EXPERIENCES WITH HERMETIC SCROLL COMPRESSORS WITH INTERMEDIATE VAPOR INJECTION PORT IN HEAT PUMPS FOR HIGH TEMPERATURE LIFT HEATING APPLICATIONS

M. Zehnder, Research Associate, Ecole Polytechnique Fédérale de Lausanne (EPFL), Lausanne, Switzerland
D. Favrat, Professor, Laboratory for Industrial Energy Systems (LENI), Ecole Polytechnique Fédérale de Lausanne, CH-1015 Lausanne, Switzerland

ABSTRACT

In order to extend the application range of air-water heat pumps to high temperature residential heating systems, new small hermetic compressors with remodeled injection ports for vapor injection have successfully been tested in laboratory. The injection flow has been controlled by means of either an electronically regulated expansion valve or of a capillary tube. Extreme load and operating conditions could be achieved (air –12°C / water 65°C), using R-407C, while keeping the limits in discharge temperature and pressure of the manufacturer. Using an economizer heat exchanger in the mid pressure stage of the cycle, the heat output in very low external temperatures could be raised up by 30%, compared to the reference cycle without injection. The COP could be improved on a broad range of operating conditions.

Based on the experiences of tests with three distinct injection flow paths, a new developed generalized compressor model is presented and the geometric design aspects are discussed in order to further improve the system efficiency.

Key Words: ***heat pump, economizer cycle, vapor injection port, scroll compressor, retrofit***

1 INTRODUCTION

The application range of small to medium sized heat pumps was for a long time limited to a restricted range of temperature lift due to the high condensation pressure, to the extreme discharge temperature of the compressor and to limits in chemical instability of the refrigerant-oil pairs.

With the now widely introduced high pressure zeotropic refrigerant R-410A, many high pressure components are commercialized allowing pressure levels up to 45 bar. Using these components and with adequate synthetic refrigerant-oil pairs, high temperature lift heating applications are theoretically reachable. But high discharge temperatures still result in increased thermal load of the compressor, which can results in premature failure during operation. The highly decreasing heating capacity and COP at extreme operating conditions are major drawbacks to face competition in these market segments. This therefore requires further development of heat pumps based on improved concepts and components, rather than trying to extrapolate the application range with conventional components.

Several solutions of improved heat pump cycles allow a substantial broadening of the operating range, and include the following features:

- Highly reduced system temperatures at the compressor discharge.
- Increased specific heat output at extreme operating conditions.
- Improved COP over a broad application range.

The most advanced heat pump cycles have been identified by (Granryd, 1992) and have found their adaptation in very large sized heat pumps (Bailer, 2002), achieving a temperature lift of 80 K with two- and three-stage heat pump cycles, using centrifugal compressor technology, or in (NEDO, 1993), which is a multi stage condensing Lorenz cycle unit. The downsizing of these optimal configurations to small units suffers from the complexity, the induced costs, and the unavailability of key components. The most promising heat pump cycles for the residential heating could be identified by a series of projects in the framework of the *Swiss Retrofit Heat Pump*, sponsored by the Swiss Federal Office of Energy and a summary of these activities is included in the thesis of (Zehnder, 2004). **Two stage compression heat pump cycles** show the largest potential for improving the performances, but are technically and commercially hard to be realized in a short time horizon. Due to the complexity and the still not easily resolved problems of oil migration, causing a lack of reliability in the compressor lubrication, the two stage compression cycle is still the future option in advanced heat pump design (Favrat et al, 1997). As an intermediate solution, a “booster” concept has demonstrated a substantial increase of the heat output (Zehnder, 2002b).

Using **economizer cycle heat pumps** (Fig. 1) with a compressor allowing intermediate injection during the compression, is the most effective short term approach to achieve the technical and economic requirements of such units. Within the framework of the multi national collaboration effort, different compressor models have been tested and some of the results are presented herein. A generalized simulation model has been developed, allowing a reasonable reproduction of the experimental data and a prediction of the cycle performances at new operating conditions, with new fluids and with new geometrical setups.

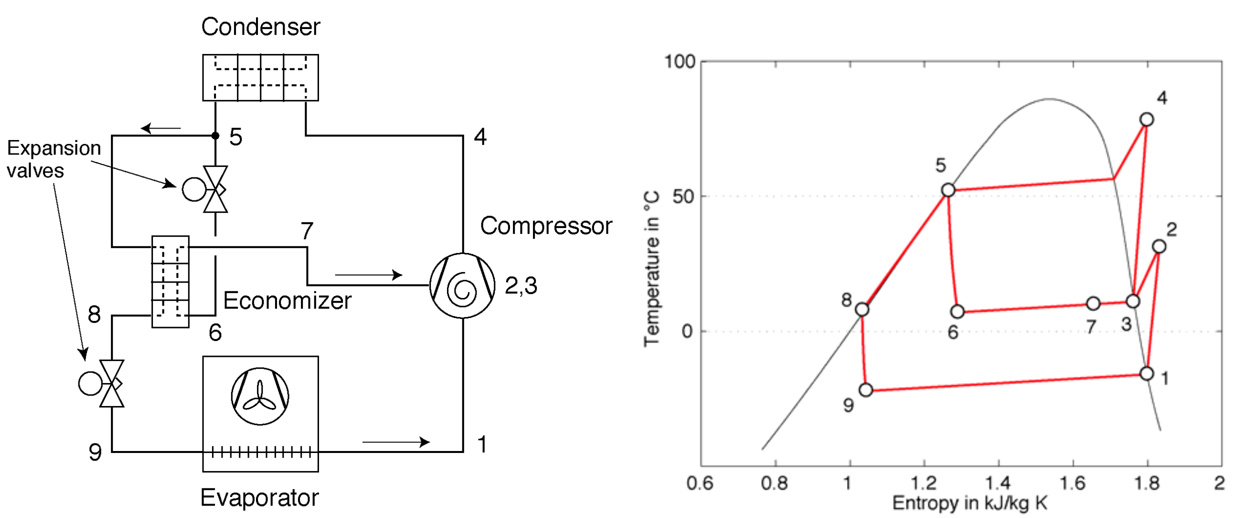

Fig. 1. Economizer heat pump: Flow chart and thermodynamic concept in T-s diagram.

The theoretical design of the economizer stage is basically function of the intermediate pressure level. As in two-stage compression heat pumps the optimal design pressure is close to the geometrical average pressure between the condensation and evaporation pressure. This rule is often applied in the refrigeration applications, but has to be verified for heating applications, when the upper stage is approaching the critical point. In the latter case, the optimum design pressure for the economizer stage is shifted upwards (Fig. 2) reinforcing the importance of the lower stage.

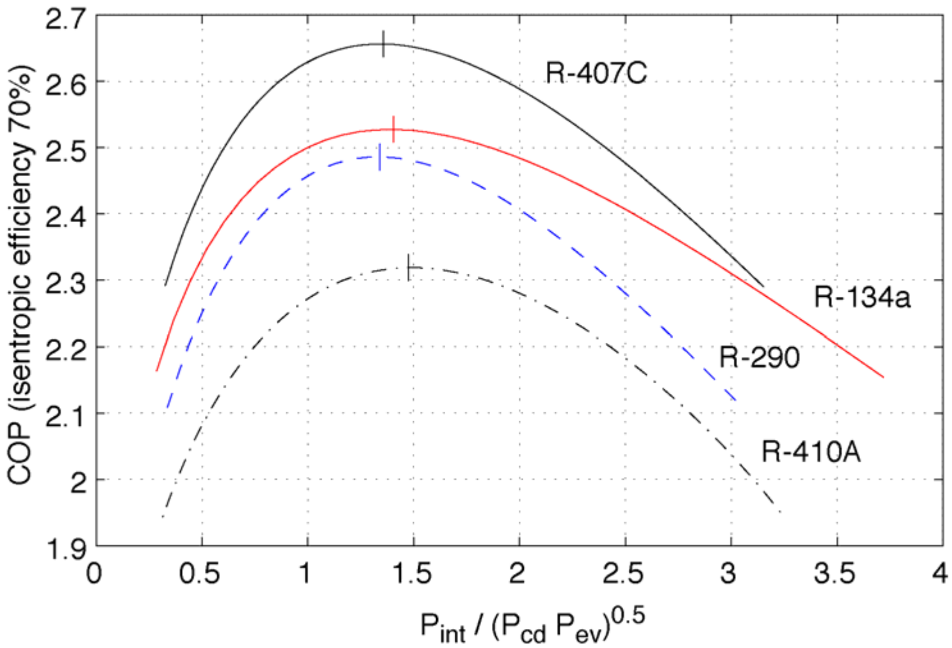

Fig. 2. Optimal design intermediate pressure in theoretical economizer or two-stage heat pump cycles. Air inlet –12°C / heating water exit 60°C.

2 COMPARISON OF TEST RESULTS WITH TWO DIFFERENT SCROLL COMPRESSORS (OLD VERSUS NEW INJECTION FLOW GEOMETRY)

Two different compressor types, both equipped with an injection port allowing refrigerant injection during compression have been tested in two different test setups. The aim of the first setup was to test, on a brine/water heat pump, an already commercially available compressor (the ZF15KVE Copeland scroll) in an economizer heat pump cycle. Test runs with variable injection flow showed a rapid decrease of the COP and maximum injection rates limited to 50% relative to the suction rate as represented in Fig. 3. The evolution of the intermediate pressure shown in Fig. 4 demonstrates, that this type of compressor was not adequate for the injection of vapor, and has limitation to liquid injection in low temperature refrigeration applications. However these tests showed, that a substantial increase of the heat output could be obtained without penalizing the COP (at relative injection flow rates of about 20%).

In the second test setup an air/water heat pump has been successively equipped with two different prototypes of scroll compressors with a modified injection flow path enabling higher injection rates of saturated vapor and reduced pressure losses in the injection channel. The results with the first prototype compressor have been presented in (Zehnder, 2002a) and were based on a scroll compressor with a low built-in volume ratio, thus not adapted for the high temperature lift heating applications, which was our main objective. The laboratory test runs with the second prototype compressor (a pre-version of a now commercially released Copeland ZF13KVE EVI model) have resulted in a boost of the injection flow rate (up to 200% relative to the flow rate at the compressor suction line) with reduced intermediate pressure levels. The successful field tests of this new compressor design has resulted in its commercialization as referred to in the paper of (Beeton, 2003) in a recent issue of the ASHRAE Journal.

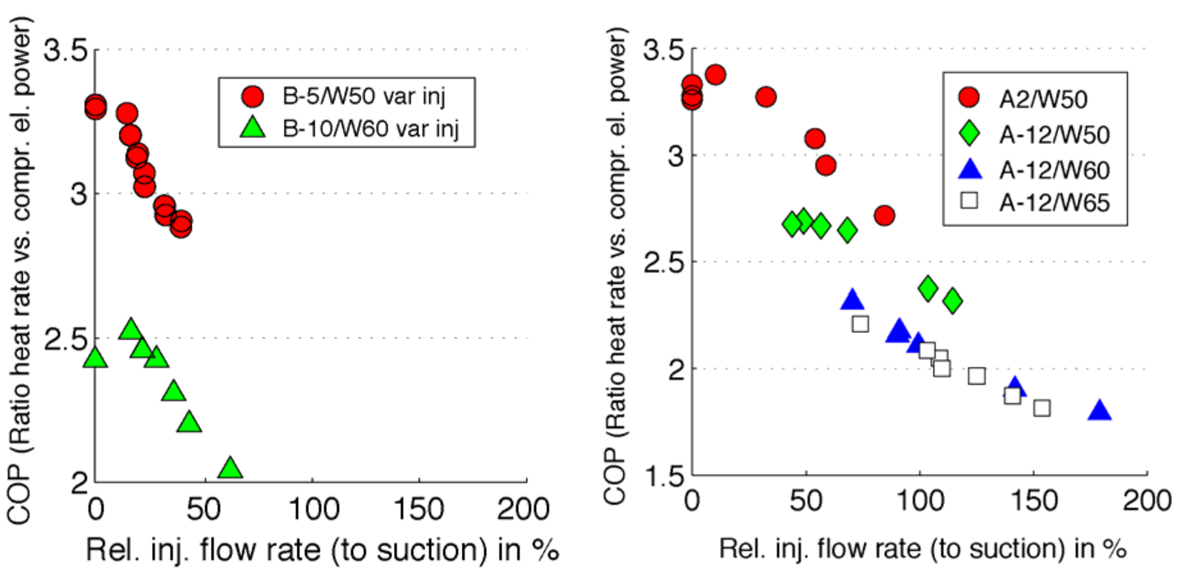

Fig. 3. Test results (COP) of a brine-water heat pump (test conditions given with Bxx/Wxx: brine inlet and heating water exit temperatures in °C) equipped with a conventional hermetic scroll compressor with liquid injection ports (ZF15KVE) and of an air-water heat pump (test conditions given with Axx/Wxx: air inlet and heating water exit temperatures in °C) with a new injection flow path for vapor injection (ZF13KVE EVI-Prototype).

The achieved high injection rates imply a substantial reduction of the discharge line temperatures, but injection rate control has to be done carefully, in order to avoid partial two-phase compression. Critical injection flow rates are calculated with the help of an extended simulation model presented hereafter.

3 EXTENDED SIMULATION MODEL, INCLUDING COMPRESSOR WITH INTERMEDIATE INJECTION PORT

A steady state simulation model for air/water heat pumps, including different high performance cycle configurations, which can be considered in high temperature lift applications, has been developed and includes a generalized model of hermetic rotary volumetric compressors with the option of an intermediate injection (Zehnder, 2004). The models are programmed with Matlab and are coupled with an adaptive fluid interface code, called FLINT, internally developed at LENI[1] allowing an easy communication with fluid properties programs like REFPROP (with version 7, (Lemmon et al. 2002)).

[1] LENI=Laboratoire d'Energétique Industrielle (Laboratory for Industrial Energy Systems) at the Swiss Federal Institute of Technology of Lausanne, Switzerland.

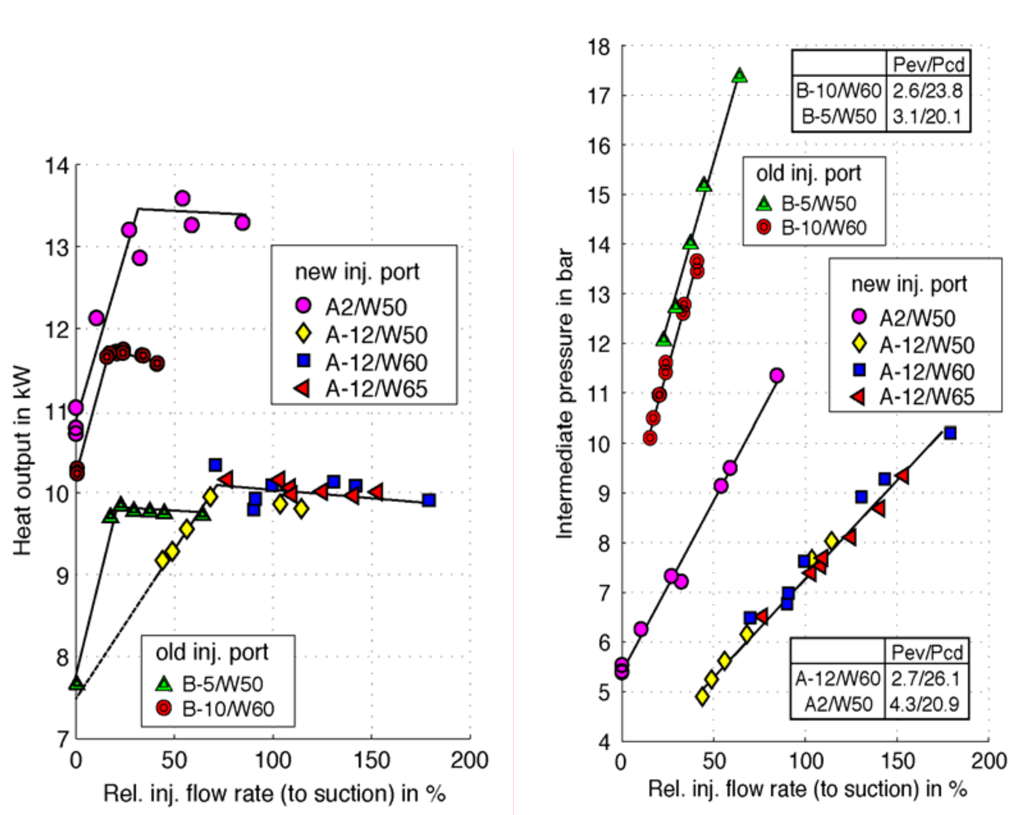

Fig. 4. Test results (heat output and intermediate pressure) for conventional hermetic scroll compressor with liquid injection port (ZF15KVE) and with a new injection flow path for intermediate vapor injection (ZF13KVE EVI-Prototype).

Neglecting dissipation and in-chamber heat transfer, the theoretical transformation during compression corresponds to an isentropic transformation. Considering the geometrical characteristics of fixed built-in volume ratio compressors, such as scroll compressors, over-compression and under-compression losses occur due to non adapted pressure conditions and are the main sources of reduced performance. The generalized form of the theoretical compression efficiency, accounting only for these losses is according to (Sauls, 1982):

$$\eta_{cp,th} = \frac{\int_1^{2s} v\,dP}{\int_1^{2} v\,dP} = \frac{\dot{M}_{refr}\ v_{in}\ P_{in}\ \frac{\kappa-1}{\kappa}\left[\Pi^{\frac{\kappa-1}{\kappa}}-1\right]}{\dot{M}_{refr}\ v_{in}\ P_{in}\ \frac{\kappa-1}{\kappa}\left[\Pi_i^{\frac{\kappa-1}{\kappa}}-1\right]+\left(P_{ad}-P_{dis}\right)V_{dis}} \qquad \text{Eq. 1}$$

$$= \frac{\kappa\left[\Pi^{\frac{\kappa-1}{\kappa}}-1\right]}{\frac{\kappa-1}{VR_i}\Pi + VR_i^{\kappa-1} - \kappa}$$

$$\text{with } \Pi = \frac{P_2}{P_1},\ \Pi_i = VR_i^{\kappa} \text{ and } 1 = \text{in},\ 2 = \text{out},\ 2s = \text{isentropic transformation}$$

A simple model, which predicts with a good accuracy the performances of scroll compressors in many application cases, is developed by adding a constant multiplier to the theoretical efficiency expressed by Equation 1. The multiplier factor accounts for additional losses occurring in the real transformation (inlet and discharge, generated turbulences during the compression flow path due to volumetric losses in the clearances,…). This simple approach is limited to single stage configurations with or without intermediate injection and therefore cannot predict the performances of two-stage compression configurations. In order to follow more realistically the effective transformation path from the inlet of the compressor to the discharge, the losses attributed to the suction and the discharge zones are considered separately. This decomposed compressor model is used as the starting point for the more extended model including the injection flow, which is described in the following section.

The proposed one point mixing approach found in literature (Winandy, 2002; Guoyuan, 2003) for the simulation of the intermediate injection has been extended by a model, which accounts for the distributed injection (during the opening of the injection port to the compression chamber). This approach allows one to follow closely the transformation during the compression and to find the maximum injection flow rates avoiding two-phase compression. In order to enable extrapolations from the measured characteristics, the geometry of the injection flow path is defined by a capillary tube analogy.

The extended model includes the following zones for calculation (see also Fig. 5):

in – 1: Admission preheat of the refrigerant due to internal transformation losses linked to the electromechanic conversion. No pressure drop is assumed. The rate of preheat is calculated using the relation:

$$\dot{Q}_{preheat} = f_{transm} \dot{V}_{displ} \qquad \text{Eq. 2}$$

with the displacment $\dot{V}_{displ} = V_{asp} \cdot n \quad \text{in } m^3/s$

and the empiric coefficient $f_{transm} = 0.09 \cdot 10^{-6} \text{ J m}^{-3}$

1 – 2: Adiabatic compression until the injection ports open to the compression chamber[2]. The compression efficiency is calculated with the nominal efficiency $\eta_{cp,is}^{nom}$, which is adapted for each compressor. In the absence of specific knowledge of the compressor nominal efficiency a value of 70% can be assumed.

2 – 3: Injection phase, subdivided into multiple calculation steps during the opening of the injection port to the compression chamber. Each step consists of the partial operations a) isobaric mixing of the added fluid with the refrigerant in the compression chamber, b) isentropic transformation for regaining equilibrium conditions, respecting the total volume of the compression chamber at corresponding angular position. c) Differential compression step for a fixed change of the compression volume, assuming a nominal efficiency. If the partial compression step drop into the two phase region, an significant efficiency drop of an empiric value of 50% is assumed.

3 – 4ad: Adiabatic compression using the nominal efficiency value until the end position, when the compression chamber opens to the discharge port.

4ad – 4: Instantaneous pressure adaptation of the compression chamber pressure to the system pressure and discharge of the vapor through the discharge port. The compressor is assumed not to have any discharge valve. The transformation is given by the relation:

[2] In actual compressor design the injection port is opening to the compression chamber directly at the sealing position of the mobile scroll member.

$$h_{dis} - h_{ad} = \frac{(P_{ad} - P_{dis})\, V_{dis}}{\dot{M}_{refr,dis}} \qquad \text{Eq. 3}$$

In order to calculate the refrigerant mass flow rate at the compressor suction line, it is important to relate the flow rate to the theoretically expected mass flow rate, which results from the displacement of the compressor $\dot{V}_{displ}$. This is calculated with the volumetric efficiency of the compressor and is evaluated by:

Compressor with lower built-in volume ratio (Proto 1, vap. inj. port):

$$\eta_{vol} = 1.012 - 0.027\,\Pi \qquad \text{Eq. 4}$$

Compressor with higher built in volume ratio (ZF15 (liq. inj. port) and Proto2 (vap. inj. port)) :

$$\eta_{vol} = 0.97 - 0.014\,\Pi \qquad \text{Eq. 5}$$

These two expressions are valid for operational modes without injection flow. When the injection line is opened the volumetric efficiency is influenced by the added mass flow from the injection line and an empirical correction is proposed for all tested compressor models:

$$\eta_{vol,inj} = f_{vol}\ \eta_{vol,no\,inj} \quad \text{with} \quad f_{vol} = 1 - 0.1323\,\frac{\dot{M}_{inj}}{\dot{M}_{adm,th}} \qquad \text{Eq. 6}$$

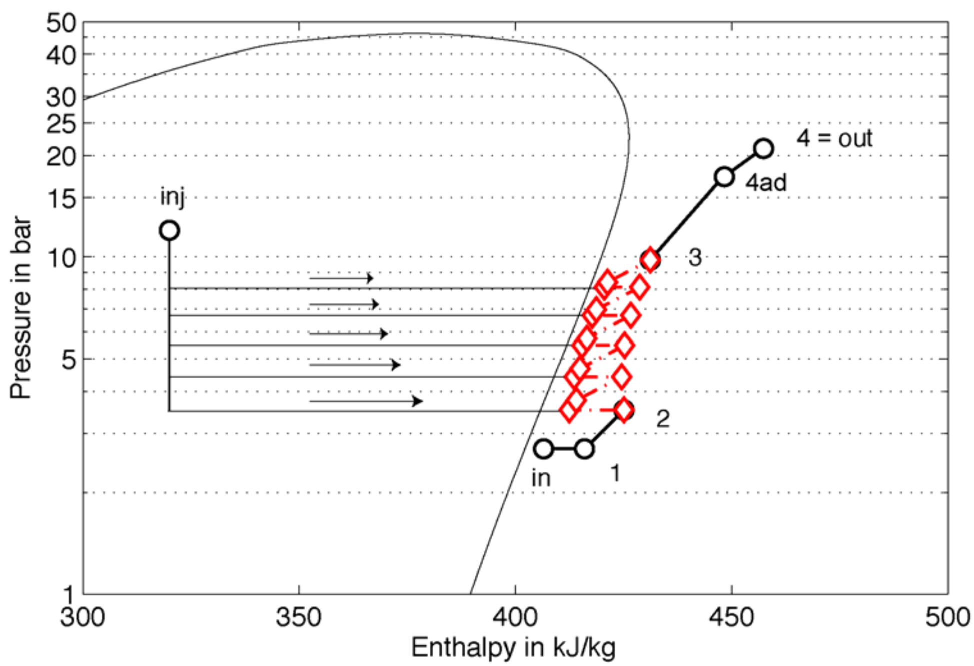

Fig. 5. Schematic representation of the calculation zones in a decomposed compressor model including intermediate injection flow.

The correction proposed in Equation 6 is referred to the theoretical mass flow rate at the suction port of the compressor in order to avoid iterative resolution of the compressor simulation model. The flow rate of the injection line is calculated by the empirical approach proposed in Equation 7, or with the capillary tube analogy including the various geometrical data for each compressor.

An experimental fit to the data set of three compressor models has shown, with a good agreement, a linear dependency of the relative injection flow rate in relation to the partial pressure ratio between the evaporating pressure and the pressure in the economizer stage (see Fig. 6).

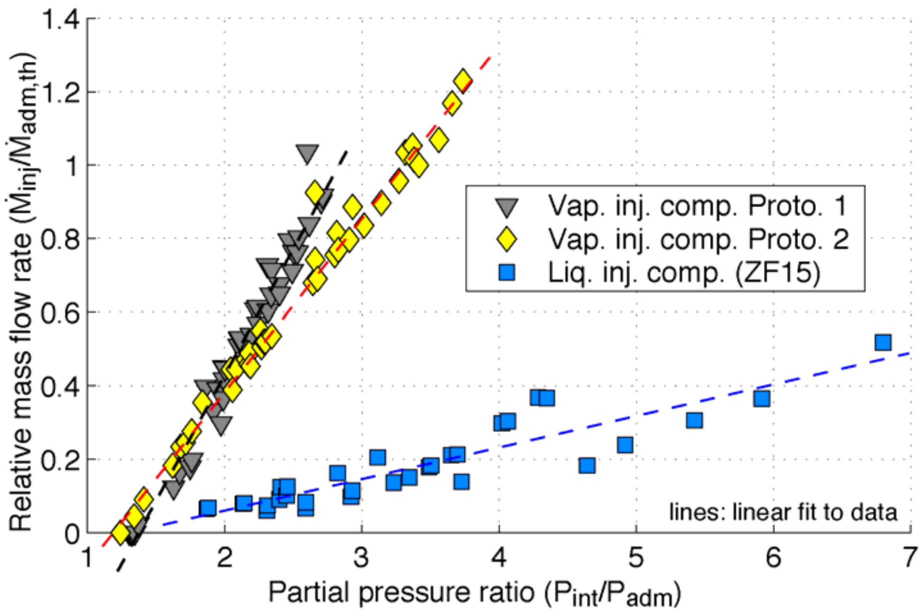

Fig. 6. Linear fit relating relative injection mass flow rate to the partial pressure ratio for three distinct hermetic scroll compressors with intermediate injection port. All experimental data are with refrigerant R-407C.

$$\dot{M}_{inj} = \left(a + b\frac{P_{int}}{P_{adm}} \right) \dot{M}_{adm,th} \qquad \text{Eq. 7}$$

$$\begin{aligned} \text{with} \quad & \text{Proto1:} \quad a = -0.919,\ b = 0.675 \\ & \text{Proto2:} \quad a = -0.566,\ b = 0.476 \\ & \text{ZF15:} \quad a = -0.110,\ b = 0.0875 \end{aligned}$$

In order to best match the data, analog capillary tube (characterized by an estimation of the tube diameter D and the tube length L) have been calculated and results are provided by the data in Table 1.

Table 1. Injection geometries of the capillary tube modeled for the tested compressors

Compressor	Diameter D in mm	Tube length L in mm
Vapor injection port #1	3.6	300
Vapor injection port #2	3.5	300
Liquid injection port (ZF15)	2.3	600

Using this approach the injection flow rates can be predicted with an overall precision of +/- 20% except for the low injection flows, where the effective flow rate is systematically overpredicted by the simulation. Using the proposed approach for a compressor with intermediate injection, the measured compression power can be predicted with a precision of +/-10%.

4 SYSTEM ANALYSIS USING THE EXTENDED SIMULATION MODEL

The proposed simulation model has been implemented to a heat pump cycle including theoretical heat exchangers (calculated with a specified minimal local temperature difference), in order to demonstrate, using a parameter analysis, the effect of variable injection flows (Fig. 7) and variable design positions of the injection port on the cycle performances (Fig. 8) and on the system key values (Fig. 9). The geometric setup of the latest compressor design with injection port for vapor injection has been used to produce the following graphs. As an example of highly demanding operating conditions, air inlet at –12°C and heating water output at 60°C have been chosen. R-407C is used as refrigerant.

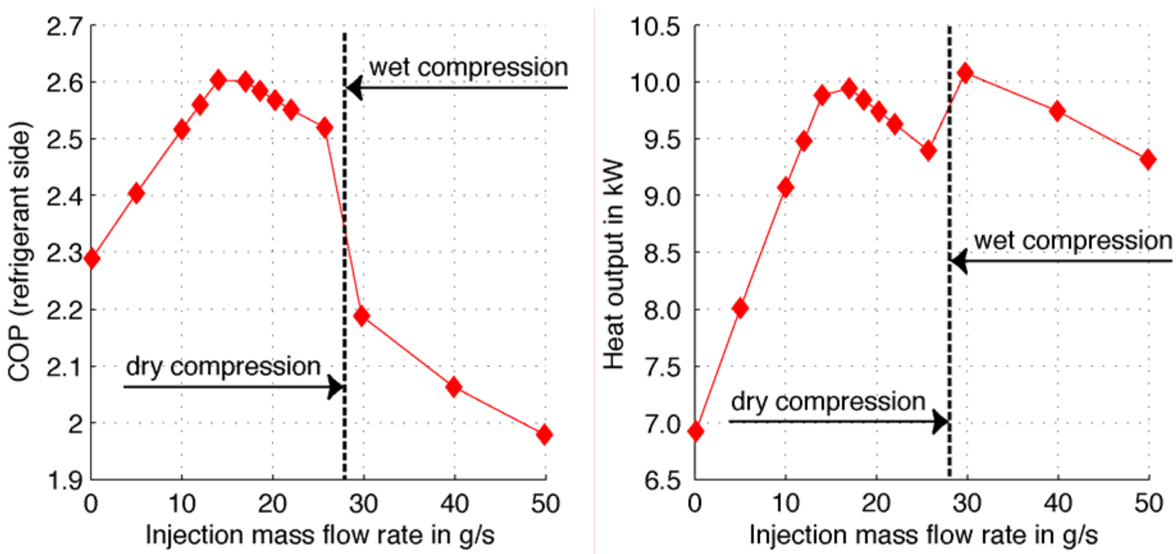

Fig. 7. Simulated system performances versus. variable injection flowrate with indication of the limit rate at which two-phase compression is occurring.

The calculated evolution of the cycle performance, shown in Fig. 7, are in agreement with the experimental data. The simulated discontinuities in the evolution of the COP and heat output originate from the distinction being made between the compression efficiency in vapor phase and in the two-phase region, where increased leakage in the gaps of the compression chamber are expected (Afjei, 1992). Due to important system fluctuations generated by the expansion control device (thermostatic expansion valve) it was not possible to clearly reproduce experimentally the predicted discontinuities. Moreover and in order to limit the thermal load of the compressor only test runs with higher injection rates than required at the COP peak have been performed. The predicted peaks have however been found in operating conditions at lower temperature lifts (Figs. 3 and 4). The best compromise between reduced thermal load and highest performance is achieved at conditions approaching the limit of two-phase compression.

Assuming a redesign of the angular position of the injection ports, the effects on the system performances can be simulated by the proposed model. Figure 9 shows the trends for two operating conditions (A0/W50: air inlet at 0°C / water outlet at 50°C and A-12/W60: air inlet at –12°C / water outlet at 60°C). The setup of the tested prototype of compressor corresponds to an angular position of the injection ports at 0° (injection is starting at sealing (Perevozchikov, 2003)).

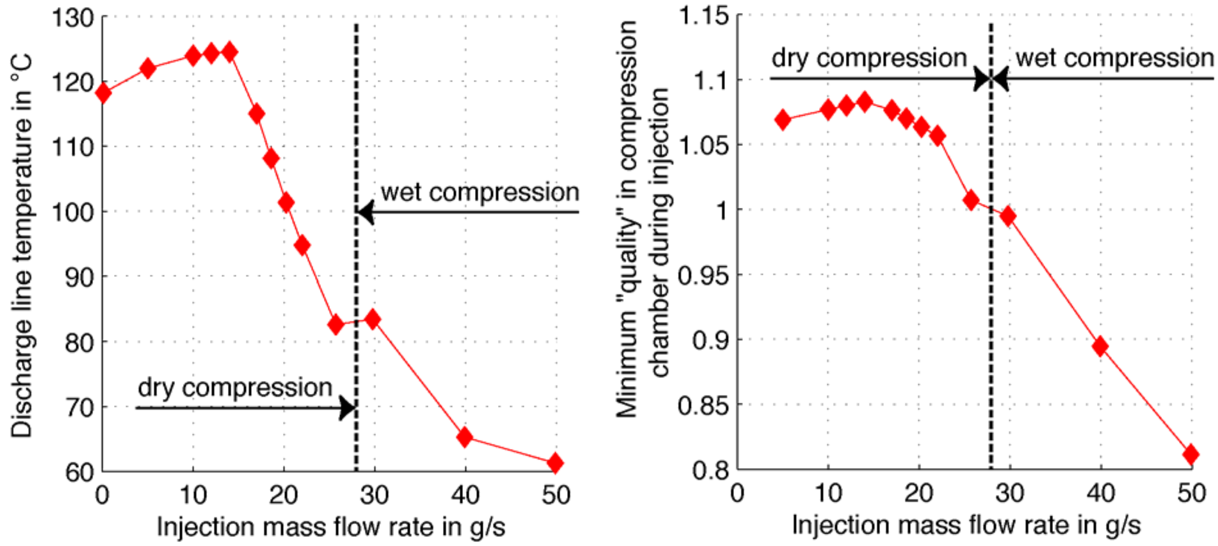

Fig. 8. Simulated discharge line temperature and minimal "quality"[3] during compression vs. variable injection rate with indication of the limit rate at which two-phase compression is occurring.

A delayed positioning of the injection port would increase the system's COP. This is due to the reduced specific power demand of the vapor to be recompressed. The higher evaporation temperature in the economizer heat exchanger however reduces the capacity of liquid subcooling. This implies a reduction of the injection rate and of the total mass flow rate in the condenser. In these conditions, the heat pump's heat rate is therefore reduced.

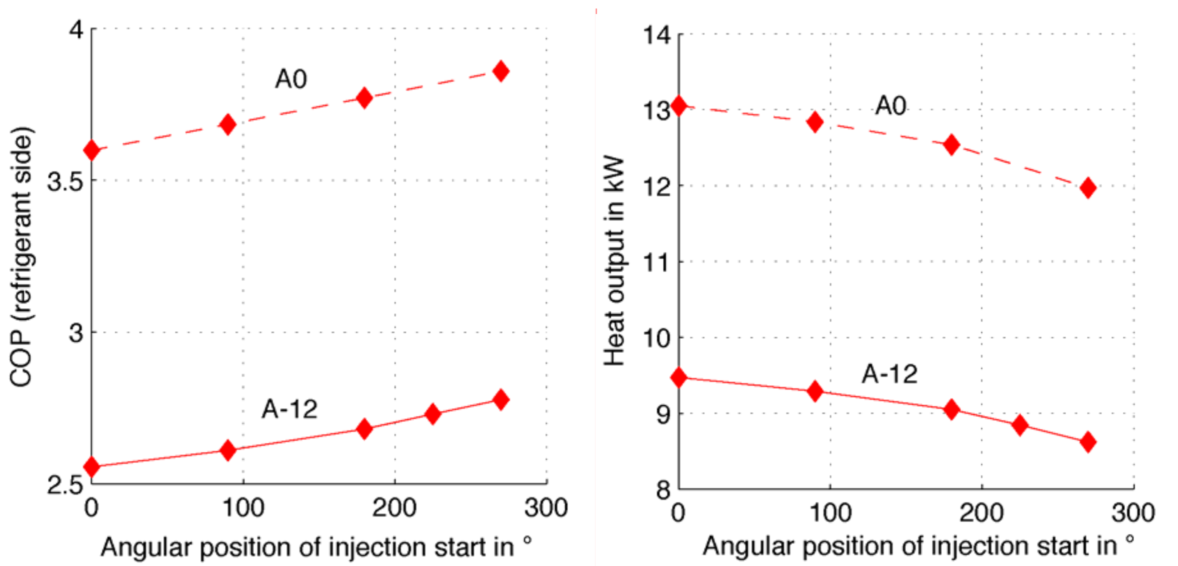

Fig. 9. Evolution on the system performances expected from a redesign of the injection port's angular position in an economizer heat pump cycle, assuming maximum allowed injection rate (touching the dew line during compression).

[3] The same quality definition is used for saturated refrigerant, as well as for refrigerant in superheated vapor phase: $x = (h - h_{liq}) / (h_{vap} - h_{liq})$

5 CONCLUSIONS

This paper compares the experimental results of hermetic scroll compressors, which are equipped with an injection port, allowing liquid or vapor injection during the compression phase. The data show the high heat rates allowed by the new generation of compressors, which allow the residential heat pumps to run in an extended application range, making them applicable for the retrofit of fuel based boilers in relatively high temperature hydronic heating systems.

An extended simulation model is proposed, which includes some design parameters of the injection flow path. This model allows an extension of the calculation beyond the test limits. It has been used to evaluate the limit injection rates, at which two-phase compression would start to occur with potential material destruction. By variation of the injection port position it can be shown how the system performances are expected to change. Design optimization could be imagined in a further step.

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NOMENCLATURE

<u>Symbols</u>

COP		Coefficient of performance
f_{transm}	J/ m^3	Empirical transmission losses coefficient
f_{vol}		Volumetric efficiency correction factor
h	J/kg	Enthalpy
$\dot{M}$	kg/s	Massflow rate
n	rev/s	Rotational speed
P	bar, Pa	Pressure
$\dot{Q}$	W	Heat rate
T	K, °C	Temperature
V_{asp}	m^3/rev	aspirated compression volume / revolution
V	m^3	Volume
$\dot{V}$	m^3/s	Volumetric flow
VR		Volume ratio
x		Vapor quality
κ		Isentropic exponent
Π		Pressure ratio
η		Efficiency

<u>Subscripts</u>

ad	adapted
displ	displacement
in, adm	inlet, admission
inj	injection
int	intermediate
liq	(saturated) liquid
out, dis	outlet, discharge
th	theoretical
vap	(saturated) vapor
vol	volumetric