A New Approach to Mechanical Loss Measurement of a Reciprocating Compressor

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012057 doi:10.1088/1757-899X/604/1/012057

A New Approach to Mechanical Loss Measurement of a Reciprocating Compressor

Husnu Kerpicci[1], Caglar Sahin[1], Ahmet Refik Ozdemir[1]

1 Arcelik A.S. R&D Directorate 34950 Tuzla Istanbul Turkey

Corresponding author: +902165858496
E-mail: husnu.kerpicci@arcelik.com

Abstract. Changing energy regulations urge all appliance manufacturers to use more efficient components in their products. Reciprocating compressor is one of the most crucial part for the energy efficiency of a refrigerator. Not only electrical and thermodynamic but also mechanical efficiency is the crucial issue determining the coefficient of performance (COP) of a compressor. Therefore, precise and trustworthy measurement comprising the frictional losses is one of the critical parameters to anticipate the real effect of novel design concepts. The objective of this study is the development of a new approach to measure mechanical losses of a reciprocating compressor. The method is based on the precise measurement of deceleration. A test rig, where the oil temperature, ambient pressure and temperature are able to be adjusted, is established. Measurements are performed under different average load for varying oil viscosity level. Load conditions are provided by adjusting ambient pressure of the chamber in which compressor body is placed. A heater is used on the body of the compressor to simulate temperatures corresponding with real temperatures acquired from the measurements of the compressor. Compression work is calculated by measurement of absolute cylinder pressure with respect to angular position of the crank shaft. Results are promising to use this measurement system for more detailed investigation.

1. INTRODUCTION

Parametric experimental studies coupled with engineering software has become an essential way for improvement the performance of reciprocating compressors. A lot of research dealing with the experimental set ups, modelling and simulating the compressors and their components are present in the literature. There are also some remarkable researches carried out in order to develop optimum compressor bearing design with minimum mechanical losses.

Journal bearings in hermetic reciprocating compressors were analyzed experimentally and numerically by B.J. Kim, et al [1]. In this study, the critical Sommerfeld number was proposed as a design criterion. Okaichi, et al [2] investigated the relationship between the bearing specifications and bearing loss by using hydrodynamic lubrication analysis of the main and the lower bearing. In their measurement system, the low voltage was applied between bearing bush and crank shaft. The change of the voltage indicates the shift of lubrication condition from hydrodynamic to mixed lubrication.

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012057 doi:10.1088/1757-899X/604/1/012057

Rolling piston-type rotary compressor's shaft torque measurement was carried out with torque sensor by Sakitani et al [3]. Matsushima et al [4] measured the torque of the compressor directly attaching a strain gauge to the part of the rotary compressor crankshaft.

Sato et al [5] investigated both experimental and analytical approach focusing on the behavior of thrust bearing in scroll compressor and optimization of thrust bearing achieved approximately 2% improvement of total efficiency. A technique of measuring directly the friction losses occurring on a single piston reciprocating compressor mechanism presented by Dietmar et al [6]. Won-sik Oh et al [7] developed a technique of measuring the friction losses of a compressor by using accelerometer.

In all various operating conditions, providing adequate oil to the bearings is necessary for safety operation of the compressor. During compression cycle, the minimum oil film thickness in the bearings occurs where the pressure in the cylinder reaches to maximum value. In order to provide safe hydrodynamic lubrication in the bearings, minimum oil film thickness must be approximately 3 times that of the surface roughness value of the bearing.

In this study, a new approach based on the precise measurement of deceleration for measuring mechanical losses of a reciprocating compressor was developed. Mechanical losses at various oil temperatures under two different load conditions were measured and acquired results presented.

2. MEASUREMENT SYSTEM

2.1 Description of test rig

Schematic view of the set-up is shown in Figure 1. The set-up consists of an electromotor with a gear fixed on its shaft (left side), another gear wheel placed in the middle, which is controlled with a pneumatic system, functioning as a coupling and on the right side the compressor body with shaft. Instead of the rotor, another gear wheel is positioned on the compressor shaft. This latter gearwheel is used to rotate the compressor shaft and next to that it significantly increases the inertia of the compressor (Figure 2). This gearwheel is designed according to the original weight of the rotor. On the top of the compressor shaft an encoder is positioned, which can measure the position and speed of the shaft. On the compressor cylinder a spacer plate has been mounted to increase the cylinder volume. On top of the plate the original suction valve leaf is placed and the original valve plate with cylinder head. A pressure transducer and a thermocouple have been placed on the discharge port therefore it is closed. In this way compression takes place when the piston is moving to the top dead centre, increasing the force on the piston, and an expansion takes place when the piston is moving to the bottom dead centre, decreasing the force on the piston. Under these circumstances, no discharge of the gas takes place. Due to piston leakage a small amount of the gas is leaking from the cylinder during one cycle, this leakage is compensated by a small flow through the suction port when the piston is near the bottom dead centre. Moreover, an optical encoder positioned on the top of the crank shaft is used for the measurement of the angular velocity and angular position of the crank shaft for each rotation. The pressure sensor placed onto the discharge port is used to calculate the force on the piston. Hence, position-force diagram has been constructed with position measurement provided by the encoder. Therefore, the work performed by the piston has been calculated with integration of the that diagram.

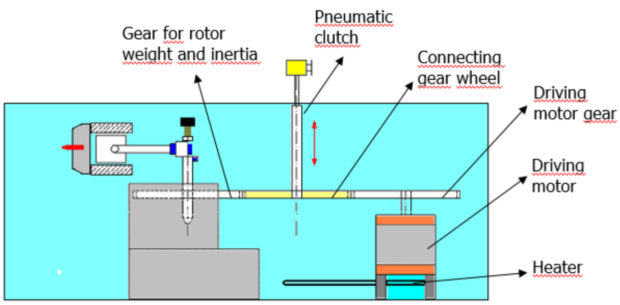

Figure 1. Schematic view of the measurement system

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012057 doi:10.1088/1757-899X/604/1/012057

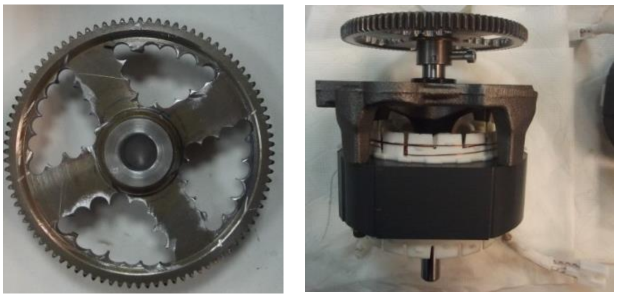

Figure 2. Gearwheel used for the rotor weight and inertia

The general measurement methodology works with the deceleration of the shaft. The basic idea is that when the gear is disconnected, the kinetic energy of the compressor mechanism is dissipated by friction and by the work performed by the piston on the gas in the cylinder:

$$T_{tot} = J \,.\, \dot{\omega} \qquad [1]$$

$$P_{tot} = T_{tot} \,.\, \omega \qquad [2]$$

$$P_{friction} = P_{tot} - P_{piston} \qquad [3]$$

where

T_{tot} : Total torque on shaft [Nm]
P_{tot} : Total work on shaft [W]
$P_{friction}$: Friction work [W]
P_{piston} : Piston work [W]
J : Total inertia of shaft, rotor and translating parts [Nms^2]
$\dot{\omega}$: Deceleration of the shaft [rad/s^2]
ω : Angular velocity of the shaft [rad/s]

The deceleration of the shaft can be accurately measured using an optical encoder. The force on the piston is generated by the pressure in the closed volume on top of the cylinder. The work of the piston is calculated from the measured pressure of transducer. The rotation is created using the variable speed motor, which can be disconnected using a pneumatic coupling to start the deceleration measurement. Oil pumping system and submerged depth of the crank shaft were kept identical to the real circumstances. Set temperature of the oil is managed using a PID controlled heater.

Temperature measurements were carried out with T-type thermocouples which were placed on various positions of compressor body. Ambient and oil temperatures were measured with RTD sensors positioned in the chamber and oil sump respectively. Because of the flammability of the refrigerant the measurements were carried out with air. The average load is predetermined with ambient pressure which was balanced with a relief valve and a vacuum pump according to desired operating conditions. Since cylinder wall temperature is also critical for the compression work, an exclusive heater placed on the compressor body is used to simulate compressor's real operating conditions. Figure 3 shows the compressor prepared for the measurements on which encoder, heater and thermocouples were assembled.

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012057 doi:10.1088/1757-899X/604/1/012057

Figure 3. Assembled encoder, heater and thermocouples onto compressor body

2.2 Determination of the piston work

When the measurement is started, the pressure in the cylinder is measured at every A pulse generated by encoder. The encoder has 360 A-pulses per rotation, which means that the pressure is measured at every degree. Since the measurement is started at the first N-pulse, which is calibrated on the top dead centre of piston, for every pressure measurement the position of the piston can be calculated using the following equation:

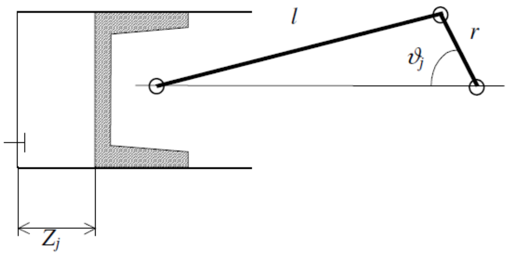

Figure 4. Notations used in equations

$$z_j = r - r\cos\vartheta_j + l\left[1 - \sqrt{(1 - \frac{r}{l}\sin\vartheta_j)^2}\right] \quad [4]$$

$$\vartheta_j = \frac{j}{360}2\pi + \vartheta_0 \quad [5]$$

where

z_j	=	momentary piston position [mm]
r	=	crank length [mm]
l	=	rod length [mm]
ϑ_j	=	momentary angle
ϑ_0	=	angle shift of N – pulse from top dead centre
j	=	number of A – pulses after N - pulse

Notations used in the equations [4] and [5] are also illustrated in Figure 4.

By multiplying the pressure with the piston surface area, the force can be calculated, and a position force diagram can be constructed (Figure 5). By integration of this position force diagram and multiplying this integrated value with the compressor frequency, the piston work can be determined for every revolution.

$$P_{piston} = f \, . \, \oint F dx = \sum_{j=1}^{n} \left(\frac{F_j + F_{j-1}}{2} \right) (x_{j-1} - x_j) \qquad [6]$$

where

F_j = momentary force [N]

x_j = momentary position [m]

f = frequency [1/s]

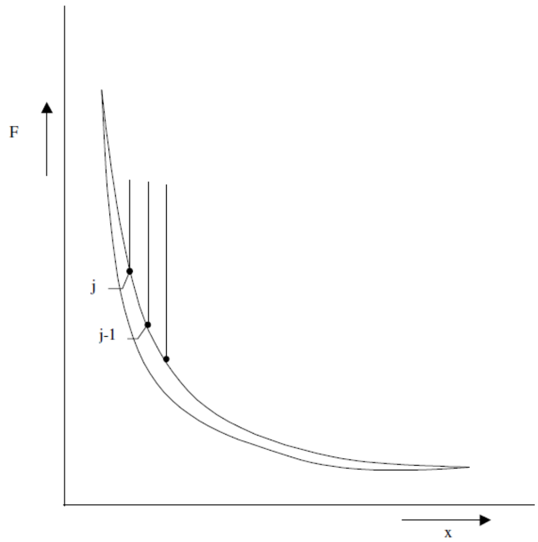

Figure 5. Integration of the position force diagram

3. RESUTLS

The mechanical efficiency of the compressor is directly influenced by the oil viscosity. In the present study, the mineral oil having a viscosity of 5 cSt was used. Since the speed of the compressor on the refrigerator changes with respect to the needed capacity, the oil viscosity depending on the temperature is also changing. Therefore, mechanical losses of compressor change with respect to the viscosity as well as speed of the compressor.

The measurements were carried out under different average load conditions with various operating temperatures. As it is shown in Figure 6, the variation of the average load with rotational speed is negligible. Temperatures were fixed to the target value with accuracy of <0.5 K. During experiments, compressor speed was raised to value of 4500 rpm and gear wheel driving the compressor was disconnected. The discharge pressure was continuously measured while crank shaft slows down with its own moment of inertia. Deacceleration was calculated with crankshaft speed which was measured with optic encoder permanently.

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012057 doi:10.1088/1757-899X/604/1/012057

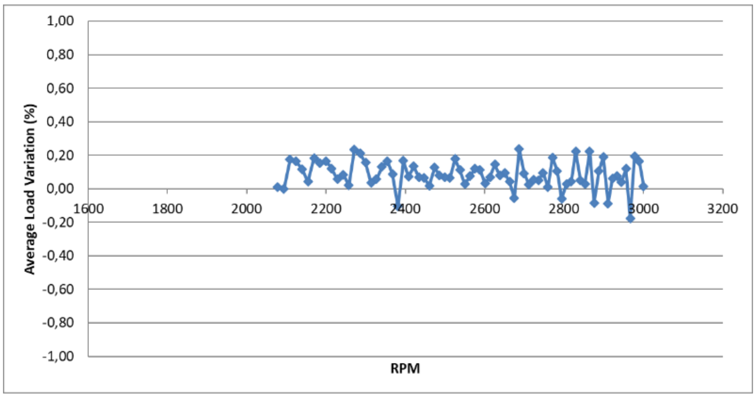

Figure 6. Variation of average load with rotational speed

The mechanical loss values obtained at different oil temperatures are shown in Figure 7. All data acquired were normalized with lowest mechanical loss value. As temperature of the oil was increasing mechanical losses decreased with decreasing oil viscosity. Moreover, lower mechanical losses were measured at low rotational speeds. For a better interpretation, the results obtained are shown with a fitted polynomial for each measurement as well. After regression analysis, calculated R^2 values for 27 °C and 65 °C oil temperatures were 0.98 and 0.95 respectively.

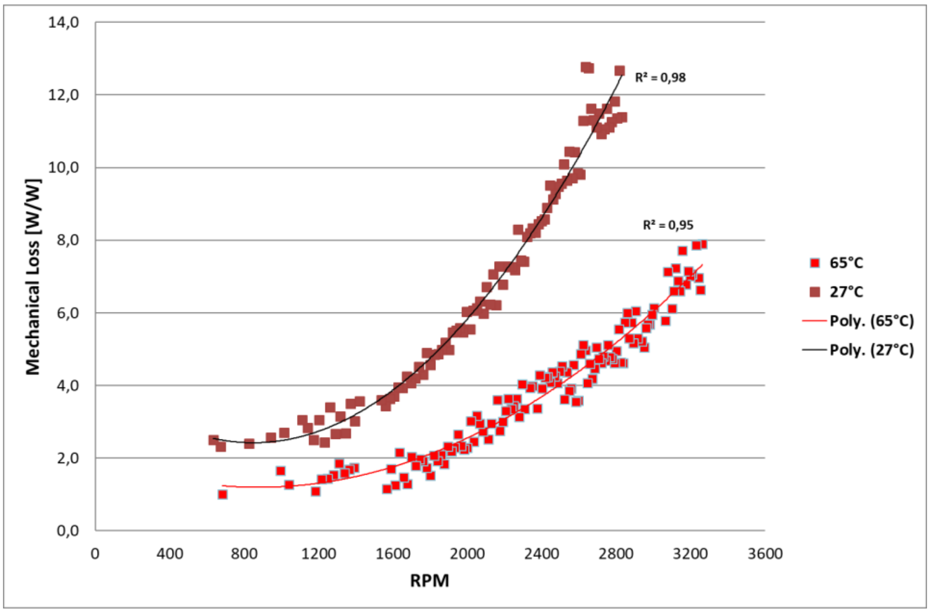

Figure 7. Mechanical losses at various oil temperature

The change of the mechanical losses at various average loads is illustrated in Figure 8. Measurements were conducted at 55N and 61N average load conditions while oil temperature was kept at 57 °C constant. For both cases, the data acquired were normalized with the minimum measured mechanical loss. As the average load was increased from 55 to 61 N very small amount of variation in mechanical losses was observed. The results are here also shown with a fitted polynomial for each measurement as in the investigation of the oil temperature effect on mechanical loss. After regression analysis, R^2 values for both 55 N and 61 N load conditions were calculated 0.97.

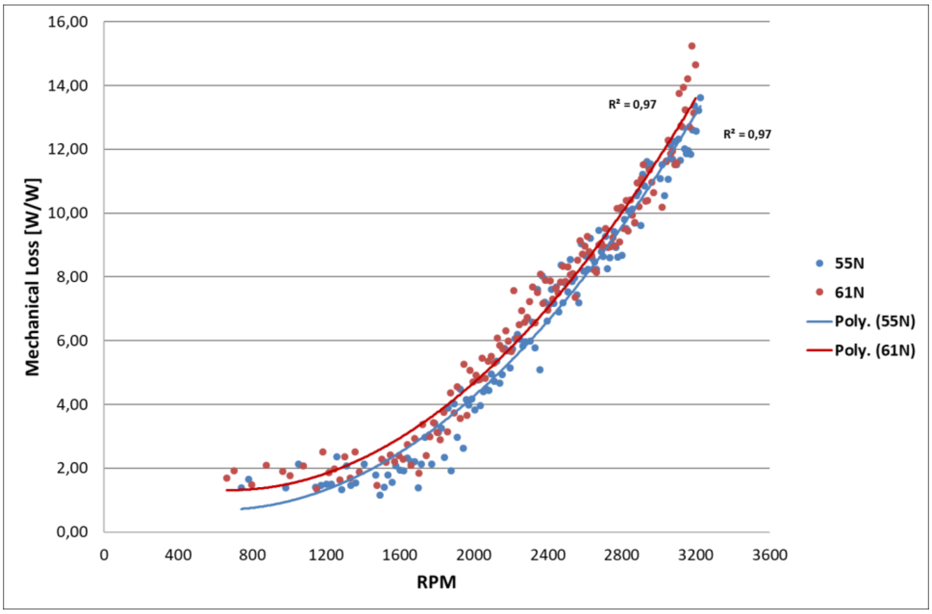

Figure 8. Mechanical losses at various load conditions

The measurements carried out for the investigation of oil temperature and various average load conditions show that mechanical losses decrease with decreasing rotational speed. However, due to logarithmic change of the oil viscosity with respect to temperature, mechanical losses seem more sensitive to the variation of the viscosity in concerned test conditions. Nevertheless, it has been noticed that in both cases, variation rate of the mechanical losses diminishes at lower speeds.

As it is well known that lubrication is the function of the shaft speed, load carrying area of the bearing, applied load, surface roughness and viscosity of the lubricant. Oil film thickness in the bearing gets thinner while the speed decreases. Therefore, the crankshaft rotating inside of bearing approaches to the surface of the bearing at the maximum pressure inside the cylinder while piston reaches to the TDC. Consequently, the asperities on the bearing and the crankshaft surfaces may contact each other which is called mixed lubrication. Even this phenomenon results not only in increasing mechanical loss but also damaging of bearing in long term operation. This kind of transition in the lubrication may also be appearing in our measurements. Therefore, it needs to be clarified by measurements at higher average loads at lower viscosity levels.

4. CONCLUSIONS

In this paper a new approach to mechanical loss measurement system was introduced. Mechanical losses of a variable capacity hermetic reciprocating compressor were measured at various temperature and load conditions. The followings can be concluded:

- In concerned test conditions oil viscosity has higher influence on mechanical losses.
- Mechanical losses decrease with decreasing rotational speed of crankshaft.
- At low speeds the variation rate of the mechanical losses diminishes. This might be the indicator of the changing lubrication regime. In the future work, this phenomenon will be clarified with more detailed higher number of measurements.
- New measurement approach to mechanical losses seems promising to conduct precise and trustworthy measurements. Therefore, it can provide the chance to the more detailed investigation on the bearing design of compressors.

Acknowledgement
The authors would like to thank Re/GenT BV for the building of experimental setup.

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012057 doi:10.1088/1757-899X/604/1/012057

References

[1] B. J. Kim, J. S. Park, et al 2002, Design of Journal Bearings in Reciprocating Compressors, *International Compressor Engineering Conference at Purdue*

[2] A. Okaichi, H. Hasegawa, F. Nishiwaki, 2004, A Study on Lubricaiton Characteristics of Journal and Thrust Bearing in Scroll Compressors", *International Compressor Engineering Conference at Purdue*

[3] K. Sakitani, J. Koiwa, T. Maekawa, 1986, Performance Evaluation of Hermetic Refrigeration Compressor Using Torque Measurement Method, *International Compressor Engineering Conference at Purdue*

[4] M. Matsushima, T. Nomura, N. Nishimura, 2000, Direct Measurement of Hermetic Rotary Compressors Using Strain Gauge, *International Compressor Engineering Conference at Purdue*

[5] H. Sato, T. Itoh, H. Kobayashi, 2004, Frictional Characteristics of Thrust Bearing in Scroll Compressor, *International Compressor Engineering Conference at Purdue*

[6] E. Dietmar, B. Lilie, M. Krueger, 1990, Friction Losses Measurements On a Reciprocating Compressor Mechanism, *International Compressor Engineering Conference at Purdue*

[7] W. Oh, H. Lee, I. Lee, 1994, Measuring Friction Losses Using Accelerometer, *International Compressor Engineering Conference at Purdue*