Novel Radial Compliance Mechanism For A Scroll Compressor

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ABSTRACT

Scroll compressors are widely used popular in HVACR industry due to their efficiency and versality. Radial compliance is one of key mechanisms allowing these compressors to achieve the high level of efficiency. Conventional radial compliance mechanism is typically integrated in the orbit scroll bearing. Because of uncompensated scroll inertia, the conventional mechanism results in higher noise and flank friction at high speed and/or leakage at low speed. There is a wide variety of mechanisms for inertial unloading at high speed, however they introduce extra parts and result in higher scroll overall inertia. A novel compact compliance mechanism requiring minimum additional parts was proposed in the paper. The new mechanism is located within the main bearing. It results in scroll flank force to be almost independent on the compressor speed. The paper describes functionality and design parameters of the mechanism, force analysis, analytical and experimental evaluation of the mechanism functionality and performance.

1. INTRODUCTION

Radial compliancy is a feature of many scroll compressor designs and have been one of the most important key to achieve high compressor efficiency (Hayano *et al*., 1986), (Tojo *et al.*, 1988), (Inaba *et al*., 1986). It introduces a controlled force sliding contact between the flanks of the scrolls by introducing a degree of freedom in a radial direction of the scroll motion and also to develop a mechanism to control flank interference force. It allows for a tighter flank to flank sealing to minimize leakage while tolerating imperfections in scroll geometry, running gear, deflection of components due to gas and inertia forces, as well as thermal growth. Typically, it is done in a form of a drive pin with a flat on the shaft eccentric pin and an unloader bushing located over this pin and being rotatably engaged with an orbiting scroll through a drive bearing. The flank interference force result in a combination of scroll inertia, radial gas force and a fraction of tangential gas force (compression force) to be added (or subtracted) based on the selection of the orientation angle of the flat at the drive pin. If the compressor is operating in a variable speed mode (Perevozchikov *et al*., 2004) and (Elson *et al*., 2006), the inertia component of this flank force would be proportional to a square of rotating speed, providing excessive flank frictional loss at high speed and low flank interference force (resulting in leakage) at low speed. The most common way to counteract this is to introduce a counterweight radially coupled with the bushing so that inertia force from this counterweight would be directly transferred through the drive bearing to the orbiting scroll to counteract the inertia of the latter (Kondo et al., 1993). This bushing coupled with counterweight can be further modified such that the flank load is optimized for high and low speeds (Hahn et al, 2020).

The novel mechanism (Ignatiev et al., 2015), (Ignatiev et al., 2020) which is the subject of the paper utilizes the existing counterweights attached to the compressor shaft, direct coupling between the scroll drive bearing and drive pin, and a radial compliancy mechanism located at the main bearing.

2. DESCRIPTION OF THE MECHANISM

Fig.1. illustrates a schematics of the mechanism. Orbiting scroll is rotatably engaged with the drive pin of the shaft. Due to selection of main and lower counterweight from the compressor balancing perspective, inertia forces and moments from the orbiting scroll, main and lower counterweights counteract each other. If we consider the lower bearing of this mechanism to be a joint and we move radial compliancy mechanism from the drive bearing to the main bearing, radial compliance can be achieved by shaft inclination in the radial direction. In order to provide necessary radial force on the scroll flank, a drive flat is introduced inside main bearing. To prevent excessive shaft tilt during operation and start/stop, an appropriate clearance was selected between the shaft and main bearing, so that during operation it is not limiting the radial compliance travel but at the same time prevents excessive tilt in the lower bearing and motor airgap eccentricity.

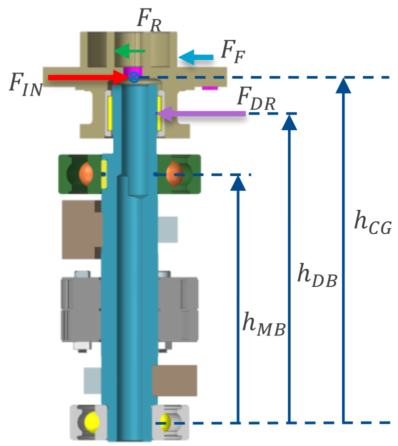


Fig.1. Radial Forces.

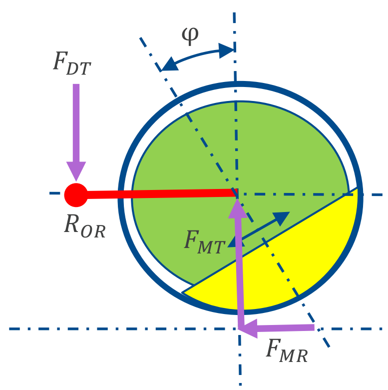


Fig.2. Forces at Main Bearing

Forces on the Main Bearing:

$$F_{MT} = F_T \frac{h_{DB}}{h_{MB}}, \quad F_{MR} = F_T \frac{h_{DB}}{h_{MB}} \mathrm{tg}(\varphi). \tag{1}$$

Forces on the Drive Bearing:

$$F_{DR} = F_T \,\mathrm{tang}(\varphi) - F_{IN} \frac{h_{CG}}{h_{DB}}, \quad F_{DT} = F_T. \tag{2}$$

Force on the scroll flank:

$$F_F = F_{IN}\left(1 - \frac{h_{CG}}{h_{DB}}\right) - F_R + F_T \mathrm{tg}(\varphi). \tag{3}$$

From the equation (3) one can see that the resulting flank force dependency on a scroll inertia force is a very slight and it is actually slightly reducing with the increase of speed which allows for better flank sealing at low speed and reduction of flank friction at high speed. With a proper selection of drive angle, flank force can be kept in the appropriate range within the specified compressor operational envelope.

3. EVALUATION OF THE MECHANISM

3.1. Shaft Motion within the main bearing

The mechanism described above allows for radial compliancy to take place at the main bearing. In order to understand the radial motion of the shaft at the main bearing, an experimental investigation was conducted with proximity probes installed in the compressor just below the main bearing. Together with a timing mark proximity probe used for synchronization of the signal, we were able to plot shaft motion trajectory. Fig.3 illustrates positioning of the probes in the prototype compressor

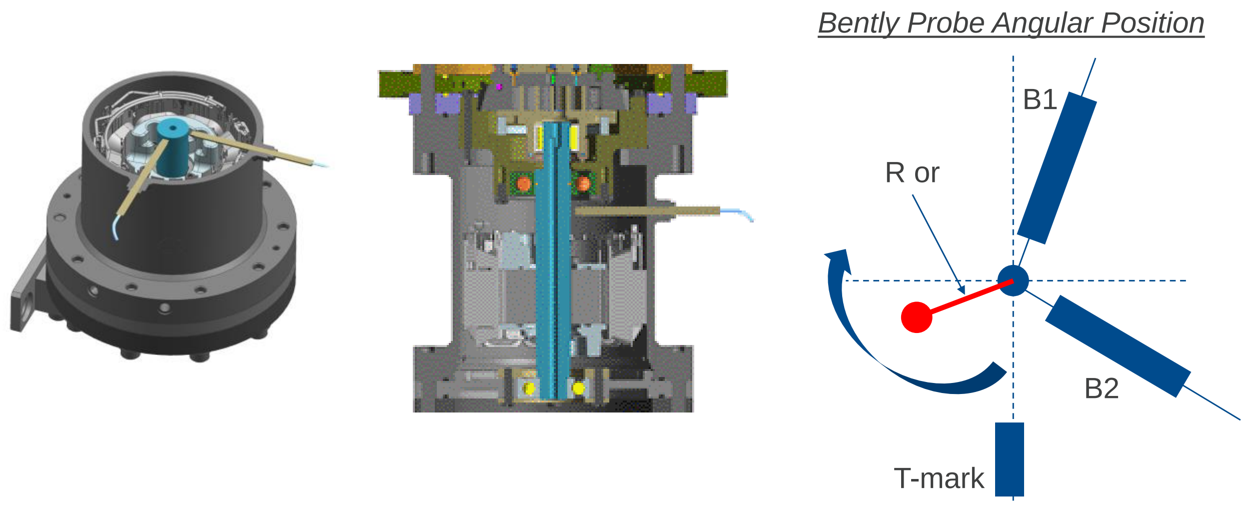


Fig.3. Illustration of the Proximity probe installation in the prototype compressor

3.2. Scroll flank force measurement

In order to understand the flank contact force of the mechanism, a separate experimental test was developed. A prototype fixed scroll was modified by milling out a gap at the outermost flank. This gap was allowed strain gauges to be placed at the base of the outer wrap as seen in Fig.4. As the orbit scroll passes the strain gauge location, a change of strain caused by the bending stress in the flank caused by the flank force interaction, would be captured.

Experimental compressor prototype with the novel radial compliance mechanism, with the fixed scroll equipped with the strain gauges, was tested at different speeds at a selected operating condition. After the series of tests were complete, the experimental compressor mechanism was converted to a conventional radial compliance mechanism, and another series of tests were completed, at the same condition and speeds.

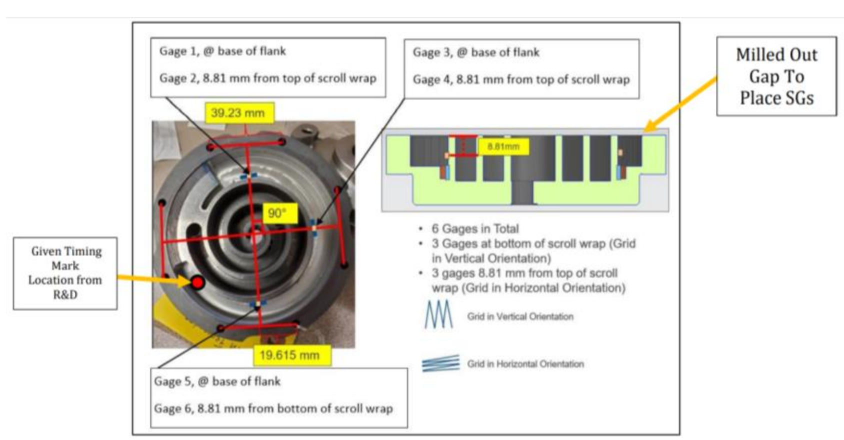


Fig.4. Fixed scroll instrumentation with strain gauges

4. RESULTS

4.1. Scroll flank force analysis

Fig. 5illustrates a diagram of tangential gas force and resulting flank forces at a high pressure ratio operating condition for the novel axial compliance mechanism, compared to a conventional radial compliance mechanism, as well as conventional radial compliance mechanism, for MIN and MAX operating speeds- 2000RPM and 8000 RPM respectively.

Considering tangential gas force being unchanged, it can be clearly seen that, by applying the novel radial compliance mechanism provides slight reduction of flank force while increasing the compressor speed from 2000RPM to 8000RPM (solid lines). On the other hand, conventional radial compliance mechanism exhibits very substantial increase of the flank force (dashed lines).

Fig. 6 illustrates average (within a revolution) forces on drive, main and lower bearings as a function of rotating speed for novel mechanism (solid lines) and conventional radial compliance mechanism (dashed lines). Since the inertia force of the scroll is directly transferred to the shaft and counteracted with shaft counterweights, drive bearing force is increasing with higher speed. At low speeds, drive bearing exhibits some increase in force, since the flank force at this condition is generated by tangential force and drive angle. While increasing eth speed, this component of drive bearing force is getting compensated by inertial components from counterweights. At the same time, main bearing force does not increase with speed, while lower bearing force exhibits some increase with speed.

Compared to the conventional radial compliance, drive bearing force is higher for the novel mechanism, while main and lower bearing forces are lower than the conventional radial compliance.

4.2. Shaft motion at the main bearing.

Fig. 7 illustrates shaft motion trajectories below main bearing, at different speeds- 3600 RPM, 6000 RPM and 7000 RPM at the same operating conditions. These diagrams show the amplitude of motion and also motion along the drive flat. It can be clearly seen that the mechanism is operating, and, most important, that it operates within the selected clearance. This study was used to select correct clearance between the shaft and inner ring of the main bearing, to restrict the motion of the shaft, keeping it within allowable angular tolerance for the lower bearing and air gap within the motor.

Fig.8. illustrates strain gauge signal diagram from strain gauge location 1 (as shown on Fig. 4). The diagrams were recorded at the same operating condition, at two speeds (4600RPM and 6000RPM). The strain gauges installed measure compressive strain at the root of the scroll vane. The diagrams show 3 distinctive sections as illustrated on the Fig.8:

-Horizontal zone, which reflects suction pressure behind the scroll

-declining zone, reflecting the beginning of compression process

-Abrupt negative dip and then quick increase, leading back to the horizontal zone of the next cycle

Negative dip reflects the strain imposed by the passing vane of the orbiting scroll, therefore it indicates the flank contact force.

As it can clearly be seen at the Fig.8, for the case of novel mechanism, there is no substantial difference in strain gauge diagrams for different speeds, while for the case of traditional radial compliance there is a substantial difference in the relative size of the dip reflecting the substantially increased magnitude of the flank contact force.

Similar conclusions can be drawn from the Fig. 9, illustrating strain gauge traces at the location3, with the same operating condition and the same values of speed.

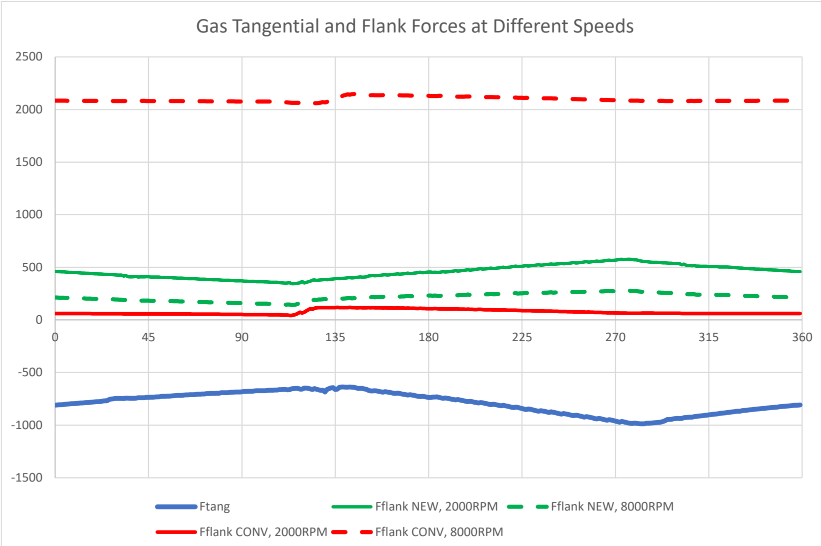


Fig.5. Tangential Gas forces and Flank Forces Vs. crank angle, Radial Compliance and new concept

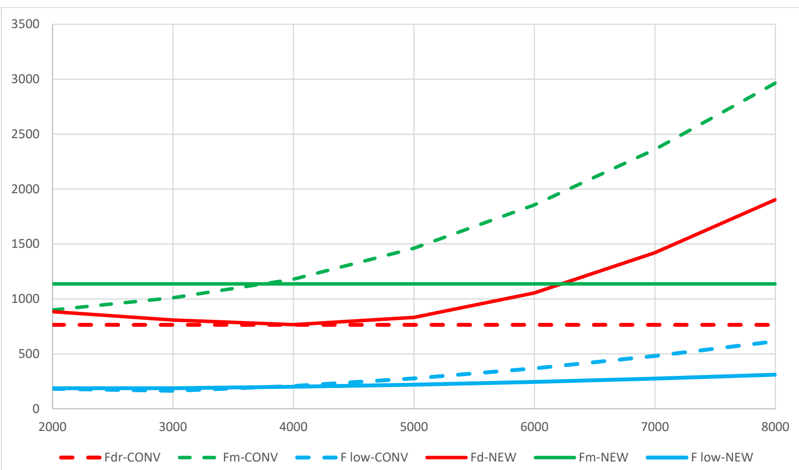


FIG. 6. Average bearing loads Vs. Speed: Conventional Radial Compliance Vs. New Concept

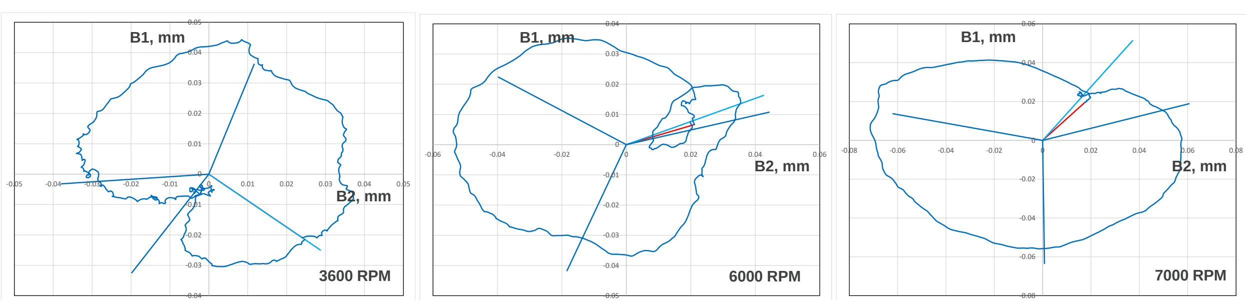


Fig.7. Shaft trajectory at different speeds

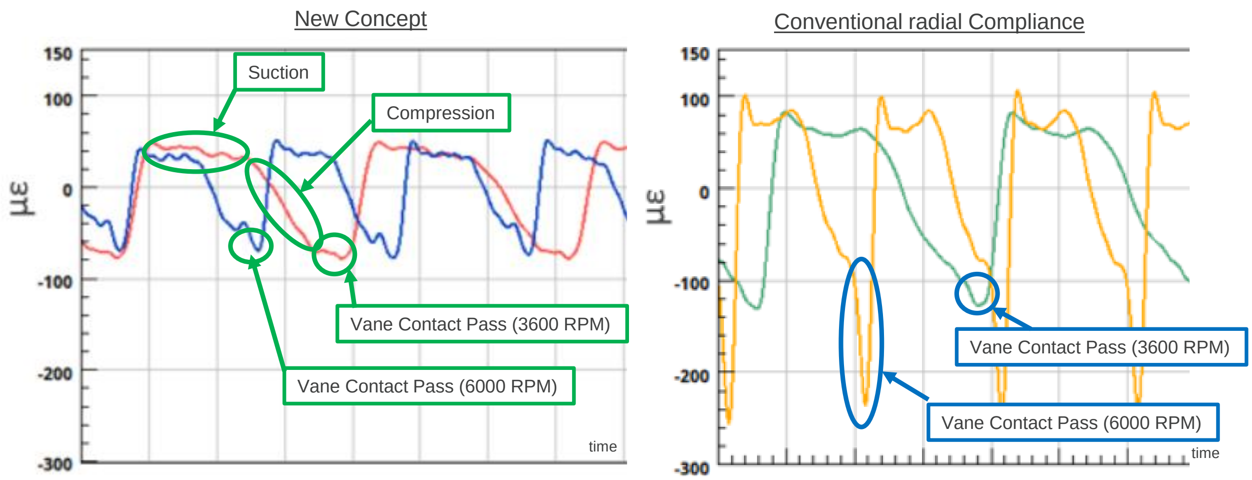


Fig.8. Strain gauge signal at Location 1 at 3600 and 6000 RPM. Conventional radial Compliance Vs. New Concept

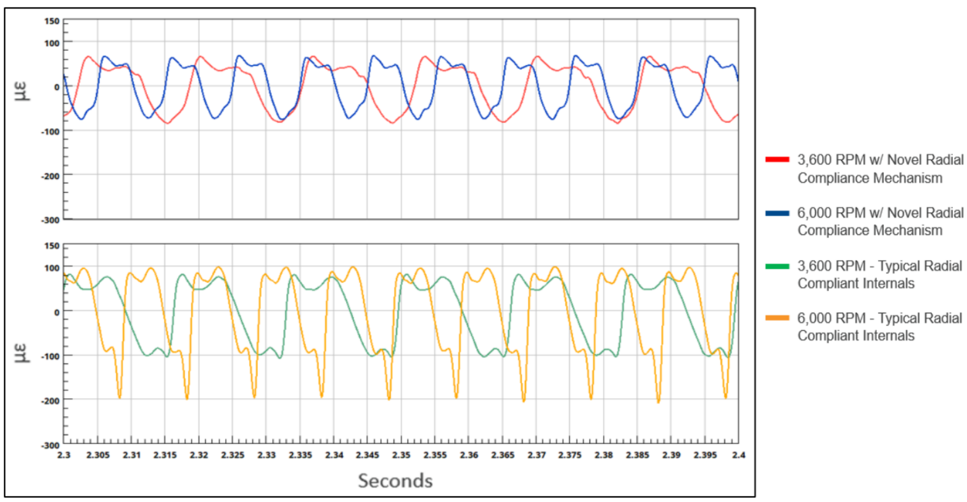
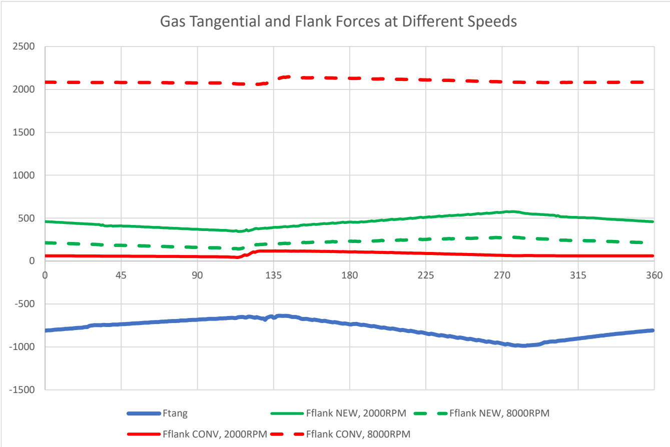


Fig.9. Strain gauge signal at Location 3 at 3600 and 6000 RPM. Conventional radial Compliance Vs. New Concept

5. CONCLUSIONS

1. A novel simple scroll compressor radial compliance mechanism is introduced, allowing to reduce flank contact force and therefore friction at high-speed operation.
2. Experimental investigation of shaft motion was conducted, providing guidance for design of radial compliance mechanism parameters, range of motion and tolerance control.
3. Experimental investigation of flank contact force was conducted in order to verify the intended operation of the mechanism



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6. NOMENCLATURE

F- Force [N]
h- vertical dimension [mm]
φ- angle [deg]

Subscripts:

T – tangential gas (force)
R- radial gas (force)
IN – Inertial (force)
MT- related to Main bearing Tangential (force component)
MR- related to Main bearing Radial (force component)
DT- related to Drive bearing Tangential (force component)
DR- related to Drive bearing Radial (force component)
CG- related to scroll center of gravity (vertical dimension)
DB- related to drive bearing (vertical dimension)
MB- related to main bearing (vertical dimension)

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