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Estimation of the Mechanical Position of Reciprocating Compressor for Silent Stoppage

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ABSTRACT This paper proposes the solution for decreasing of the reciprocating compressor noise and vibrations, which happens at stoppage. These vibrations and resulting noise are caused by the gas pressure in the cylinder, which impacts the compressor's piston and can reverse the motor at stoppage. To avoid this situation the motor must be stopped, when the gas is not pressurized, i.e., at the desired mechanical position, however motor is not equipped with a mechanical position sensor as the control system utilizes an electrical speed and position estimator. As a rule, it is impossible to estimate the mechanical position using an electrical position, however it has been noticed that at low speeds a detrimental feature of reciprocating compressors – varying load torque can help identify the mechanical position. Load torque impacts the quadrature current, making it oscillate at the mechanical frequency of the motor, so this mechanical frequency and position can be estimated using a Phased Locked Loop (PLL) algorithm, applied to the quadrature current.

INDEX TERMS Compressors, frequency estimation, permanent magnet motors, phase locked loops, position estimation, predictive control.

I. INTRODUCTION

On the one hand, a quiet working and living environment is very important to people, however with the rise of the spectrum and variety of available modern day home appliances and the increase in their power, noise pollution is becoming stronger and stronger. Subsequently the number of complaints regarding noise from household electrical appliances is increasing rapidly [1]. Nowadays customers consider sound comfort in their homes to be of importance, so noise levels have become extremely important when selecting a home appliance.

On the other hand, national standards of sound radiation are getting tougher and tougher. For this reasons, manufactures of home appliances pay a lot of attention to the noise emissions of their devices and wage a war on every decibel [2]–[6].

This paper describes the problems and solutions when a previously developed reciprocating compressor drive, which had been successfully put into Mass Production (MP), was found to be faulty. This compressor drive has been used in refrigerators and air conditioners, all of which successfully passed all necessary certifications and had been put into MP. However, when the developed drive was used in a refrigerator, where the compressor is mounted at the top, the device did not pass the acoustic test. Numerous experiments showed that at that moment, when drive stopped, the compressor produced unpleasant noises with a probability of approximately 60% and stopped relatively silently in the other 40%. Typical acoustic profile of compressor operation and stoppage is shown in Fig. 1. Furthermore, due to the top placement of the compressor, its vibrations were amplified by the frame of

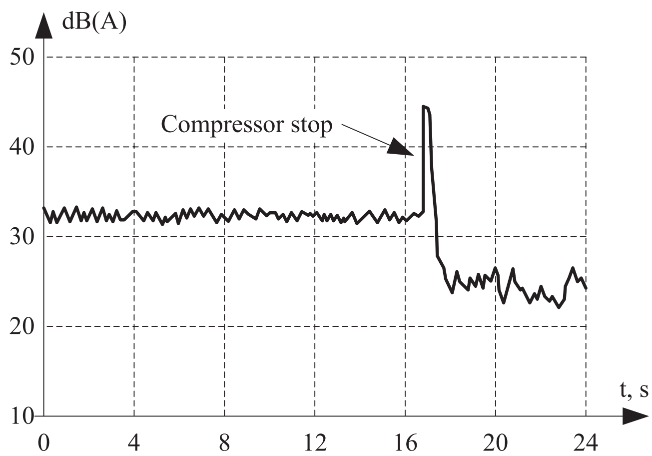

FIGURE 1. Compressor acoustic noise at operation without controlled stoppage.

the refrigerator, resulting in loud and unpleasant noises lasting for 1–2 seconds. Thus, the problem which was not significant for the bottom-mounted compressors, became critical for the top-mounted appliances.

Taking into account the fact that this problem related to the MP drives, all hardware modifications were unwelcome and the target of this research was to improve noise characteristics of the reciprocating compressor drive only by means of software methods without any modification to the developed hardware. Furthermore, due to the high utilization of the microcontroller, the proposed algorithm could not be calculation intensive and also could not use a lot of memory.

In this work we studied methods used for decreasing of compressor noises, and tried to use them in our case. However results were unsatisfactory and we had to go furhter and to find root cause of the noises and method for improvement. These results are discussed in Section II.

Our detailed study showed, that the noises occurred at stoppage of compressors were caused by the gas pressured in the cylinder, which impacted the compressor's piston and reversed the motor at stoppage. Therefore, the potential energy of the pressurized gas turned into kinetic energy of the piston, and then into the counter torque. This torque was applied to the stator of motor, and then transferred through the compressor shell and mounting points to the frame of refrigerator, which produced unpleasant noise.

After that, we studied the cases, when compressor stopped silently, and found that reversed rotation did not happen, if the stoppage was started at the certain range of piston positions. Therefore, the unpleasant noises at the stoppage of compressor can be eliminated if the stoppage started at the proper position of the piston. However, for this purpose, we need information on the mechanical position of the motor, which typically can be obtained from any position encoder. At the same time, electric motors of modern compressors are not equipped with position encoders and use sensorless position estimation techniques, which estimate only electrical position of the rotor. If motor has more than one pole pair, mechanical position cannot be obtained from electrical position because electrical position in terms of mechanical degrees is: (360 mod p), where p is number of pole pairs. Thuswise, special algorithm for estimation of the mechanical position must be involved, and problem of silent stoppage is decomposed into two tasks: estimation of the mechanical position of the rotor and definition of the best position to start stoppage.

Unfortunately tracking of the mechanical position was and is not a popular problem and merely a few papers describe the topic. Authors in [7]–[11] proposed monitoring of the load torque for better control of the commutation points. Papers [12], [13] suggest performing tracking of the load, decreasing the energy consumption of reciprocating compressors. They utilized the electrical position in order to calculate mechanical position and used the quadrature current to find phase shift between them. Authors in [14] enhanced a conventional system with an accelerometer in order to detect load variation and track load torque, however it was not suitable for proposed task since it needed additional hardware. Paper [15] proposes design modifications of the IPMSM rotor, which makes detection of the mechanical angle possible. Authors of [16] paid attention to the reduction of the speed ripple due to load variation over mechanical revolution and suggested utilizing Proportional-Resonant (PR) controller to regulate i_q. There are two US patents [17] and [18], which track maximum of the pressure in cylinder to detect starting of discharge phase in compressor operation. This information is used in the feedback to control the motor. However those reciprocating compressors are equipped with linear motors, and these ideas and control schemes are not suitable for our problem.

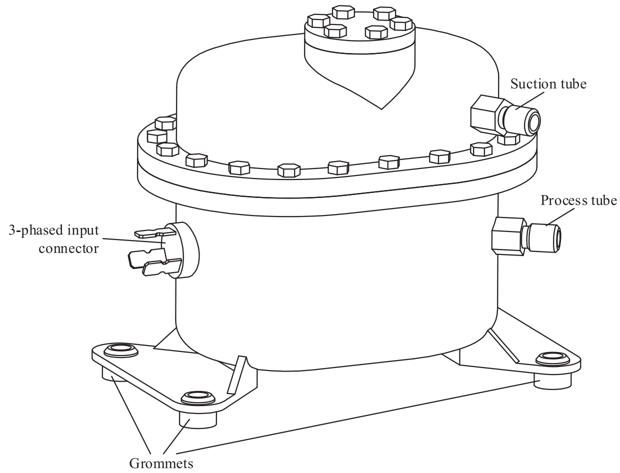

FIGURE 2. Reciprocating compressor.

After a detailed analysis of the published researches it was concluded that they could not be utilized for resource saving and time effective tracking of mechanical position, so new algorithm had to be developed.

II. EXISTING METHODS

When starting to solve this problem, previous researches and conventional approach were examined and analyzed.

A. MECHANICAL MODIFICATION

The easiest way to solve noise and vibration problem is minor modification of mechanical part, i.e., changing of dampers to shift resonant frequencies of the whole mechanical system. Typically, refrigerator compressor is mounted on the rubber grommets, Fig. 2, which partly absorb energy of vibrations and decrease vibrations transferred to the frame of refrigerator.

We tried to use different grommets, which differ by the shape and material, but the results were no good. Typical acoustic profile obtained for grommets of the same shape as used before, but softer material is shown in Fig. 3. It is clearly seen that noise peak at the stop of the compressor decreased, but did not disappeared completely. At the same time peak length increased, which is undesired. Therefore, this approach

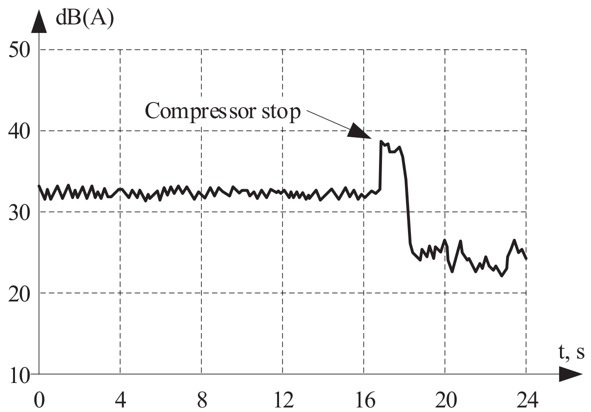

FIGURE 3. **Compressor acoustic noise with modified grommets.**

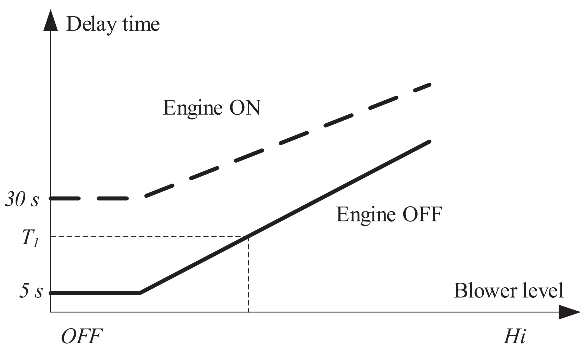

FIGURE 4. **Calculation of the delay time.**

could not solve the problem, so we had to try other methods to eliminate undesired vibrations.

B. SPEED PROFILING AT BRAKING

Engineers of one of Japanese corporations faced with the similar problem, when developing vehicle air conditioner. They proposed solution to decrease unpleasant compressor noise at stoppage, which is described in two US patents [8] and [9]. Authors of those patents proposed not to stop compressor immediately after receiving command to stop. They suggested to decrease compressor operational speed smoothly and only after that to turn off the motor. They claimed several similar methods to create speed profile at stoppage, and the most interesting of them will be discussed here.

In the first method, when command to stop is received, a delay time T_1 is calculated from the map stored in the microcontroller as shown in the Fig. 4. This map represents the relationship between the blower (a fan cooling evaporator) level and the delay time T_1, This delay time may also be varied depending on whether the engine is on or off. After that the inverter decreases compressor speed at the maximum rate of change and turns it off, when time T_1, passes, Fig. 5.

In the second method authors proposed to use compressor stop pressure difference ΔP_0 (between suction and discharge pressure) as a criterion to stop the motor of compressor. When

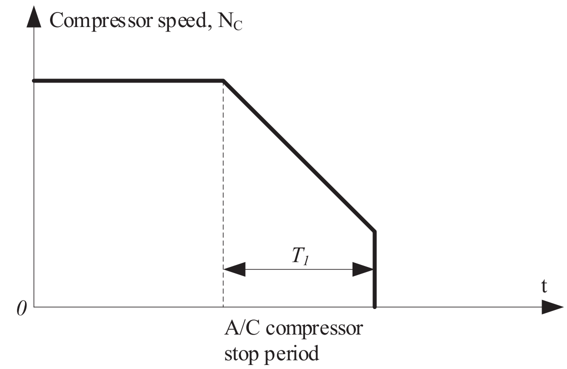

FIGURE 5. **Speed profile at stoppage.**

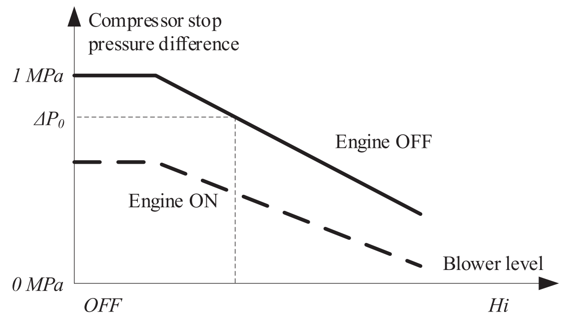

FIGURE 6. **Calculation of compressor stop pressure difference.**

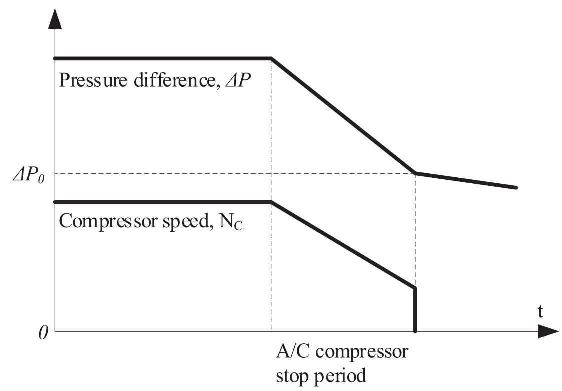

FIGURE 7. **Speed profile at stoppage.**

command to stop is received, a stop pressure difference ΔP_0 is calculated from the map stored in the microcontroller as shown in the Fig. 6. This map represents the relationship between the blower level and the stop pressure difference ΔP_0, and may also be varied depending on whether the engine is on or off. After that the inverter decreases compressor speed at the maximum rate of change and turns it off, when stop pressure difference ΔP_0 is reached, Fig. 7.

Third approach suggests using of compressor stopping speed of operation as a criterion to turn it off. When command to stop is received, a compressor stopping speed of operation N_{C0} is calculated from the map stored in the microcontroller

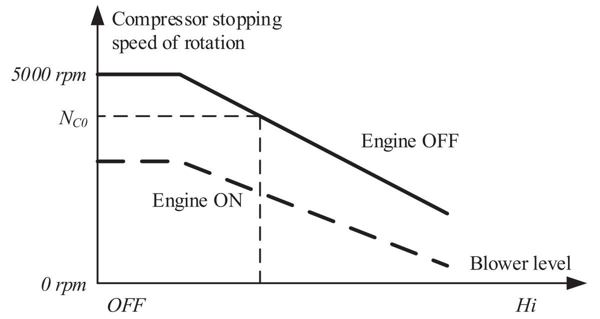

FIGURE 8. **Calculation of compressor stopping speed of rotation.**

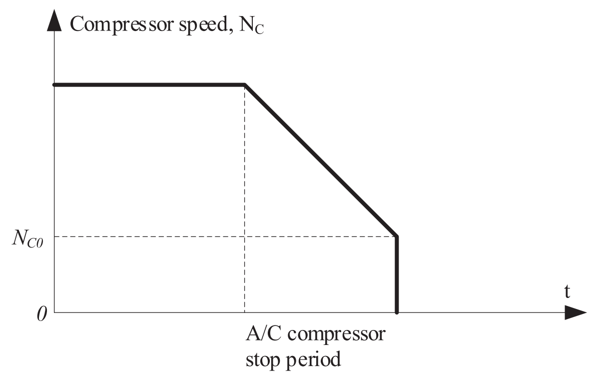

FIGURE 9. **Speed profile at stoppage.**

as shown in the Fig. 8. This map represents the relationship between the blower level and the compressor stopping speed of operation N_{C0}, and may also be varied depending on whether the engine is on or off. After that the inverter decreases compressor speed at the maximum rate of change and turns it off, when its speed reaches compressor stopping speed of operation N_{C0}, Fig. 9.

Unfortunately, authors claimed in the patents only general idea without any details. In the first method, they propose braking with maximum possible deceleration, however it is not clear how to handle the case, when delay time T_1 is long enough and zero speed can be reached before time passes. They also do not explain how to make dependence of delay time on the blower level, which is key component of this algorithm. We implemented this algorithm and used different values of T_1, but results were unsatisfactory for all values.

Same problems are present in the second method. The method of obtaining the dependence of the compressor stop pressure difference ΔP_0 on the blower level is incomprehensible. Furthermore using pressure as a control variable needs additional pressure sensor, which increase price and decreases reliability of the total system. At the same time, the dependence of pressure difference on compressor speed is not described, and authors do not explain how to handle the case, when maximum speed deceleration rate is high enough and zero speed can be reached before pressure difference decreases to target value ΔP_0.

Third method seems to be the most practicable, since proposed speed profile is easy to implement and proposed algorithm does not need additional sensors. However it is still unclear how to specify dependence of the compressor stopping speed of rotation on the blower level. Compressor noise at stopping depends on the pressure difference and authors do not explain how to connect in with compressor speed. This dependence is not simple, especially if dynamic modes of operation are considered.

Authors of [8] and [9] also claimed that moving vehicle produces noises, which level depends on the speed and engine operation and that produced noises grow up with the speed increase. So compressor noises may not be eliminated completely, they just must be decreased to be lower than vehicle noises. It is useful note, which simplifies design of vehicle compressor control, but cannot be used in living appliances.

III. PRINCIPLE OF OPERATION OF RECIPROCATING COMPRESSOR

After target definition the author studied detailed reason for the noise emitted at stopping, which revealed to be closely connected the nature of the reciprocating compressor. Before any further explanation, the principle of operation of the reciprocating compressor must be explained.

The structure of the reciprocating compressor is shown in Fig. 10. It consists of a piston moving inside the cylinder by means of the connecting rod and crankshaft connected to the motor. Such mechanical system converts the rotational motion of the motor into the reciprocating motion of the piston. The cylinder is equipped with two valves: a suction valve for drawing refrigerant gas into cylinder and a discharge valve for delivery of the pressed gas.

The different stages of compressor operation are shown in Fig. 10. Let us suppose that initially the piston is at the top position of cylinder, which is called Top Dead Center (TDC). At this position, refrigerant gas has already been pressed and delivered via the discharge valve, Fig. 10, Stage I. Discharge valve is opened, pressure in cylinder P_C is greater than discharge pressure P_D. From this position, piston starts moving to the bottom of cylinder, P_C decreases, and discharge valve closes. However the pressure in the cylinder is higher than the suction pressure P_S, therefore the suction valve is still closed, Fig. 10, Stage II. When the pressure in the cylinder becomes less than P_S, the suction valve opens and refrigerant from the suction pipe is deposited inside the cylinder, Fig. 10, Stage III. The piston continues its movement until it reaches the bottom of the cylinder, which is called Bottom Dead Center (BDC), and refrigerant continues being deposited into the cylinder via the suction valve, Fig. 10, Stage IV. After reaching BDC, the piston starts moving to the opposite direction compressing gas in the cylinder. Pressure in the cylinder increases, the suction valve closes, however P_C is less than P_D, therefore the discharge valve is still closed, Fig. 10, Stage V. When the pressure inside the cylinder reaches more than the discharge pressure, the discharge valve opens, delivering compressed

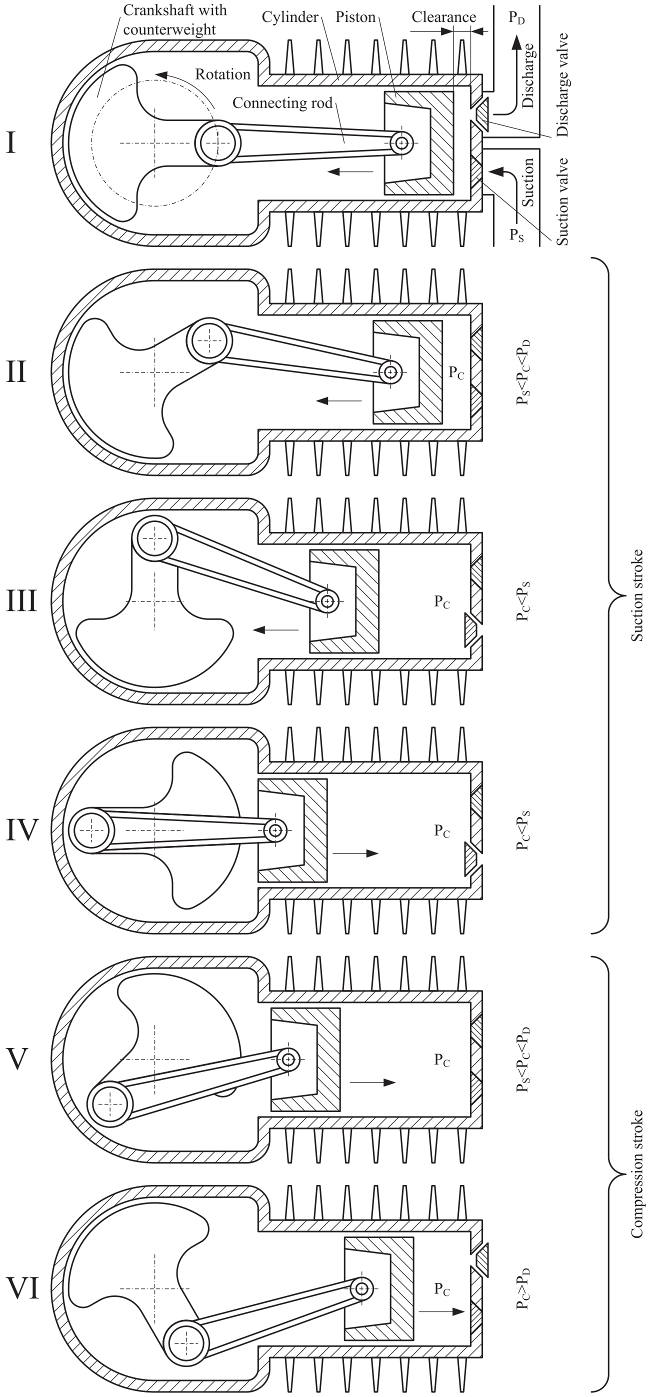

FIGURE 10. **Stages of the operation of reciprocating compressor.**

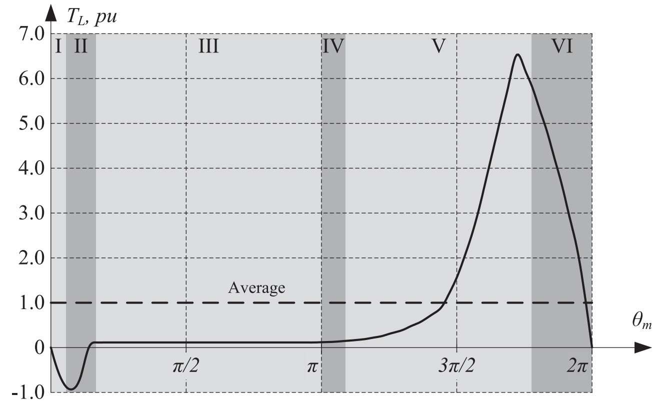

FIGURE 11. **Reciprocating compressor load torque.**

gas to the discharge line, Fig. 10, Stage VI. Thereafter, the piston reaches TDC, where compressed gas inside the cylinder is ejected and next cycle starts. Each revolution of the crankshaft produces one suction and one discharge stroke, as described in stages I–VI.

When the piston reaches TDC position it does not come in contact with the cylinder, thus there is space the between piston and top of the cylinder, which is called clearance. The volume of the cylinder corresponding to the clearance is called clearance volume of the compressor. This volume decreases the performance of the compressor as some portions of the pressured gas are not discharged; and because gas in the clearance volume expands at the suction stroke, less gas can be injected. However, clearance is required to keep the piston from striking the top of cylinder and for the normal operation of suction and discharge valves. The typical clearance volume is 4–30%.

It should be noted that the direction of motor rotation is not important for gas pressuring, but it impacts on the proper operation of the oil lift – a helix channel carved on the shaft and designed to supply oil from the bottom of compressor to moving parts at the top.

The load torque of the reciprocating compressor is shown in the Fig. 11. It is drawn in per–units, where the base unit corresponds with the average torque over revolution. Small negative peak corresponds to the expansion of the pressed gas in the clearance volume and large positive peak indicates the compression of the gas in the cylinder. The values of these peaks strongly depend on the compressor design, but generally negative peak is (0.5–1) p.u. and positive peak is (5–8) p.u.

IV. CONTROL SYSTEM

The control system used to operate the compressor is similar to the control systems utilized in other appliances [19]–[25] and is shown in Fig. 12. It drives the Interior Permanent Magnet Synchronous Motor (IPMSM) with sensorless control, which has almost become a standard in compressor applications. The performance of the sensorless estimation algorithm was verified at the dynamo test jig, and it showed that estimation algorithm perfectly operates at the speeds over 10 Hz with maximum error less than five electrical degrees. The motor of the compressor is driven by a three-phase inverter, which is based on the "ST Microelectronics" STGIPN3H60 (3 A/600 V) small low-loss intelligent molded module, which contains six MOSFETs and gating circuits. It was designed to be supplied from a standard 220 V (50 – 60) Hz source. The

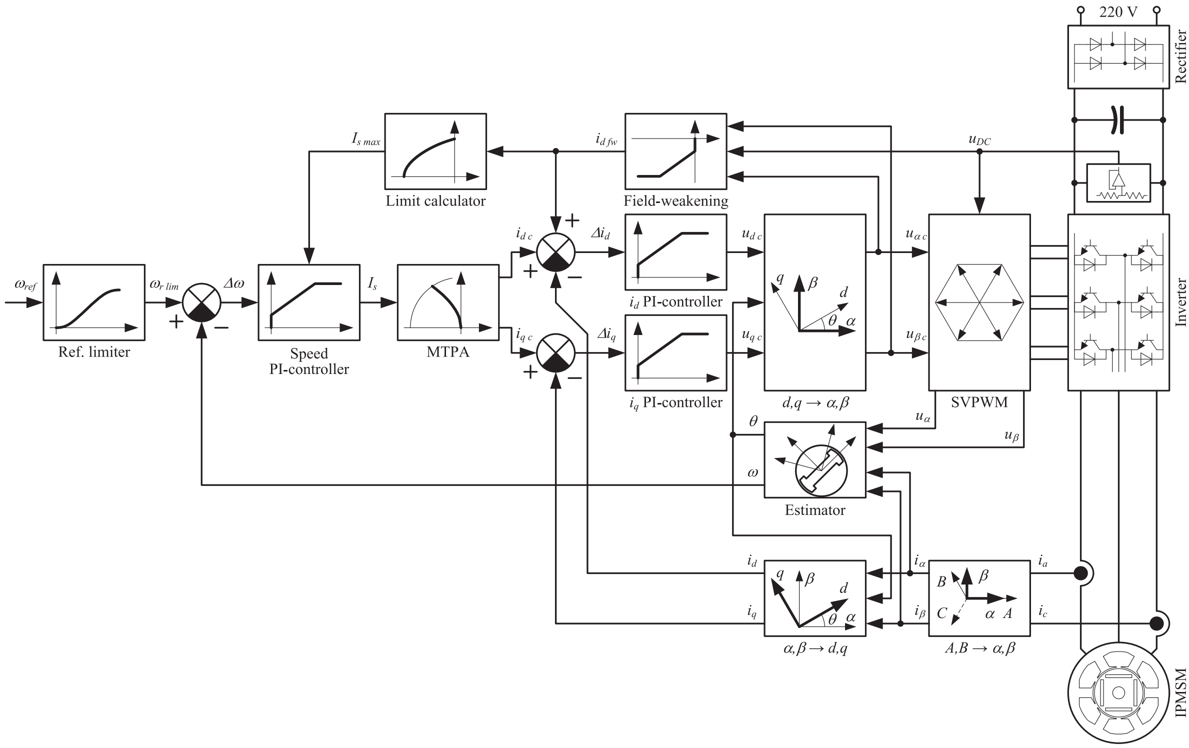

FIGURE 12. Control scheme of the reciprocating compressor drive for refrigerator.

control system is built on the base of a 60 MIPS Cortex-M3 microcontroller, which controls the inverter with 4 kHz PWM. Electrical signals are sensed by the DC-link voltage sensor and two current sensors [23], which are converted by 12-bit ADC of microcontroller with a sampling time of 250 μs.

The compressor drive uses open-loop starting and acceleration with instantaneous closing, which was described in detail in [24]–[27]. Due to the mechanical requirements of the compressor, it operates in oil pumping mode straight after starting, where the motor accelerates to 1800 rpm and runs at that speed for 20 seconds. This time is enough to supply and spread oil from the bottom of the compressor to moving parts at the top.

Control system used for this drive is conventional sensorless vector control with some modifications. It has an outer speed loop and two inner current loops in *dq* frame, where speed for the feedback and electrical position for phase transformations are provided by the sensorless estimator.

The control system contains a field-weakening controller used to increase the maximum speed up to 30% of the rated speed by weakening the rotor magnetic field with direct axis current i_d.

Motor drive also involves Maximum Torque Per Ampere (MTPA) block for the full utilization of the motor potential. This block is fed with stator current from the speed controller and decomposes it into direct and quadrature currents to provide highest possible torque, which is sum of active torque due to the rotor's field and reactive torque due to the magnetic asymmetry along direct and quadrature axes.

The received commanded speed is passed through the S-shaped reference limiter, which limits the maximum acceleration and deceleration at 1 Hz/s^2.

For better stability of the control algorithms and better utilization of input voltage cut-off frequencies of speed and current PI-controllers were selected as low as possible and dynamically vary with the change of operational speed. For this reason, motor currents and torque rise only twice at the point of maximum load, whereas load torque increases approximately seven times, Fig. 11. The price of such adjustment is motor speed variation over mechanical revolution up to 50 rpm, which is not critical for the compressor operation.

V. PROPOSED ALGORITHM

As it is seen from the review of previous arts, there are not so many ideas in this area. Several proposed methods are not suitable for the refrigerator compressor so new method must be proposed.

After detailed analysis of the problem author concluded that in the developed system braking started in the moment of receiving of command to stop, which happened in the random position of the compressor motor. Moreover, if 40% of braking were successful, there was a specific position or sector, where if braking was initiated it produced no noise. Therefore, the problem could be solved by estimation of mechanical position and definition of silent position to start braking.

This task is simple, when the drive is equipped with a position encoder, but could be a problem for sensorless system.

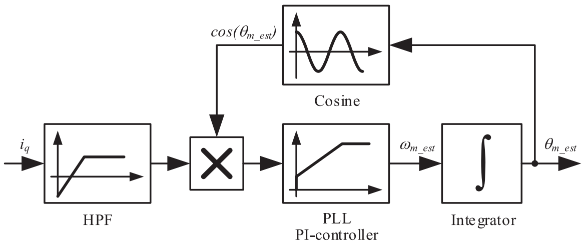

FIGURE 13. PLL structure.

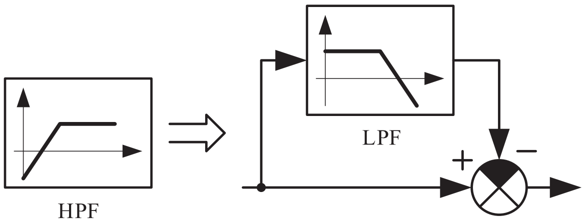

FIGURE 14. HPF implementation.

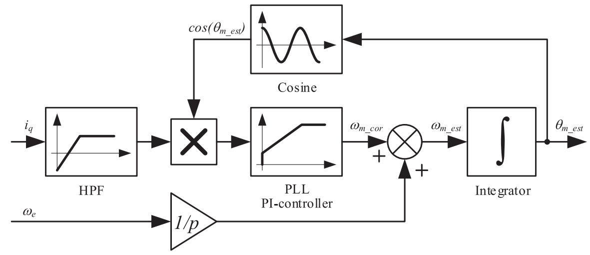

FIGURE 15. Modified PLL structure.

TABLE I Experimental Motor Parameters

Parameter	Value
Pole pairs, p	3
Stator resistance, R_s [Ω]	7.2
Direct axis inductance, L_d [mH]	77
Quadrature axis inductance, L_q [mH]	117
Flux linkage, ψ [V/rad/s]	0.143
Rated power, P_{rated} [W]	200

Position estimators used for the control estimate only electrical position, which is not enough, so additional estimation technique must be developed.

When testing reciprocating compressors author noticed that phase current contains oscillation with a mechanical frequency ω_m, which is p times less than electrical frequency, where p is number of motor pole pairs. The amplitude of the current oscillation increases at lower speed, and becomes almost negligible at higher speed. Therefore, this harmonic of the current can be filtered and used for the estimation of mechanical position.

For this purpose, author proposed to utilize simple Phase Locked Loop (PLL) controller shown in Fig. 13. It was designed to detect the frequency and phase of the quadrature current, of which the type was expected to be:

$$A(t) = A_0 + A_m \sin(\omega_m t), \tag{1}$$

where A_0 is direct component of the signal, A_m is amplitude of the harmonic at ω_m.

The PLL controller contains a High-Pass Filter (HPF) for removing the direct component, which was implemented utilizing first order Low-Pass Filter (LPF) as shown in the Fig. 14. As LPF was already implemented and utilized in the control system, HPF implementation was simple and did not need additional resources. The cut-off frequency of the filter can be selected as mechanical frequency ω_m.

Taking into account the fact that mechanical frequency of the motor could be obtained using electrical frequency, it was advantageous to involve this signal and increase stability of the PLL. After this modification, PLL controller estimated only the speed variation around its average value, so controller gains could be decreased making it more stable. The Modified PLL controller is shown in Fig. 15.

The design of the PLL PI controller can be made according to [28] and [29] for the systems where signal frequency is known and only phase must be estimated.

As earlier stated, the cut-off frequency of the speed controller was selected as low as possible, so that the motor current does not vary significantly with load torque variations. Furthermore at higher speeds, the mechanical system has a higher kinetic energy, which is proportional to the squared speed. It helps the compressor to pass the point of maximum load with lower speed reduction, so the quadrature current variation over mechanical revolution decreases, which makes it difficult to detect its oscillation at the mechanical frequency. For this reason, when command to stop is received, motor speed must be decreased to the minimum, where quadrature current oscillation becomes more pronounced, and after that PLL algorithm can be started.

VI. EXPERIMENTAL RESULTS

Feasibility of the proposed method has been verified utilizing a standard compressor with IPMSM, which parameters are given in the Table I. The inverter board used to control the compressor, remained the same as before, but its software had to be enhanced with the developed compressor stopping algorithm, based on the proposed PLL scheme. The test jig used for the algorithm development and verification is shown in Fig. 16.

As stated earlier, after receiving the command to stop, motor speed must be decreased to the value, where the quadrature current harmonic due to load variation can be easily detected. This speed is compromise between amplitude of harmonic and stable operation of the compressor. Our experiments showed that for used equipment 1200 rpm is the best choice. Fig. 17 demonstrates motor phase current at 1200 rpm and it can be seen that current contains significant harmonic at the frequency trice less than current frequency. This harmonic

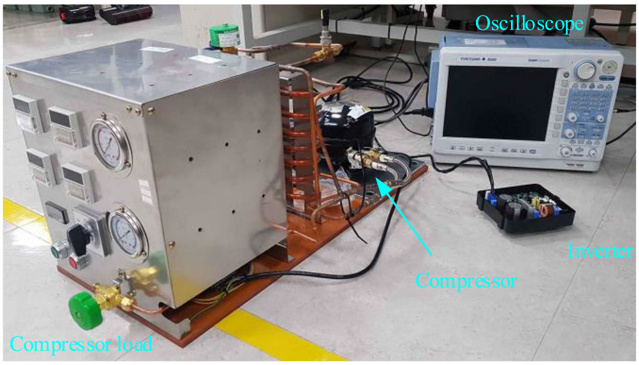

FIGURE 16. Experimental set-up.

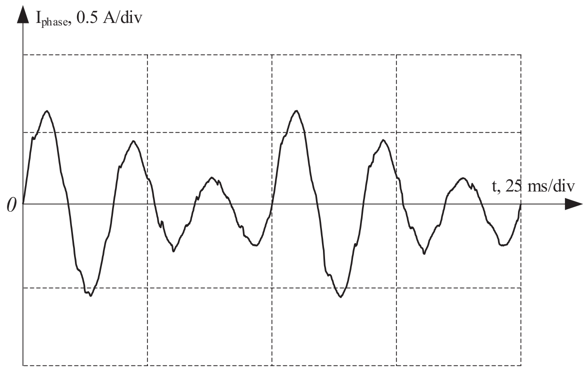

FIGURE 17. Phase current at 1200 rpm.

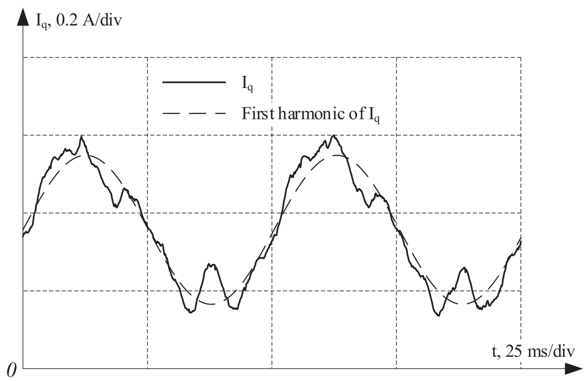

FIGURE 18. Quadrature current and its first harmonic at 1200 rpm.

contains information on the mechanical position of the rotor, so it can be used for estimation. Fig. 18 shows quadrature current with its harmonic at mechanical frequency. It can be clearly seen that current variation due to the load is strongly pronounced and easily detectable.

Thereafter, when stoppage was started, the speed was initially decreased to 1200 rpm, and motor continued operation at this speed. When the motor reached 1200 rpm, PLL algorithm was executed, using i_q as input signal and the motor continued operating at that speed for two seconds. During

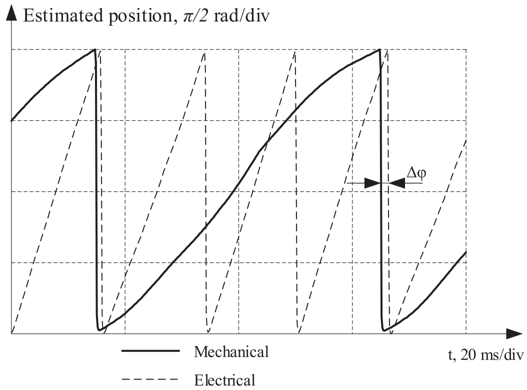

FIGURE 19. Estimated mechanical and electrical positions at 1200 rpm.

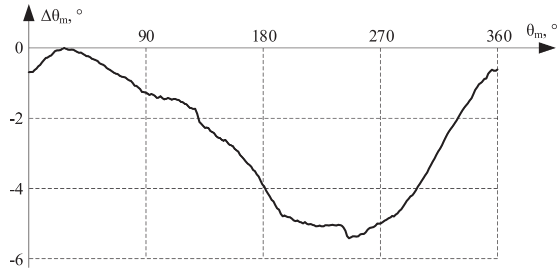

FIGURE 20. Error of the estimated mechanical position.

this time the PLL algorithm stabilized its operation, detected mechanical frequency and position of the rotor and piston.

Fig. 19 shows electrical angle provided by the estimator and mechanical position estimated by the PLL. It can be seen that the electrical position was changing three times faster than mechanical position (since motor has three pole pairs) and that mechanical position had angle shift of $\Delta\varphi$. This appeared because mechanical position was estimated by the load harmonic of quadrature current, which was shifted due to the dynamic torque. This angle difference was about 10° at maximum and could be easily corrected, if we needed to ascertain the exact mechanical position. However, it was not necessary in our application as we only needed to know the point at which the motor had to start braking. Error of the estimated mechanical position is depicted in Fig. 20.

The next step was detection of the optimal mechanical position to stop. This experiment was divided into two parts for speeding-up the test and decreasing of usage time of the silent chamber for acoustic measurements. In the first step, we roughly evaluated sectors with low vibrations at stoppage and at the second step, we thorough measured vibrations in the selected areas.

Firstly, we stopped the compressor at different positions with the step of 30 degrees, roughly evaluating quality of the braking using fingers touching the compressor and feeling vibrations. It allowed to detect strong vibrations at stoppage and exclude such cases from the detailed study.

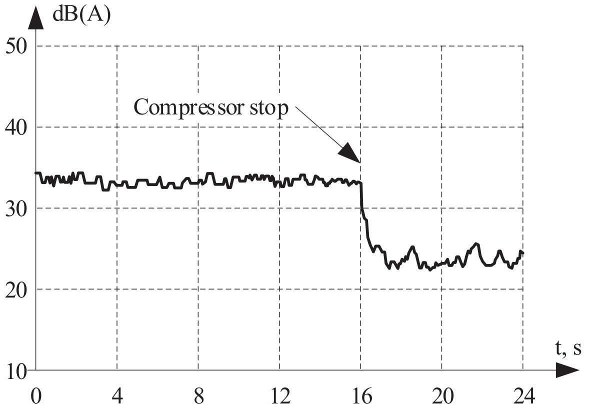

FIGURE 21. Compressor acoustic noise at operation with controlled stop.

The test was performed at 25%, 50%, 75% and 100% of the rated load, but results of these attempts were the same. Three positions of 0°, 30° and 60° were relatively silent, so position range of [−15°..75°] was planned for the detailed exploration using special equipment.

In the second part of the test, the selected sector was checked with the step of 5 degrees and the acoustic noises were measured in the special silent acoustic chamber. The target of this experiment was to find the most silent point to start braking. This test was also performed at 25%, 50%, 75% and 100% of the rated load, but no significant difference in acoustic profiles were detected. The best noise level was detected, when the compressor braking was started at the 35 degrees position, thus this value was used for the final setting.

For check of the performance of the tuned control algorithm a series of ten experiments was carried out in four load modes: 25%, 50%, 75% and 100% of the rated load. In each experiment compressor operated at the selected load during five minutes, which was enough for the pressure stabilization, and then stopped. All experiments showed that noise peak at the moment of compressor stoppage was eliminated and algorithm operated as expected. Improved acoustic noise profile of compressor operation and its stop at the rated load is shown in Fig. 21, where it can be clearly seen that noise peak at stopping is not present and the compressor is stopped as desired.

As mentioned earlier, PI-controllers have decreased gains, so the control system is not very sensitive to load variations, thus resulting in decreased quadrature current harmonic at the mechanical frequency, so PLL algorithm cannot detect it at low loads. Experiments showed that if a load decreased to 15% of the rated load, PLL algorithm failed and compressor cannot be stopped at the desired position. However, at low loads, it was not a problem as the gas in the cylinder was not under high pressure and could not reverse the piston producing noise and vibrations.

The author understands that a favorable result on the mechanical position estimation could be achieved using electrical position for angle calculation and PLL output for the correction of its phase. Such algorithm can be used in control schemes, which need precise mechanical position and which are not limited with resources. However, the main target of our research was to develop algorithm, which is not calculation intensive and can be executed at the currently using inverter board. Therefore, for the purpose of algorithm simplification we excluded unnecessary elements and decreased precision.

VII. CONCLUSION

This paper describes noise and vibration problem happened at the stoppage of reciprocating compressor, when pressurized in the cylinder gas reverses the piston. Author analyzes reason of the problem, explains operation of the reciprocating compressor, shows utilized software and hardware and describes known methods to solve the problem. Then he proposes solution for the noises elimination, which does not need additional hardware, and is implemented as additional software algorithm executed, when command to stop is received.

The experimental results discussed in the paper demonstrate perfect operation of the proposed method and complete elimination of the noise and vibration, which happened at the stoppage of reciprocating compressor.

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