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# DEVELOPMENT OF CARBON DIOXIDE ROTARY COMPRESSOR

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## ABSTRACT

A prototype hermetic rotary compressor has been developed for use with carbon dioxide as natural working fluid. Two-stage compression cycle has been employed to reduce effect of the high operating pressure usual for the compression cycle of carbon dioxide compressors. The paper presents description of the compressor lubrication system, suction and discharge gas flow pattern, discharge valve design, results of the analytical study of the journal and thrust bearings, and some experimental test data. The modular design of the compressor elements helps to simplify assembly, reduce production cost, and improve performance of the compressor

## 1. INTRODUCTION

In the last few years' naturally occurring carbon dioxide received increasing attention as possible replacement of fluorocarbon-based refrigerants used at present for vapor compression cycle technology. The anticipating benefits of ecologically desirable carbon dioxide (R744) use as refrigeration working fluid include the following:

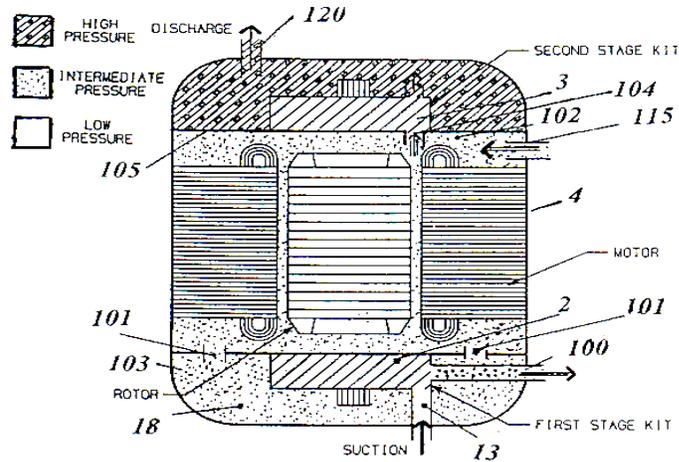
- ❖ Safety. It is chemically inert (non-flammable, not toxic) and produces no toxic or irritating decomposition products in a fire.
- ❖ Cost-effective due to abundant natural supply, reduced (up to 30%) refrigerant charge and tax saving where relief is allowed on alternative refrigerants (some EU countries imposed taxes on HCFCs).
- ❖ Reduced operating cost resulting from more than 10% of energy savings and easier operating and maintenance procedures. It is not necessary to recover, reclaim, or recycle R744. This contributes also to significant saving in personal, training and service equipment cost.
- ❖ Favorable thermodynamic performance. Better heat transfer properties due to lower viscosity, higher latent heat, specific heat and thermal conductivity compared to HCFC's and CFC's refrigerants.
- ❖ Reduced Unit Size. Due to the fact that the volumetric heat capacity of CO<sub>2</sub> is much higher than that of common refrigerants, the mass flow rate is much smaller for the same capacity. The compressors, vessels, diameters of the refrigeration piping, heat exchangers etc. are smaller for the units utilizing CO<sub>2</sub> as refrigerant.

One concern with R744 is the high operating pressure and suction/discharge pressure difference compared to common refrigeration process. Large difference between the compressor discharge and suction pressure will increase load on the bearings and required most robust seals. Fatigue problems due to higher impact velocities (lift and closure of the valve, for example) when compressing such a relatively dense gas as CO<sub>2</sub> (density is  $\approx 5$  times higher than that of R-22) must be considered. The large pressure difference also required a larger torque of the compressor motor. The fittings, piping, bolts, accumulators (mufflers), and housing of the compressor have to be stronger to meet the high working pressure of R744 and satisfy UL compressor enclosure strength requirements. That makes cost an issue. A CO<sub>2</sub> system could cost 15% more than conventional technology, though cost should decline to a price equivalent to current technology with increased production volume.

## 2. THE COMPRESSOR CONCEPT

Developed concept of the compressor aims at eliminating the drawbacks specified above by use of two-stage

compression process that eases the compressor workload (Dreiman N., Bunch R., 2003a). The two-stage compression cycle operates with three levels of pressure, which will be referred to as low, intermediate and high. Fig.1 presents the schematic cross-sectional view of a preferred embodiment of two-stage two- cylinder rotary



*Fig.1.Schematic cross-sectional view of the compressor gas flow*

compressor superimposed with the gas flow. The low-pressure suction gas is supplied directly to the lower end compression module 2 (first stage). As the refrigerant gas is compressed in the first stage it is discharged directly through the refrigerant discharge port (not shown) and discharge pipe 100 to the intercooler (not shown) located outside of the compressor casing. Subsequently, the cooled refrigerant gas is introduced into the cavity 102 (an intermediate pressure cavity) through the fitting 115 of the electric motor module 4 which is in fluid communication with the cavity 103 of the lower end compression module 2 through the oil passages 101 which allow lubricant to be reclaimed by oil sump 18. Introduction of the cooled refrigerant gas into the electric motor cavity 102 helps to cool as motor so the lubricant, and reduce the heat transfer to the suction gas of the first stage. The refrigerant gas introduced to the upper end compression module 3 through the suction port 104 made in the wall 105, discharged in cavity of the module 3, and distributed to a unit through the fitting 120.

### 3. DETAILED DESCRIPTION OF THE COMPRESSOR DESIGN

The modular design compressor shown in Fig.2 provides a positive displacement two-cylinder two stage rotary hermetic compressor comprising the lower end compression module and the upper end compression module which are coaxially coupled to the opposite axial ends of the motor module to form vertically upstanding compressor. In the exemplary embodiment of the compressor lower end module are similar in design to the upper end module and both modules comprise integral parts combining main bearing 1, housing segment 5, optional axial or radial suction and discharge ports, compressor external mounting 6 (lower module), rotary pumps 7. Reduced diameter of the upper module having the highest pressure will lift max. limits of allowable burst and working pressures without increase thickness of the walls ( $P = 2St/D$ , where  $P$ -internal burst pressure;  $S$  -allowable stress;  $D$ -diameter of housing;  $t$ - nominal wall thickness) and reduce weight of the compressor.

#### 3.1 Lubrication system

Taking into consideration the oil-returning ability, suitable miscibility, higher chemical stability and better lubricity under high pressure, PAG oil (glycol oil) have been selected as the lubricant. In the developed compressor the positive displacement oil pump 8 is operable associated with the end of crankshaft 17, all of which are submerged in the oil sump 18. The crankshafts of the compressor rotary pumps are coupled by sleeve 20 so that the oil delivery channels 30 of both crankshafts located along the axis of rotation are in fluid communication and connected also to the recess 31 machined at the top end of the upper crankshaft (Dreiman N., Bunch R., 2003b). Oil delivered to the crankshafts lubrication channel 30 is able under centrifugal forces to flow into a series of radially

extending passages of the crankshafts to the journal bearings surfaces. The rest of the lubricant supplied to the cavity 31 (see Fig.3a), is directed by a downwardly inclined channel 29 machined in the upper outboard bearing, axial channel 32 machined in

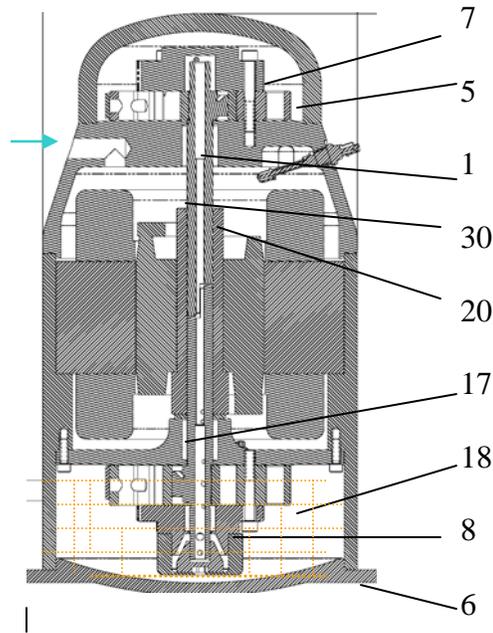


Fig.2. *Two-stage hermetic carbon dioxide rotary compressor*

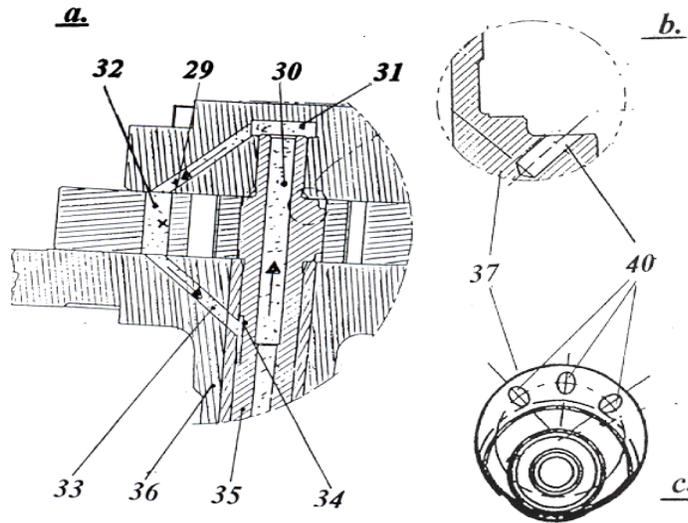


Fig.3. *Lubrication system of the compressor.*

the upper cylinder, downwardly inclined channel 33 machined in upper pump main bearing, to the annular cavity 34 defined by recessed area in upper crankshaft 35 and upper journal 36. The system of

such interconnected channels are used for delivery of the oil to the bearings and final return of the oil to the sump18. The Oil Circulation Rates (OCR) data for the compressor tested without oil separator in the system and with the oil separator are shown in Table 1.

Table 1: Oil Circulation Rates

OCR Measurements <i>without Oil Separator</i>							
Test No.	Suction Temp., °C	Pressure, kPa		Mass Flow Rate, g/s	Results		
		Suction	Discharge		CO <sub>2</sub> , g	Oil, g	OCR, wt. %
1	20.2	4476.8	8251.0	66.5	79.6	1.3	1.61
2	21.9	4531.9	9543.1	62.8	108.9	1.7	1.54
3	21.1	4507.8	11064.7	56.7	143.5	1.5	1.03
OCR Measurements <i>with Separator</i>							
Test No.	Suction Temp., °C	Pressure, kPa		Mass Flow Rate, g/s	Results		
		Suction	Discharge		CO <sub>2</sub> , g	Oil, g	OCR, wt. %
1	20.5	4496.8	8289.6	68.8	105.4	0.2	0.19
2	20.9	4521.6	9478.2	61.2	115.2	0.4	0.35
3	20.7	4525.7	10973.7	58.2	114.3	0.4	0.28

Due to extended length of the oil channels, distribution of the oil to the bearings, especially upper bearings, can be delayed, so it will affect the formation of an oil film that separate the interfacing surfaces. During the peak load operation of the compressor the frequency of starting and stopping is high, so some amount of oil used to form the film will at stop flow downwards due to the gravity forces and rest of the oil between the external surface of the shaft and facing surface of the bearing at start will not support formation of adequate film thickness. To improve lubrication at start blind holes 40 have been machined at the outlet part of radially extended lubrication channels of the crankshaft and in the upper planar surface of the lower and upper kit shaft eccentrics 37 (see Fig.3b, 3c) in the way that remaining oil fill dead-end holes cavity at the compressor stops, and supplied oil immediately to the inner surface of the bearings surfaces due to centrifugal force at the compressor start.

### 3.2 Crankshafts Bearings

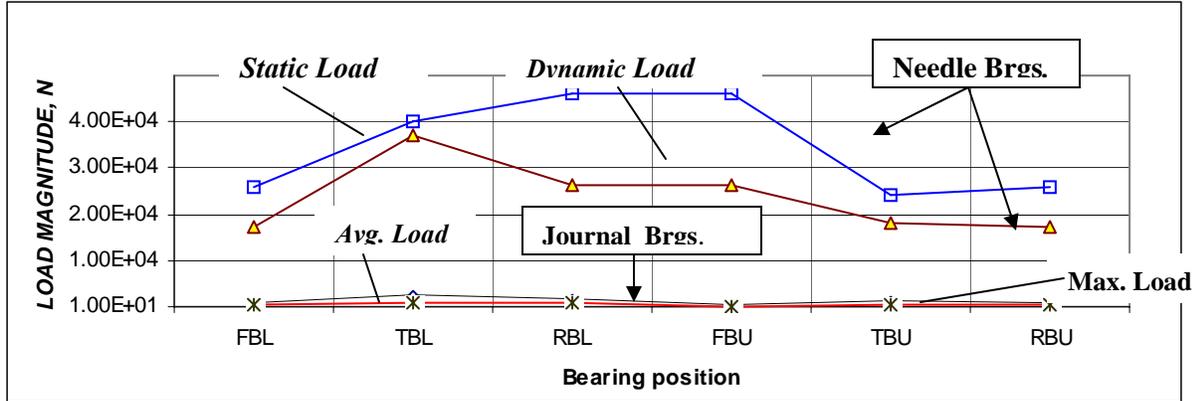
The study of a trans-critical CO<sub>2</sub>-compressor shows that the crankshaft bearings experience about five times more force load than a typical R-22 compressor (Hubacher, B., Groll, E., 2004). The journal bearings design of TPC production single cylinder rotary compressors used in the two-stage prototype have been not significantly modified to handle the combination of high pressure and low oil film thickness due to dissolving of carbon dioxide in the refrigeration oil. The analysis of the hydrodynamic journal bearing proved that the force loads acting on the journal bearing of given CO<sub>2</sub>-compressor are factors higher than for a conventional fluorocarbon-based compressor. Relatively high load magnitude and comparable thinner oil film have been analytically predicted for the outboard bearing of the lower pump, eccentric and outboard bearing of the upper pump (Dreiman N, Bunch R., 2004a). It was found that the oil film thickness of the hydrodynamic bearings couldn't be improved to satisfaction without changes of the crankshaft and kits parts parameters, so the needle bearings have been used as an alternative to the hydrodynamic lubrication bearings. The maximum force load magnitude for existent journal bearings and dynamic force load limits for alternative needle bearings are shown in Fig.4. The life limit of the chosen bearings calculated for the maximum load at 3450 RPM speed conditions is 144000hrs.

### 3.3 Thrust bearing

Thrust bearings of contemporary rotary compressors are positioned (as usual) on the opposite sides of the crankshaft eccentric that is located inside of the compression chamber. The thrust area is relatively small due to the limited space inside the compression chamber and has to support the suspended weight of two crankshafts and motor rotor in the developed CO<sub>2</sub> compressor. It creates conditions for partial or total overloading of the bearing. The total axial force  $F$  applied to the thrust surface

$$F = F_M + F_R + F_C, \quad (1)$$

where  $F_M$  is the motor axial (solenoid) force,  $F_R$  and  $F_C$  correspondingly is gravity force of the enlarged rotor and two crankshafts.



**Fig.4. The journal and needle bearings force loads.**

The motor axial (solenoid) force can be computed from the equation below:

$$F_M = 0.0117 P_H (60/f) (I_{M0} E_0 / L_0) (L_0/L)^2 [1 - 2\pi^{-1} (h/g)] \quad (2)$$

where  $P_H$  - phase number (for single phase =2),  $f$  - line frequency,  $I_{M0}$  - magnetizing current in amperes,  $E_0$  - line voltage,  $L_0$  - stator core stock height,  $L$  - effective core height,  $h$  - misalignment, and  $g$  - rotor-stator air gap. The dynamic of the thrust bearing during start and operation of the compressor is governed by the torques exerted on it. Reduction of the oil viscosity due to the fact that carbon dioxide is dissolved into the lubricant and high load squeeze the oil and thus, reduce the oil film thickness to a critical minimum. Due to the sector-of-circle shape of the thrust bearing (production crankshafts), a geometric converging wedge for fluid friction is not shaped. This factors trigger possibility for metal-to-metal contact, increased friction and accelerated bearing failure. The boundary friction loss  $F_L$  is

$$F_L = 2\mu W_S (R_{S2}^3 - R_{S1}^3) / 3 (R_{S2}^2 - R_{S1}^2) \quad (3)$$

where  $\mu$  - coefficient of friction,  $R_{S1}$  and  $R_{S2}$  - inside and outside radius of the thrust surface,  $W_S$  - weight of rotor and two crankshafts. With the addition of the axial solenoid downward force the loss factor will be significantly higher. In the developed rotary compressor traditional thrust bearing has been eliminated by relocating the thrust surface from inside of the compression chamber of the kits to the top of the main bearing hub 40 to improve the bearing load conditions (Dreiman N. et.al., 1996). New circular thrust bearing with radial grooves made from Vespel **SP-21** material (DuPont TM) has been located in between machined facing surfaces of the lower kit hub 1 and crankshafts sleeve 20. The polyamide material used to form the thrust bearing is characterized by a very low coefficient of static and kinetic friction. Another beneficial characteristic associated with polyamide is its broad temperature range and thermal stability. Circular shape of new thrust bearing helps to form circumferential periodic pattern of the oil film and prevents metal-to-metal contact. Radially extended grooves (Fig.5) on surfaces facing hub and planar surface of the sleeve have been provided in the thrust bearing for distribution of the oil. New thrust bearing located outside of the compression cavity and the thrust area of the bearing has been increased to reduce the load. The Pressure- Velocity ( $PV$ ) loading of the thrust bearing has been computed for numerous external and internal diameters. The following parameters have been used in calculations:  $P = 4W/\pi (D^2 - d^2) \xi$  is the static loading per unit area, psi ( $\text{kg}/\text{cm}^2$ );  $W$  - static load, lb (kg);  $A$  - area of the thrust surface,  $\text{in}^2$  ( $\text{cm}^2$ ),  $\xi$  - coefficient of used surface,  $D_m$  - medium dia. in (cm);  $N$  - speed of rotation, rpm (cycles/min);  $V = \pi (D_m N)$  is linear velocity, m/min ( $\text{cm}/\text{min}$ );  $PV$  - psi-ft/ $\text{in}^2$  min ( $\text{kg}\cdot\text{m}/\text{cm}^2$  sec). For the optimum performance of Vespel thrust bearing, it is best not to exceed a ratio of outside to inside diameter ( $D/d$ ) of 2. Ratios greater than 2 can cause overheating at the outside edge, and problems may arise from lack of flatness. New thrust bearing has up to 3 times larger thrust area than the single cylinder production compressor. Low friction coefficient of DuPont **SP-21** material has been beneficial at start and continuous operation of the compressor.

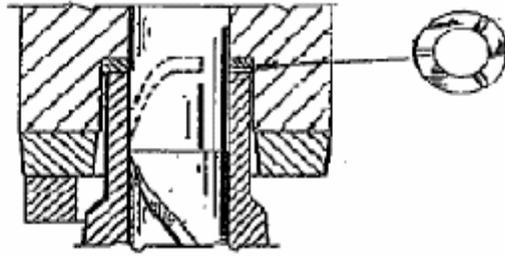


Fig. 5. Cross-sectional view of the motor-rotor, bearing hub and new thrust bearing.

### 3.4 Discharge valve

One of the factors affecting compressor efficiency is the re-expansion gas volume that is defined as the volume of space within the pumping cylinder when the piston is at the top center or the end of its pumping stroke. Since the discharge reed valve in contemporary rotary compressors located on the outer side of the plate, and the valve plate must have a certain limited minimum thickness (due to higher pressure with CO<sub>2</sub>) to give it the required strength, the volume of the discharge port and the total clearance volume will increase. Developed new type check valve provides an integral one piece valve-spring-retainer assembly that can withstand higher pressure difference

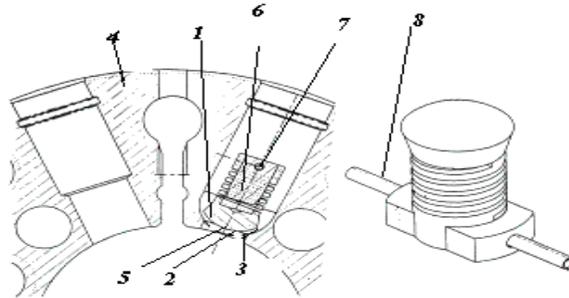


Fig. 6. Discharge valve mounted in the cylinder block

across the valve, has little re-expansion loss, permits an enlargement of the discharge port to reduce the over-compression and designed as integral multifunctional part to reduce cost and simplify assembly [Dreiman N., Bunch R., 2004b]. Referring now to Fig.6, the valve member 1 includes integral support, spring and a semi-spherical end portion 2 facing an exposed semi-spherically shaped seating surface 3 machined in the cylinder block 4. A valve seat 3 is formed about annular port 5. The spherical portion of the discharge valve member 2 substantially fills at closing annular port 5, reducing the gas re-expansion volume of the discharge port. Spherically sealing surface 2 facing compression chamber has substantially its complete surface immediately exposed to fluid pressure generated during valve opening. The curved shape of sealing surface 2 exposes a larger surface area than any exposed flat surface of the same diameter prior art discharge valve members. This maximized exposure of spherical valve surface 2 to discharge refrigerant flow accelerates the discharge valve opening thereby increasing compressor efficiency. The valve member reciprocates between a first closed position engaging the semi-spherical seat 3 and a second open position spaced longitudinally from the valve seat. The member 6 is provided as a guide and retainer. The arrangement of holes 7 permits easy assembly since alignment of valve member 1 and guide member 6 are located upon pin 8.

## 4. EXPERIMENTAL TEST RESULTS

The prototype two-stage hermetic rotary compressor has 16.1cc displacement volume at the first stage and 11.7cc at the second stage. The compressor has been design for fixed speed operation. The estimated cooling capacity of the compressor is 11kW. The performance of the compressor evaluated in terms of capacity, coefficient of performance (COP), volumetric efficiencies has been reported earlier [Dreiman N, Bunch R., 2004a; Hwang Y., Radermacher R., 2005].

One of the promising application of the compressor using CO<sub>2</sub> as refrigerant is in hot water heat pump system. In order to assure that such system can be operate at the optimum heat rejection pressure and meet the load requirement, a control system is needed which pull down or increase the temperature to more comfortable level as quickly as possible. There are three methods of compressor capacity modulation: variable speed, variable displacement, and control system that pull-down operating cycling of the power delivered to the compressor. Unlike systems that control cycling pattern or variable displacement, the compressor capacity can be changed by adjusting the speed of the compressor to bring more or less cooling or heating power to change the temperature to the required value. Tecumseh Products Company CO<sub>2</sub> Test Stand and GPD 515/G5 (Yaskawa Co.) Variable Frequency Drive has been used to evaluate effect of the compressor variable speed operation on Mass Flow Rate (MFR) and discharge gas temperatures of the compressor. The compressor operating conditions are specified in Table 2.

Table 2: Test conditions

Test No.	Evaporator Temp., °C	Evaporator Pressure, kPa	Suction Temp., °C		Discharge Pressure (Stage II), kPa
			Stage I	Stage II	
1	7.2	4198.9	18.3	54.4	9652.7
2	7.2	4198.9	18.3	54.4	11031.6
3	7.2	4198.9	18.3	54.4	12410.6

The speed variation of the compressor was in the range 1725 –5003 RPM (30 –87Hz). The changes in mass flow rate at compressor discharge pressures (Stage II) 9652.7 kPa, 11031.6 kPa, and 12410.6kPa with speed increase from 30Hz to 87Hz are shown in Fig. 6. The mass flow rate decreases at all compressor speeds with increase of the Stage II discharge pressure due to the growth of the pressure ratio.

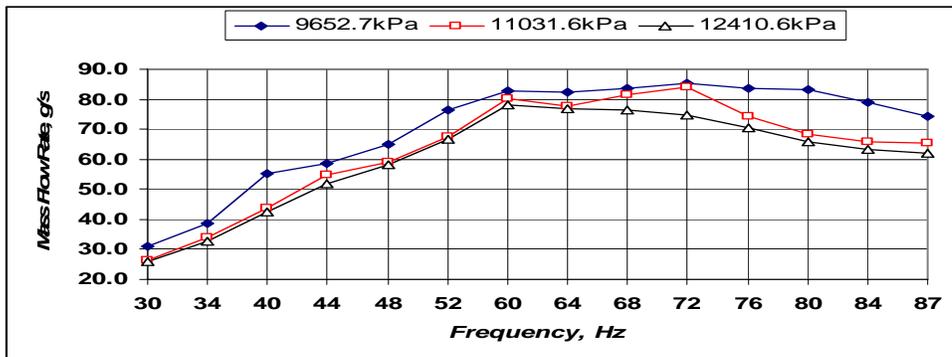


Fig.6: Effect of the compressor speed on Mass Flow Rate at Stage II Discharge Pressures 9652.7kPa, 11031.6kPa, and 12410.6kPa.

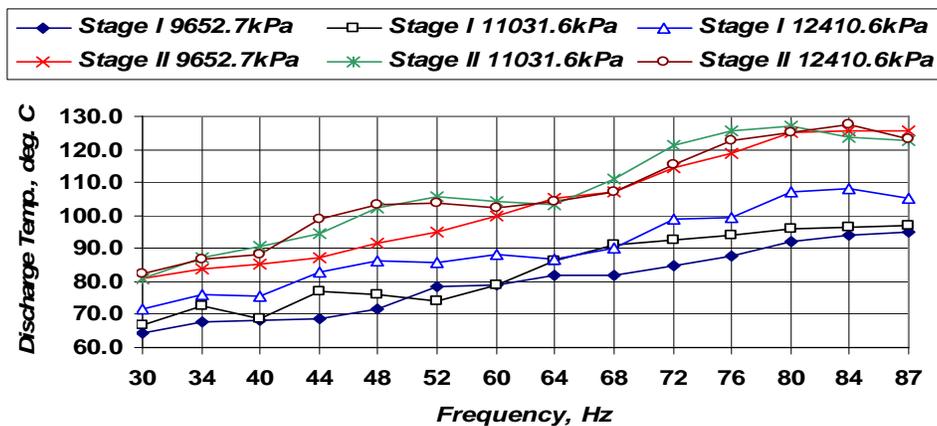


Fig.6: Effect of the compressor operational speed on the Stage I & Stage II Discharge Temperatures at Discharge Pressures 9652.7kPa, 11031.6kPa, and 12410.6kPa.

As long as volumetric efficiency reduces, the mass flow rate decreases. The most dynamic changes of the MFR have been recorded in the speed range 30Hz to 64Hz for all Stage II discharge pressures. The MFR does not show real changes after 64Hz operational speed for the tested discharge pressures due to the increase of the internal leaks of the compressor.

The discharge temperatures of Stage I and Stage II reach, correspondingly, 107.9° C and 127.8° C at 84Hz compressor operational speed with 12410.6kPa discharge pressure. The Stage I provides 20-22% higher discharge gas temperature when the frequency increases from base 60Hz to 87Hz at all tested pressures. Furthermore, the Stage I temperatures reduced 15-19% with decrease of the operating speed from base 60Hz to 30Hz. Recorded increase of the Stage II discharge temperatures with the grow of the operating speed (from base 60Hz) are in the range 20-26%. Reduction of the compressor operation speed from 60Hz to 30Hz decreases the discharge temperatures of the Stage II 19-22% at all tested discharge pressures.

## 5. CONCLUSIONS

- ❖ Developed two-stage rotary hermetic compressor utilize (with very little or no modification) the following production single cylinder compressors parts: cylinder blocks, rollers, vane, vane spring, O-rings, and crankshafts.
- ❖ Advance lubrication system distributes oil to the bearing immediately at start. The Oil Circulation Rates (OCR) data for the compressor tested without oil separator in the system was in the range 1.03-1.61 wt. % and with the oil separator 0.19-0.28 wt. %.
- ❖ Developed new thrust bearing has been formed from the polyamide material and located outside of the pumps compression chambers. It improves reliability of the compressor and reduces associated frictional losses.
- ❖ The most dynamic changes of the MFR have been recorded in the speed range 30Hz to 64Hz for the tested discharge pressures. The MFR does not show real changes after 64Hz operational speed.
- ❖ The discharge temperatures of Stage I and Stage II reach max. values, correspondingly, 107.9° C and 127.8° C at the compressor operational speed 84Hz and discharge pressure 12410.6kPa.
- ❖ The discharge temperatures of both stages decrease 19-22% with reduction of the operating speed from the base 60Hz to 30Hz. The discharge temperatures grow 20-26% with increase of the operating speed from base 60Hz to 87Hz.
- ❖ The magnitude and direction of the temperature change can be controlled (defined) by the optimum operating speed of the compressor.

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