1990

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EXPERIMENTAL ANALYSIS OF A WATER-TO-WATER HEAT PUMP WITH VARIABLE SPEED SCROLL COMPRESSOR

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ABSTRACT

As part of a broader international cooperation variable speed scroll compressors have been tested in a specially designed water-to-water heat pump test rigg at TECNARS in Bari, Italy. Experimental results for the compressor performance and efficiency are being compared to catalog data of single speed reciprocating compressors. A large number of tests were run in order to investigate effects of different voltage to frequency characteristics in the frequency inverter output. Results show a considerable impact of a properly selected supply voltage on the compressor efficiency in particular at low frequencies. Energy conservation effects when using variable speed scroll compressors in place of single speed reciprocating compressors for space heating and air conditioning applications are briefly discussed.

Introduction

Variable speed electric heat pump systems have been introduced very successfully into the Japanese residential and small commercial air conditioning market since the early 1980s. In the US, compressor and heat pump manufacturers have also investigated this technology for years and first products are available now. In Europe, only limited attention was given to this technology and heat pump manufacturers are just starting to get interested.

The work reported here forms part of a larger activity undertaken in international cooperation by 5 countries within the framework of the International Energy Agency's (IEA) R&D program. Besides Italy, represented by TECNARS in Bari, also acting as "Operating Agent" of this project, Austria, Germany, Switzerland and the US are contributing to this project. The objective of the total project is to provide the necessary basis for identifying potential benefits of advanced electric heat pump systems by means of verified APF simulation models and reliable data. At present this work is focussed on variable speed electric heat pumps.

Water-to-Water Heat Pump Test Rigg

As a basis for further analytical investigations and for validation of computer models a broad range of reliable performance data of state of the art variable speed compressors were needed. For this purpose a special compressor test rigg was developed and installed at TECNARS. The test rigg was designed as a water-to-water, or alternatively, brine-to-water heat pump equipped with temperature and pressure sensors at the inlet and outlet of each major component in the refrigerant cycle as well as in the water loops of the condenser and evaporator. In addition, mass flow metering devices are installed in both water loops and in the refrigerant cycle. All data are continuously collected and evaluated by a computerized monitoring and data evaluation system.

The expansion device is split into 4 parallel units, operated manually or automatically with conventional thermostatic expansion valve or alternatively with electronic expansion valve. The heat pump test rigg is designed for operating with compressors up to a maximum of approximately 1000 kg/h R22. A general purpose frequency inverter supplies the required compressor motor power supply at frequencies up to 150 Hz for a motor capacity up to 15 kW. The inverter allows for a free selection of the output voltage/frequency characteristic. A high-precision wattmeter is measuring the power input to the inverter, thus all inverter losses are included in the compressor energy balances.

All tests of the scroll compressors, reported in this paper, were made with the heat pump operating in well established steady state conditions. By measuring temperatures, pressures and mass flow rates simultaneously both in the refrigerant cycle, and in the water loops, a refrigerant side energy balance and a water side energy balance can be calculated for both heat exchangers, the evaporator capacity and the condenser capacity. Tests were accepted only if the difference between condenser water loop heating capacity and refrigerant side heating capacity was less than 3 percent. In general, a good agreement of both energy balances was achieved, and the difference was found to be less than 1 percent with the lower values measured on the water side. Data shown in the following diagrams are based on this water side energy balance calculation.

Two variable speed Scroll compressors, S/A and S/B, were tested with this installation. [1] In the following, experimental test results of the scroll compressors are being compared to catalog data for single speed reciprocating compressors: [2]

[illegible]	[illegible]	Compressor [illegible] [Hz] / [rpm]
Scroll S/A	64.2	30-115 / 1800-6900
Scroll S/B	45	30-115 / 1800-6900
Reciprocating MT 100 HS	171.3	50 / 2900

Test results show a very similar characteristic for both compressors with compressor S/B resulting in a somewhat better COP. Data shown in the following are derived from performance tests of compressor S/A, since in this second test series the real rotor speed was measured, which was not done in the tests of compressor S/B.

Performance of Scroll Compressor S/A

In these tests we found the scroll compressor to compare very favorable with state of the art single speed reciprocating compressors. Diagram 1 shows the measured cooling capacity Q'ev of scroll S/A at constant 90 Hz and for comparison the cooling capacity (catalog data) of the reciprocating compressor MT 100. The reason for selecting the MT 100 for comparison is the capacity which is equal to the cooling capacity of the scroll at 90 Hz and high compression ratio, here approx. suction pressure 0.32 MPa at discharge pressure 1.7 MPa, a potential design point. Depending on the requirements of any particular application, the variable speed compressor S/A can be used in place of other single speed compressors with considerably larger or smaller capacity. Therefore, the comparison in diagram 1 and also in further diagrams is intended to show the typical differences between a inverter driven variable speed scroll compressor and state of the art single speed reciprocating compressors.

With increasing suction pressure both compressors provide a larger capacity because the increasing suction gas density theoretically causes an equally increasing refrigerant mass flow rate. As diagram 1 shows, the scroll's capacity increases considerably less than the MT 100 capacity does. This feature of the scroll is very favorable for heating applications, typically requiring increasing capacity with increasing pressure ratio. Diagram 2 shows the related values of the refrigerant mass flow rates of both compressors for 2 different discharge pressures, at 1.5 MPa and at 1.9 MPa. Not only the increasing suction pressure but also the decreasing discharge pressure has got a smaller impact on the scroll's capacity relative to the reciprocating compressor. For diagram 3, the refrigerant mass flow rate of both compressors was divided by the mass flow rate at suction pressure of about 0.32 MPa, the reference point, at which both compressors provide equal refrigerant mass flow rate (and cooling capacity). Thus, the increase of the mass flow rate relative to this "design point" is used for preparing the curves for diagram 3. Finally this increase of the mass flow rates is being divided by the respective increase of the suction gas density, according to the increasing suction pressure (at constant suction gas superheat of 10 K). Diagram 3 now shows that the scroll compressor's increase in cooling capacity actually is almost proportional to the according increase in suction gas density, while the recip. MT 100 shows a considerably stronger capacity increase. The reason for this behavior is the design of the scroll compressor with its local separation of the suction and discharge process, and with the absence of suction and discharge valves. The result is a very stable volumetric efficiency with a comparatively low sensitivity to the operating pressure ratio.

Volumetric Efficiency

Diagrams 4 and 5 give an image of the absolute values of the volumetric efficiency of Scroll S/A, as calculated from our test results. We consider the accuracy of these values to be in the range of about 5 %, mainly due to the difficult reading of the vibration analyzer used for measuring the real rotor speed. Despite this difficulty, it can be shown that in the greatest part of the scroll compressor's operating range, the volumetric efficiency is well beyond 90 %. The relative small sensitivity to the operating pressure ratio is shown in diagram 4 and the equally small sensitivity to changes of the compressor speed in a fairly broad speed range from 40 up to 90 Hz is shown in diagram 5.

The hard characteristic of the volumetric efficiency shows that the scroll compressor cooling capacity is responding almost linear to a change in compressor speed. This is shown in diagram 6 for the whole operating speed range of compressor S/A at constant discharge pressure of 1.7 MPa, and at two different suction pressures of 0.4 MPa, and 0.6 MPa. For comparison, the two respective (in this diagram constant) capacities of the single speed reciprocating compressor are shown as well.

Coefficient of Performance, COP

Diagram 7 shows the cooling coefficient of performance, COP_cooling, for both compressors, the scroll S/A and the recip. MT 100. This comparison immediately shows a limitation to the potential replacement of single speed systems by inverter driven variable speed systems. Despite the scroll compressor's superior efficiency relative to a reciprocating compressor, its coefficient of performance including all "additional" losses caused by the frequency inverter, will hardly reach the COP achieved with a single speed reciprocating compressor, as long as the comparison is being made at the same evaporation and condensing pressure. A higher energy efficiency can therefore only be reached if the variable speed compressor's unique ability to respond to changing load requirements leads to, heat pump internal, altered operating conditions (unloaded heat exchangers) with lower operating pressure ratios. In this case, a lower than design capacity will then be provided by the variable speed system with a higher COP than a single speed compressor would be able to produce while operating at considerably lower than design load. In general, the existence of a broadly changing load therefore, is a condition for a successful application of variable speed compressors, at least in terms of energy efficiency.

As can be seen in diagram 7, the scroll S/A shows its best COP relative to the reciprocating compressor MT 100 when operating with a pressure ratio between 3 and 4. In particular lower pressure ratios produce a decrease of the scroll's COP relative to the MT 100 COP. This is due to the scroll's built-in fixed volumetric compression ratio, in this case corresponding to a pressure ratio of about 3.3. Since the valve free scroll compressor always operates with this minimum compression factor, the compressor power input can not benefit from externally possible lower compression ratios, which explains the smaller increase of the scroll's COP relative to the MT 100 when moving towards lower compression ratios.

With the relatively hard characteristic of the volumetric efficiency in a fairly broad compressor speed range, also a relatively stable COP can be expected for a broad range of compressor speed (and therefore broad capacity range) while maintaining constant evaporation and condensing pressure conditions. As Diagram 8 shows in the case of scroll S/A, the cooling COP changes by less than 10 percent if the compressor speed is varied from 40 Hz up to 90 Hz. In this speed range the compressor shows a flat COP characteristic with an optimum at around 60 Hz. Electric losses in the compressor motor and in particular in the frequency inverter cause a rather steep decrease of the COP at low frequencies (below 30 Hz). In the high frequency range electric losses become relatively small, but the COP is reduced by increasing mechanical losses. By appropriate design of the electric motor and the inverter but also, as discussed in the next chapter by the proper selection of the voltage to frequency characteristic, the COP characteristic can be changed in order to achieve an optimum in the main operating frequency range.

Compressor Motor Power Supply Voltage Optimization

As mentioned before, the required variable frequency power supply to the compressor motor in the test rigg is being supplied by a general purpose frequency inverter (Hitachi, HFC-VWS 22 HF 3 EH). The PWM output of this inverter is very similar to the output of those inverters that are specifically designed for air conditioning applications, broadly used in Japan. These inverters, however, are operating typically with one fixed voltage to frequency characteristic as shown in diagram 9. This characteristic does not correspond to the optimal compressor power supply for two reasons: first, the voltage should continue to increase with the frequency also in the higher speed range, and second, varying load at fixed frequency require changing voltage in order to maintain maximum motor efficiency. A recent investigation at Technical University in Karlsruhe, Germany showed that a tripling of the motor load requires approximately 15 % increase of the supply voltage in order to maintain maximum motor efficiency. [3]

In our own experiments we tried to identify the optimum supply voltage for scroll S/A and S/B in a broad range of operating conditions. Diagrams 10, 11, and 12 show the impact of varying power supply voltage on heating capacity, power input, COP, and the rotor slip (in % of synchron speed), for 40 Hz (Diagram 10), 75 Hz (diagram 11), and 90 Hz (diagram 12). All diagrams are showing two curves, one for suction pressure 0.4 MPa, and a second, for suction pressure 0.6 MPa. Discharge pressure is kept constant at 1.7 MPa, superheating at 10 K, and subcooling at 5 K constant. The inverter output voltage was not actually measured throughout the tests, and the figures given in the diagrams therefore represent the set voltage selected with the inverter controls. Some deviations of the measured data from theoretically expected values can be explained by this fact, since the highly complex wave form produced by the inverter apparently changed for different settings. Some wave forms seem to be producing less and others more losses in the compressor motor. All diagrams show a similar characteristic with the sensitivity to voltage variation in general decreasing at higher frequencies. The diagrams further show that the COP maximum always is being achieved when the rotor slip is at around 2 %. Higher voltage, with all other parameters kept constant, results in a motor torque increase, a reduction of the rotor slip, and therefore, an increase of the effective rotor speed. The result is the increasing compressor capacity until the rotor almost reaches synchron speed and a further reduction of the rotor slip becomes impossible.

The experimental data available so far are not sufficient in order to identify the optimum supply voltage for changing motor load at constant frequency. Because of the rather flat optimum observed, relatively small motor load changes at constant frequency as we applied in our experiments do not provide sufficient information. For this purpose more tests with more extreme operating conditions are required. Based on the data shown in diagrams 10 to 12, the optimum voltage/frequency supply characteristic for compressor S/A has been calculated for suction pressure 0.4 MPa and discharge pressure 1.7 MPa, as shown in diagram 9.

References

[1] ASHRAE Standard 23-78, Methods of Testing for Rating Positive Displacement Refrigerant Compressors

[2] Maneurop Software Program, MSP, Version 1.1

[3] H. Späth, personal communication, Elektrotechnisches Institut, Universität Karlsruhe

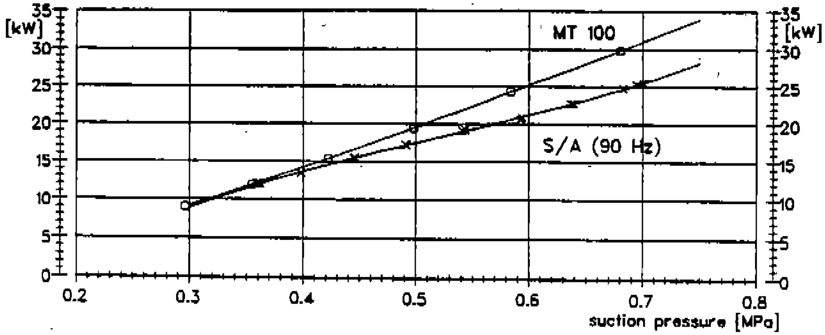

Diagram 1: Cooling Capacity Scroll S/A at 90 Hz and Recip. MT 100 HS condenser pressure 1.7 MPa, subcool 5 K, superheat 10 K

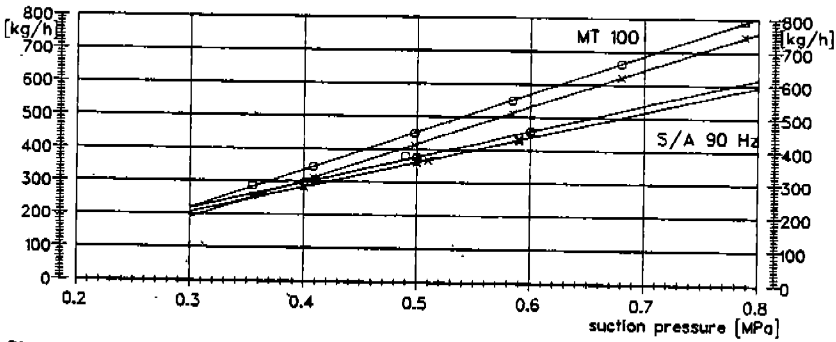

Diagram 2: Refrigerant mass flow rate scroll S/A at 90 Hz and MT 100 HS condenser pressure: 1.5 = xx, and 1.9 MPa = oo; superheat 10K

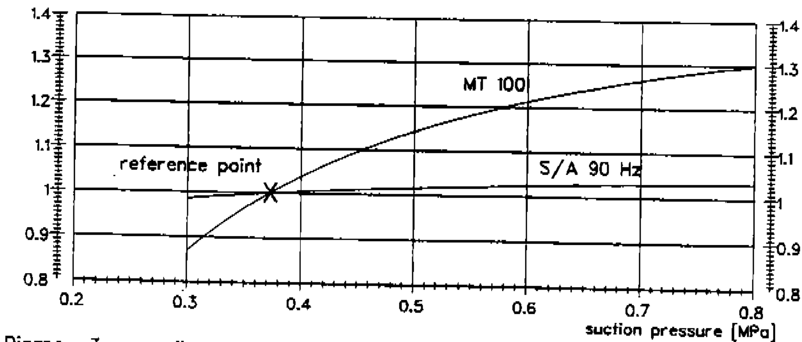

Diagram 3: mass flow rate increase per suction gas density increase

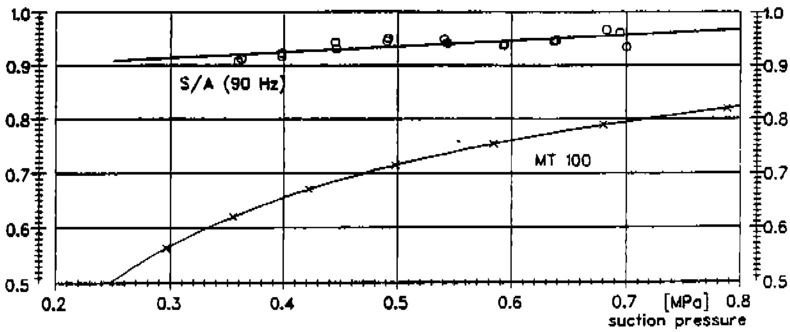

Diagram 4: Volumetric Efficiency Scroll S/A at 90 Hz and Recipro. MT 100
condenser pressure 1.7 MPa; suction gas superheat 10 K

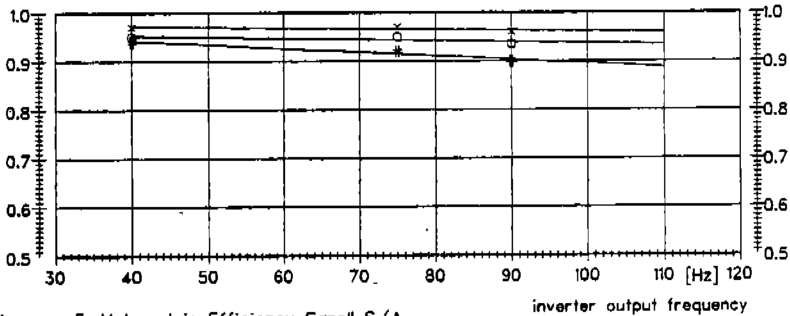

Diagram 5: Volumetric Efficiency Scroll S/A
pressure ratio (p_disch. / p_suc): x = 2; o = 3; # = 4;

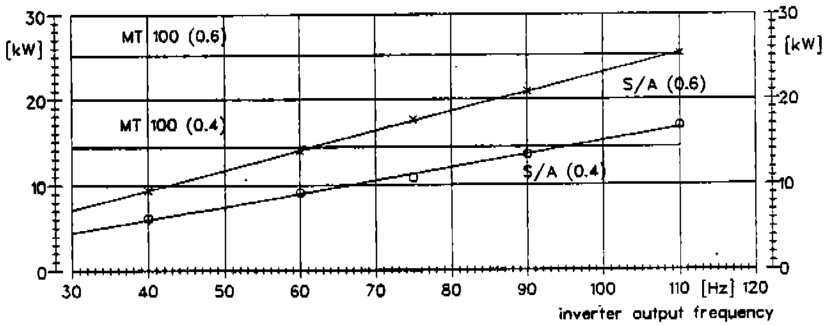

Diagram 6: Cooling Capacity Scroll S/A and Single Speed Recip. MT 100
suction pressure: o = 0.4 MPa; x = 0.6 MPa; p_disch.= 1.7 MPa

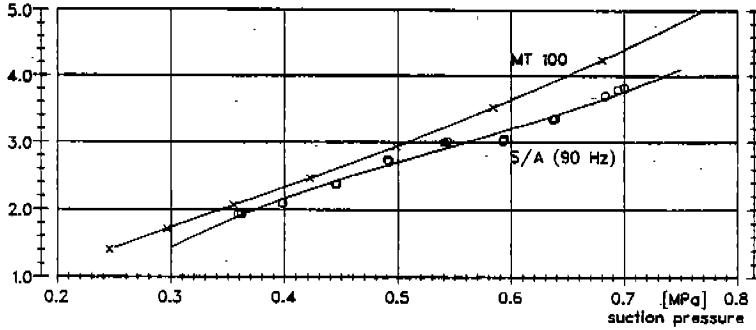

Diagram 7: COP for Scroll S/A at 90 Hz, and Recipro MT 100
condenser pressure 1.7 MPa; subcooling 5K; superheating 10K

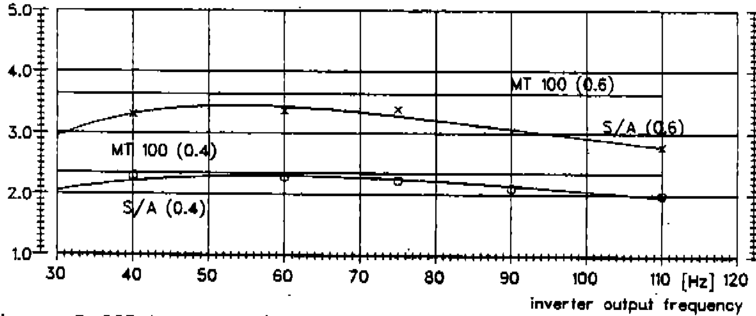

Diagram 8: COP for Scroll S/A and Recipro. Compressor MT 100
p_disch.=1.7 MPa; suction pressure: o = 0.4 MPa; x = 0.6 MPa

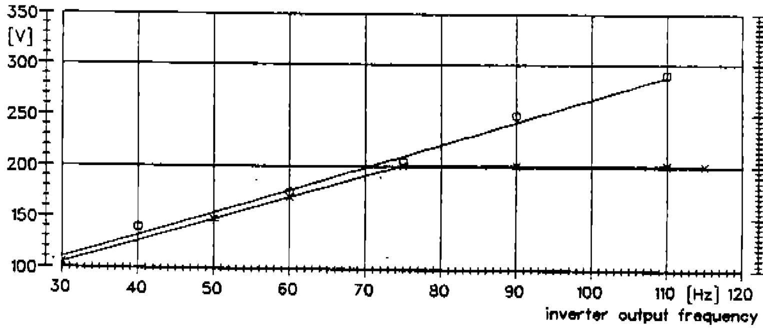

Diagram 9: Voltage versus Frequency Supply Characteristic
x = Typical for Heat Pumps in Japan; o = S/A Test Optimum

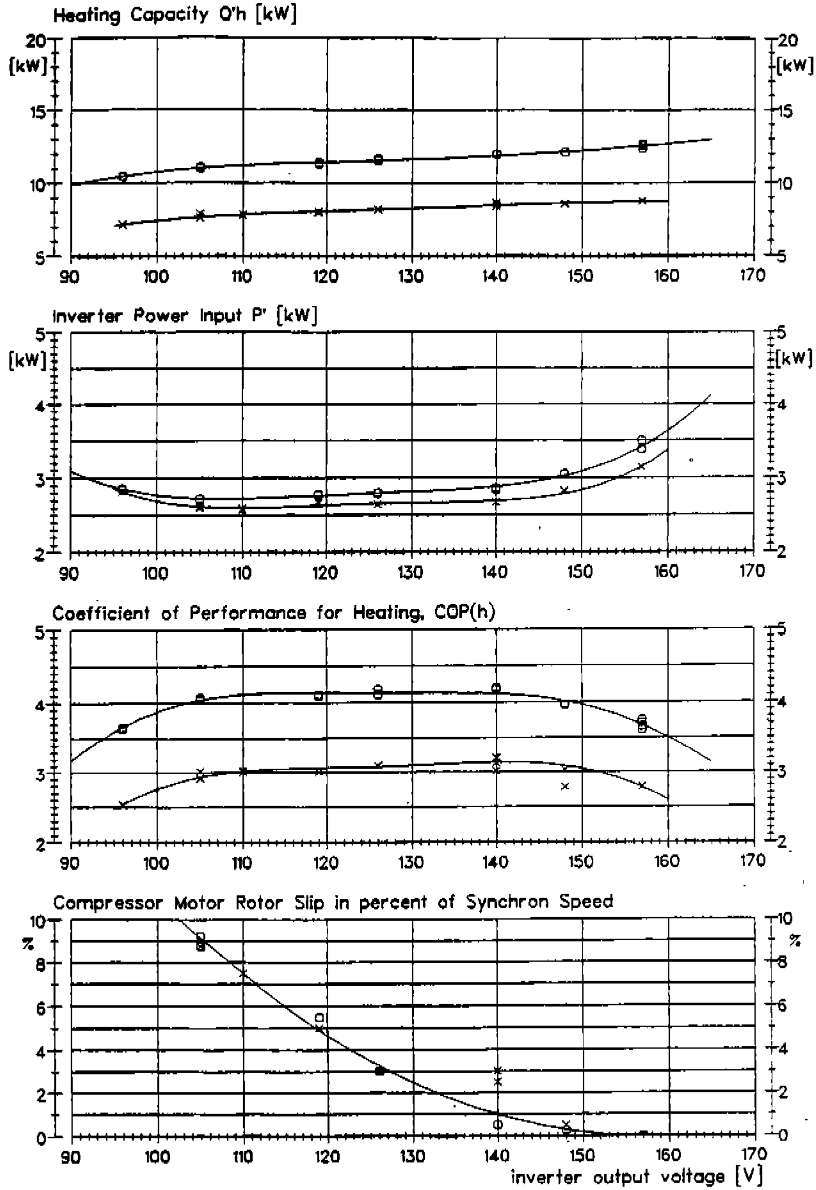

Diagram 10: Effect of voltage variation at 40 Hz with scroll S/A
p_suction = 0.4(x) MPa and 0.6(o) MPa; p_discharge = 1.7 MPa
suction gas superheating = 10 K, subcooling = 5 K

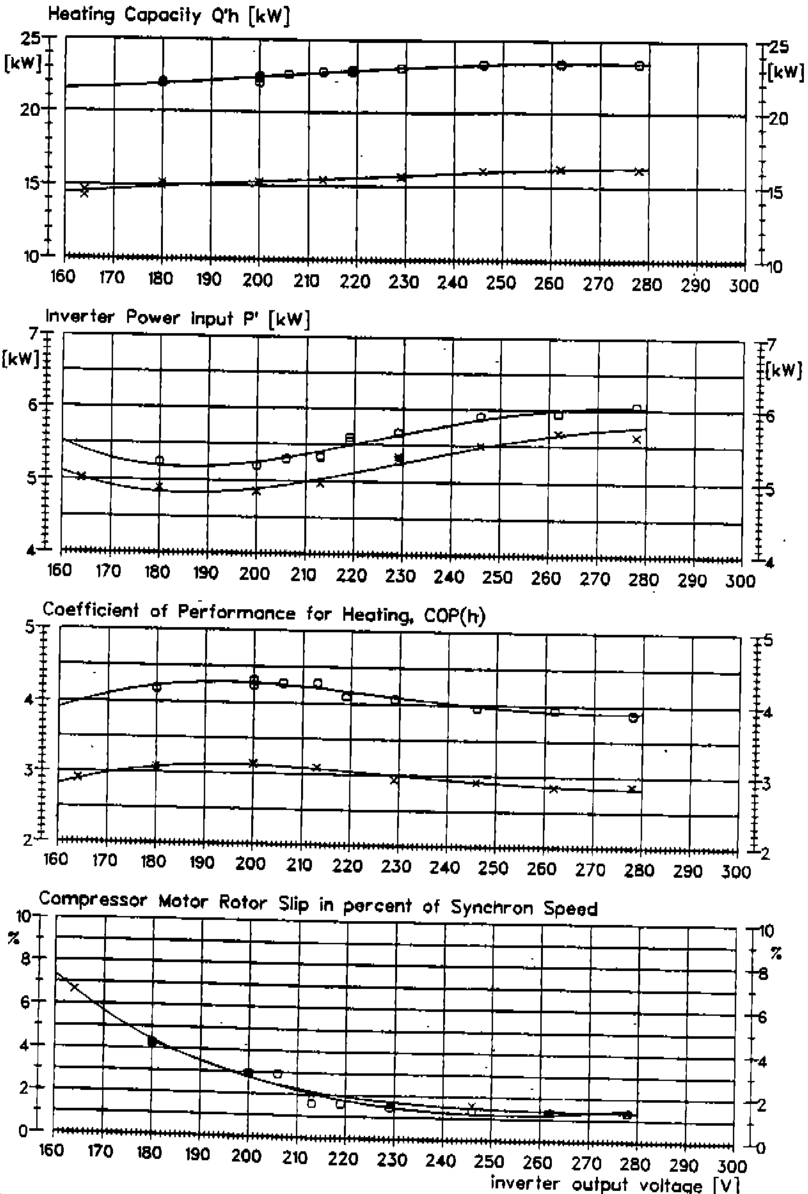

Diagram 11: Effect of voltage variation at 75 Hz with scroll S/A
p_suction = 0.4(x) MPa and 0.6(o) MPa; p_discharge = 1.7 MPa
suction gas superheating = 10 K, subcooling = 5 K

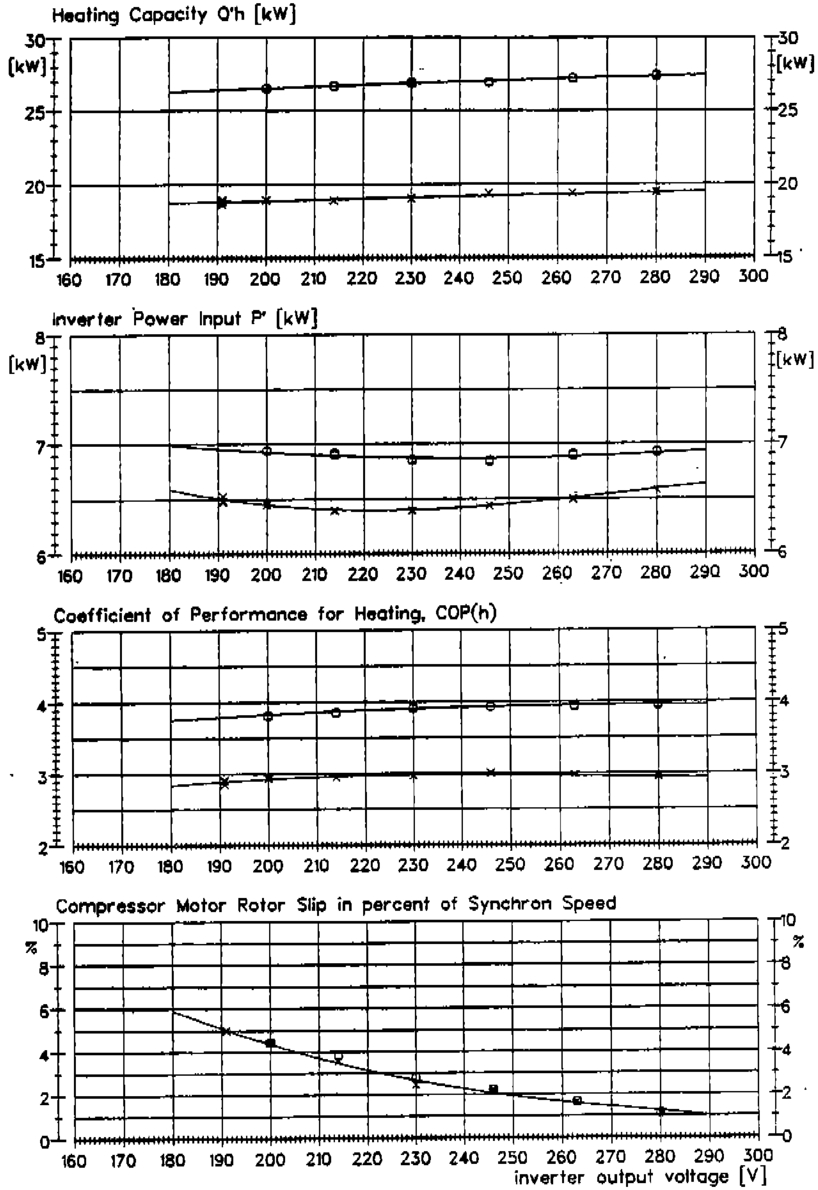

Diagram 12: Impact of voltage variation at 90Hz on scroll S/A
p_suction = 0.4(x) MPa and 0.6(o) MPa; p_discharge = 1.7 MPa
suction gas superheating = 10 K, subcooling = 5 K

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