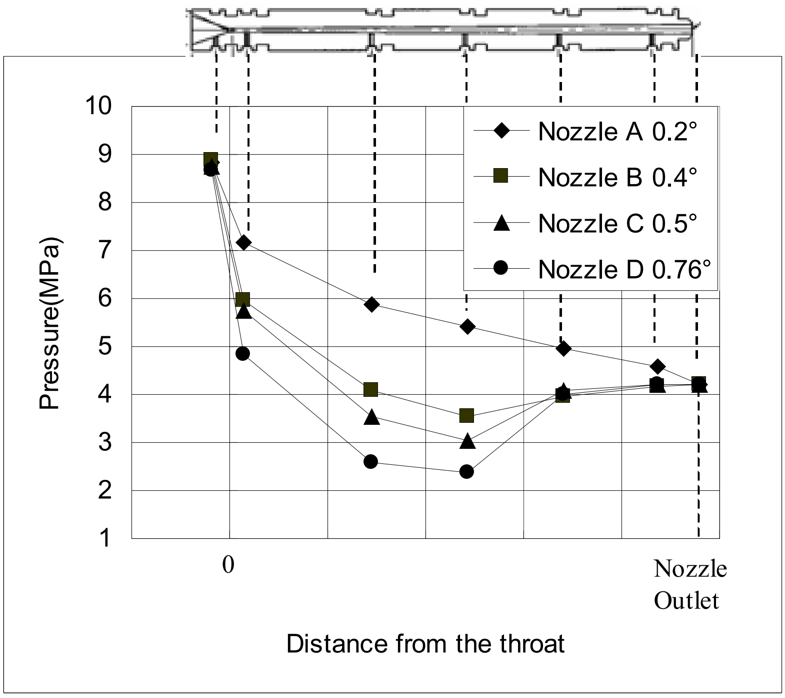



**Figure 7: Pressure Profile of the Nozzles**

**Table 2: Nozzle Test Results**

| | Nozzle A | Nozzle B | Nozzle C | Nozzle D |
|---|---|---|---|---|
| Divergent Angle(degree) | 0.2 | 0.4 | 0.5 | 0.76 |
| Nozzle Efficiency (%) | 87.7 | 54.7 | 46.9 | 27.3 |

### 4.2 Expander Experiment

Parametric experiments on the geometric specifications of the impeller were conducted to figure out the most effective structure for the Pelton turbine in the $CO_2$ refrigeration cycle. Figure 8 shows geometric parameter of the test impeller such as height and curvature.
Height means bucket height, and curvature means front concave side curvature. Since the thickness of the body is 8mm, the single arc realizes the radius of curvature beyond 2mm with sufficient concave shape. Single arc is the shape without water-cut edge shown in Figure 8.
Table 3 shows the specifications and the experiment results of the impellers for various heights and radius of curvatures. Expander efficiencies in both parametric experiments are normalized by the maximum efficiency as expander efficiency ratio.

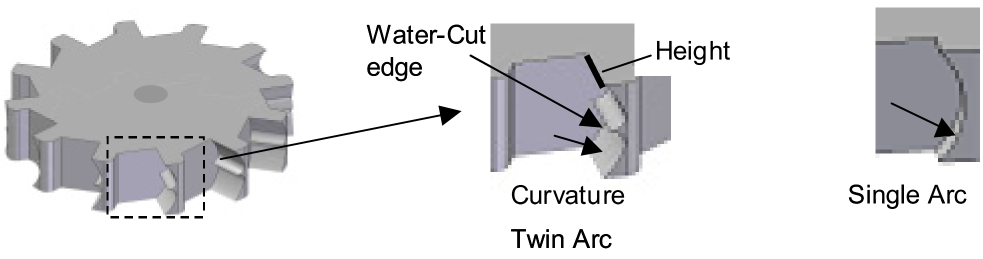



**Figure 8: Impeller Configurations**

**Table 3: Impeller Specifications and Test Results**

**(A) Height**

| | Impeller A | Impeller B | Impeller C |
|---|---|---|---|
| Height (mm) | 2.5 | 4 | 6 |
| Expander Efficiency Ratio (-) | 1 | 0.78 | 0.57 |
| Curvature | Radius of Curvature; 2mm with Arc; twin-arc | | |

**(B) Curvature**

| | Impeller D | Impeller E | Impeller F | Impeller G |
|---|---|---|---|---|
| Radius of Curvature (mm) | 2 | 4.5 | 6 | 12 |
| Arc | Twin | Single | Single | Single |
| Expander Efficiency Ratio (-) | 1 | 0.96 | 0.93 | 0.81 |
| Height (mm) | 2.5mm | | | |

As the height increases, the loss increases, shown in Table 3 (A). Height resulted in sensitive parameter. It is hypothesized that the contact power of the backward flow to the back of the bucket may increase as the height increases.
As shown in Table 3 (B), the curvature and the water-cut edge did not effect to the performance significantly. This implies that the flow didn't follow the sharp curvature appropriately, since this radius of curvature is considerably small for high-speed jet flow. Consequently, the most effective specifications as shown in Table 4 were chosen.

**Table 4: Impeller Specifications**

| Number of Buckets | Curvature (mm) | Arc | Height (mm) |
|---|---|---|---|
| 12 | 2 | Twin | 2.5 |

Lastly, expander experiment was conducted with this most efficient impeller and most efficient nozzle E under the same condition described above. The Coefficient of performance (COP) of CO2 refrigeration cycle was estimated to be improved by 8% with this expander.

## 5 CONCLUSION

High Speed Pelton Type Expander-Generator was developed for recovery of the throttling loss. Pressure distributions along the nozzle and jet momentum at the exit of the nozzle were measured with various nozzles. The nozzle with proper pressure profile turned out to be most efficient with nozzle efficiency of 87.7%.
As for impeller, different from hydraulic power Pelton turbine, simple carving twin arc design at outer rim was evaluated. Various shapes of impellers were tested, and the most effective specifications were found. The performance of the $CO_2$ refrigeration cycle applying this Pelton type Expander-Generator with highly efficient nozzle and impeller design turned out to be improved by eight percent.

Reference

T. HE, L. LI, C. XIA, P. SHU 2007 “APPLICATION OF PELTON-TYPE EXPANDER TO REPLACE THROTTLING VALVE IN REFRIGERATION SYSTEM” International Congress of refrigeration

T.Jusakon, M.Fukuta, T.Yanagisawa, Y.Ogi 2009 “Fundamental study of turbo expander for $CO_2$ refrigeration cycle” Proceedings of JSRAE Annual Conference

M.Nakagawa, M.Berana, A.Harada 2008 “Shock Wave in Supersonic Two-Phase Flow of $CO_2$ in Converging-Diverging Nozzles” International Refrigeration and Air Conditioning Conference at Purdue

E.Tøndell, 2006 “IMPULSE EXPANDER FOR $CO_2$” 7th IIR Gustav Lorentzen Conference on Natural Working Fluids

A.Zoughaib, D.Clodic 2004 “Design of micro turbo expander for use in small refrigeration systems” EUROPEAN MICRO AND NANO SYSTEMS