2004

Test Results of A Screw Type Expander/Compressor and the Implication of Phase Separators on the Refrigeration Process

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ABSTRACT

Running a refrigeration cycle at saturated liquid conditions at the evaporator inlet has a few interesting advantages in comparison to the conventional 2-phase situation. Opportunities in using novelty heat exchangers, increased cycle efficiency and significantly reduced main compressor sizes occur. Using the two rotor integrated compressor/expander idea developed by Olofsson (1993) in a refrigeration cycle theoretically offer the possibility of pure liquid evaporator inlet conditions, the device is called Phase Separator. This paper reports results from hardware tests as well as some theoretical results. To evaluate the efficiency of a Phase Separator three terms of efficiency are suggested. The test results prove that pure liquid conditions to the evaporator are obtainable with a practical Phase Separator.

1. INTRODUCTION

The Phase Separator Fig. 1, is a rotary positive displacement device performing three main functions within the refrigeration cycle.

1. Maintaining the differential pressure between the condensing level and the evaporating level by expanding condensate at a controlled rate. (Thus replacing the expansion valve).
2. Separating liquid from gas after the expansion process and feeding the liquid towards the evaporator.
3. Recompressing the gas, which is produced during the expansion process, to a pressure suitable for inserting it into the compressor of the main refrigerant gas flow.

To get a perspective on the technology a short rear-mirror view is suitable.

Using positive displacement, helical body, machinery in expansion application started already with Prof. Alf Lysholm (1938), the inventor of the screw compressor. Over the years an impressive amount of research and development has been carried out in various companies and institutions, leading to new creative solutions. McKay (1982) investigated geothermal energy production using screw, liquid/gas expanders. Platell, (1993) investigated screw expanders for steam cycles. Smith (1999), investigated a similar application as McKay using screw expanders working with flashing liquid.

In the commercial development integrated screw compressors and screw expanders were developed already in the 1950ies. Positive displacement screw machine gas-turbines, Q-motors, were in operation at company Svenska Rotor Maskiner AB (SRM) in Sweden as early as 1956. Fig. 2.
Olofsson (1993) developed the concept with expansion and compressor integrated in a singular rotor pair. Öhman (2000) as well as Brasz (2001) applied Olofsson's concept for use in the refrigeration cycle.

The development progress of an Expressor has been published several times ex Brasz (2003). According to that the difficulties with poor phase separation in the Expressor forced him into a solution along the principles of the Q-motors. (This means using two rotors for the expansion and two rotors for the compressor). Naturally this adds cost of complexity, piping, external separator etc as well as increased losses due to gas/liquid being forced to enter/leave the machine four times. Still efficiency was acceptable for its purpose.

The Phase Separator reported in this paper differs from the Expressor in two distinct ways.
Item 1. The port exiting liquid to the evaporator is situated downwards on the Phase Separator.

Item 2. The recompressor part of the rotors is designed for a low pressure ratio and to recompress all gas produced in the expansion process.

The effect of Item 1 is obvious as the opposite design creates a situation where the separation is blocked from gravitational forces as indicated by McKay (1982).

The effect of Item 2 is more complex:
Brasz (2003) recompress flash gas from evaporator pressure to condensing pressure. In order to preserve the energy balance between the produced power in the expansion process and the consumed power in the recompression process the Expressor has to be designed with a relatively small swept volume in the recompressor part. Consequently large amounts of flash gas cannot be recompressed but needs to pass the evaporator in order to be compressed in the main compressor.
The Phase Separator, using a small pressure ratio in the recompression, can be designed to recompress 100% of the flash gas produced in the expansion. Still of course the energy balance is preserved.

2. EQUATIONS

2.1 Determining Recompression pressure

Gas content after isentropic saturated liquidexpansion is: $\left(\frac{S'_1 - S'_2}{S''_2 - S'_2}\right) = X_{e.out.s} = \left(\frac{he.out.s - h'_2}{h''_2 - h'_2}\right)$

consequently isentropic expansion outlet enthalpy is: $h_{e,out.s} = h'_2 + (h''_2 - h'_2) x \left(\frac{s'_1 - s'_2}{s''_2 - h'_2}\right)$

If an isentropic expansion efficiency is introduced the outlet enthalpy is:

$h_{e.out} = h_{e.out.s} + (h'_1 - h_{e.out.s}) \text{ x } (1 - \eta_e)$

then the gas content after a real expansion is: $X_{e.out} = \frac{(h_{e.out} - h'_2)}{(h''_2 - h'_2)}$

Also the net produced expansion shaft power is $\dot{Q}_e = (h'_1 - h_{e.out.s}) \text{ x } \dot{m}_{tot} \text{ x } \eta_e$

The recompression shaft power is defined as $\dot{Q}_{RC} = \Delta h_{RC.s} \text{ x } \dot{m}_{RC} / \eta_{RC}$

As the purpose is to recompress all gas while obeying the energy equation $\dot{Q}_e = \dot{Q}_{RC}$ we get:

$\dot{m}_{tot} \text{ x } X_{e.out} \text{ x } \Delta h_{RC.s} / \eta_{RC} = \dot{m}_{tot} \text{ x } \Delta h_{e.s} \text{ x } \eta_e$

which can be expressed as $\Delta h_{RC.s} = \eta_{RC} \text{ x } \eta_e \text{ x } \Delta h_{e.s} \text{ x } \frac{1}{X_{e.out}}$ (Eq. I)

This equation defines the allowed enthalpy difference across the ReCompression to reach exactly X = 0 into the evaporator.

- Enthalpy change, isentropic expansion of saturated liquid: $\Delta h_{RC.s} = f\ (Gas, P_{RC}, P_2)$
- $\eta_{RC} \text{ x } \eta_e$ is empirically known from previous experience and correlated simulation models.
- Enthalpy change, isentropic expansion of saturated liquid: $\Delta he.s = f$ (Gas, P1, P2)
- Gas content after real expansion of saturated liquid: $X_{e.out} = f$ (Gas, P1, P2 and η_e)

From equation (I) it is trivial to determine the Recompression pressure P_{RC} vs running conditions.
In Fig. (3) a sample graph is shown for R134a.

2.2 Cooling Capacity

Compared to a conventional cycle the cooling capacity using a Phase Separator with 0 gas content is:

$$\frac{CAP_{PS.s}}{CAP} = \frac{h''_2 - h'_2}{h'_2 - h'_1}$$

2.3 COP

Cooling capacity using an ideal Phase Separator is: $CAP_{PS.s} = r\,(P_2) \times (\dot{m}_{tot} - \dot{m}_{RC})$

Required main compressor input power is then: $\dot{Q}_{C,PS} = \Delta h_c \times \dot{m}_{tot} - \Delta h_{RC} \cdot \dot{m}_{RC}$

Consequently the COP using an ideal Phase Separator can be written:

$$COP_{PS.s} = \frac{CAP_{PS.s}}{Q_{C.PS}} = \frac{r(P_2)}{\Delta h_c} \times \frac{(\dot{m}_{tot} - \dot{m}_{RC})}{(\dot{m}_{tot} - \frac{\Delta h_{RC}}{\Delta h_c} \times \dot{m}_{RC})} = COP_{PS.s} = \frac{r\,(P_2)}{\Delta h_c} \times \frac{\left(1 - \frac{\dot{m}_{RC}}{\dot{m}_{tot}}\right)}{\left(1 - \frac{\Delta h_{RC}}{\Delta h_C} \times \frac{\dot{m}_{RC}}{\dot{m}_{tot}}\right)}$$

however as $\dot{m}_{RC} = \dot{m}_{tot} \times X_{e.out}$ it can be simplified to:

$$COP_{PS.s} = \frac{r\,(P_2)}{\Delta h_C} \times \frac{(1 - X_{e.out})}{\left(1 - \frac{\Delta h_{RC}}{\Delta h_C} \times X_{e.out}\right)}$$

and as $\Delta h_{RC} = \Delta h_{RC.s}/\eta_{RC}$ and $\Delta h_c = \Delta h_{c.s}/\eta_c$ shows COP using a real Phase Separator

$$COP_{PS} = \frac{r(P_2)}{\Delta h_{c.s}/\eta_c} \times \frac{(1 - X_{e.out})}{\left(1 - \frac{\Delta h_{RC.s}}{\Delta h_{c.s}} \times X_{e.out} \times \frac{\eta_c}{\eta_{RC}}\right)} \qquad \text{(Eq II)}$$

The components of this equation are reasonably simple to determine:

- Total boiling enthalpy change: $r = f\,(Gas, P_2)$
- Isentropic enthalpy change, main compressor: $\Delta hc.s = f\,(Gas, P_1, P_2, T_{INC})$
- $X_{e.out} = f\,(Gas, P_1, P_2, \eta_c)$
- $\Delta h_{RC.s} = f\,(Gas, P_2, P_{RC})$ (According to Eq. I)
- Adiabatic main compressor efficiency η_c is an empiric or simulated value.

The above equations are not taking pressure/temperature losses in the system into account. Also subcooling/superheat is assumed to be zero.

3. SUGGESTED TERMS OF EFFICIENCY

Obviously the particular efficiencies for each sub-process: Expansion, Recompression and main compression are fundamental in order to describe the system. However, as a product, it is preferable to use efficiency terms covering the functionalities, not the sub-process.

The functionalities of the Phase Separator are: a) Separating the flash gas from the liquid condensate by recompressing it., b) Increasing the capacity of the system. c) Increasing the total energy-efficiency of the system.

3.1 Phase separation efficiency

Knowing the value of the gas content after the expansion and the total mass flow we can determine the necessary swept volume of the ReCompression. $\dot{V}_{RC} = V_{dpRC} \times n \times \eta_{vol}$.

Knowing the condensing pressure and the total mass flow we can determine the necessary swept volume of the Expansion $\dot{V}_e = \dot{V}_{d\,pe} \times n/f$.

Obviously the ratio $\dot{V}_{RC}/\dot{V}_e$ will affect the separation efficiency as a too large value would lead to ReCompression of liquid condensate and a too small number would lead to a situation where the ReCompression will not absorb all the flash gas and the gas content going into the evaporator will be larger than zero.
If fixed swept volumes are used only one specific running condition will lead to X = 0 according to theory. However, if X < 0 there will still be pure, saturated liquid entering the evaporator. (The excess liquid will be part of the ReCompression).

At any conditions and any Phase Separator the following is applicable:

Liquid flow, after expansion, ideal: $\dot{m}_{liq.out.s} = V_{dp.e} \text{ x } n \text{ x } \rho'_1 \text{ x} \left(1 - \frac{s'_1 - s'_2}{s''_2 - s'_2}\right)$ III

Gas flow after expansion, ideal: $\dot{m}_{gas.out.s} = V_{dp.e} \text{ x } n \text{ x } \rho'_1 \text{ x } \frac{s'_1 - s'_2}{s''_2 - s'_2}$ IV

Gas flow being recompressed, ideal: $\dot{m}_{RC.s} = V_{dpRC} \text{ x } n \text{ x } \rho''_2$ V

The resulting gas content reaching the evaporator is: $X_{IDEAL} = \frac{\dot{m}_{gas,out.s} - \dot{m}_{RC.s}}{\dot{m}_{gas.out.s} - \dot{m}_{RC.s} + \dot{m}_{liq.out.s}}$

Inserting III – V we get: $X_{IDEAL} = \dfrac{\rho'_1/\rho''_2 \text{ x } \frac{s'_1 - s'_2}{s''_2 - s'_2} - \frac{V_{dpRC}}{V_{dpe}}}{\rho'_1/\rho''_2 - \frac{V_{dpRC}}{V_{dpe}}}$ VI

which is gas content levaling an ideal Phase Separator for the evaporator. The terms in eq. VI are well defined and only dependent on running conditions and the geometric properties of the phase separator.

If the obtainable gas content is measured using a given Phase Separator we can define:

$$\eta_{PS} = \frac{X_{IDEAL}}{X_{measured}}$$ **[Phase Separation Efficiency]**

This term takes all inefficiencies of the Phase Separator such as throttle losses, leakages, thermal losses and inefficient physical separation, into account.

3.2 Capacity increase efficiency

If we use the ideal gas content in Eq. VI the capacity becomes: $CAP_{PS.s} = r(p_2) \text{ x } (1 - X_{e.out.s}) \text{ x } \dot{m}_{compressor}$

Measuring cooling capacity and mass flow we can define:

Capacity Increase Efficiency:

$$CIE = \frac{CAP\,(measured) / (h''_2 - h'_2) \text{ x } \dot{m}_{compressor}}{CAP_{PS.s} / (h''_2 - h'_1) \text{ x } \dot{m}_{compressor}} = \frac{\frac{\text{Measured Cooling Capacity}}{\dot{m}_{compressor}}}{r(P_2) \text{ x } (1 - X_{e.out.s})} \quad I$$

3.3 COP Increase Efficiency

COP of a system with an ideal Phase Separator, Fig. (4): $COP_{PS.s} = \dfrac{r(P_2) \text{ x} \left(1 - \frac{\Delta he.s}{\Delta h_{RC.s}}\right)}{(\Delta hc.s - \Delta he.s)}$

and an ideal conventional system has a COP of: $COP_{conv} = \dfrac{\Delta h2}{\Delta hc.s}$

To determine the utilization of the potential increase in COP for a real Phase Separator we can define a term

$$COPIE = \frac{COP_{meas} - COP_{conv}}{COP_{PS.s} - COP_{conv}} = COPIE = \frac{COP_{meas} - \Delta h_2/\Delta hc.s}{\dfrac{r\,(Pz)\ x \left(1 - \dfrac{s'_1 - s'_2}{s''_2 - s'_2}\right)}{(\Delta hc.s - \Delta he.s)} - \Delta h_2/hc.s}$$

[COP Increase Efficiency]

All terms are referring to measured COP the studied system and well defined variables of state as a function of condensing/evaporating pressures.

4. TESTING

Testing of a Phase Separator is challenging and can be conducted either in a complete refrigeration system as in Brasz (2003) or in a hot gas test stand. In the first type of measurements reliable data on COP, capacity and quality of the refrigerant to the evaporator are reasonably easy to obtain. However, the range of operating conditions is limited and unexpected phenomena has a tendency to ruin all test results.

In the second type of measurements all results are based on the measurement of the different flows. It allows for a wider range of operating conditions as well as better possibilities to sort out unexpected phenomena. Good data on refrigerant quality and parameter variation trends are obtainable. The test results referred to in this report are produced in a hot gas test stand, see Fig. 5.

4.1 Phase Separator

No. of helical rotors:	2
Lobe combination:	5+7
Rotor Profile:	FAS
L/D:	1.4
ODM:	96 mm
Wrap angle:	300 degree
Vdp_{RC} (Displacement, ReCompression):	0.5 litre/rev
Vdpe (Displacement, Expansion)	0.04 – 0.15 l/rev (variable)
VI_{RC} (Volume ratio, ReCompression):	1.5
VIe (Volume ratio, Expansion):	3 – 11 (variable)

4.2 Running conditions

Refrigerant:	R134a
T_1 (Condensing temperature):	38°C
T_2 (Evaporator temperature):	8.5 °C
Speed range:	0 – 4000 rpm

4.3 Purpose of testing

Determine if a gas content of 0% (X =0.0) into the evaporator is obtainable using a Phase Separator.

4.4 Method

Liquid refrigerant was fed to the Phase Separator through a mass-flow meter. The refrigerant leaving the Phase Separator is led to a mixing chamber where it is heated by hot gas from the main compressor. After the mixing chamber, suitably superheated, the flow is measured. The refrigerant flow leaving the ReCompression port of the Phase Separator is measured by using a calibrated main compressor as a double check. (Obtained by running each test point with disconnected Phase Separator but identical running conditions for the compressor).
Using the measured flows and the state conditions a series of energy balances produces the analysis data.

4.5 Test results

In Fig. (6) the test results are shown for max and min expansion displacement, Vdpe.

4.6 Observations and conclusions from the tests

The results prove that a Phase Separator, properly designed, will produce pure, saturated liquid to the evaporator.

An observation supporting the conclusion from the test was the behavior when the Phase Separator attempted to produce a gas content less than zero ($X < 0.0$). Obviously that is not possible so when all the gas produced in the expansion was ReCompressed and the ReCompression had excess capacity the flow through the low-pressure discharge port was reversed. This was evident as the hot gas flow to the mixing chamber was larger than the flow leaving the mixing chamber for the main compressor. Also the temperature readings in the line between the low-pressure discharge port and the mixing chamber increased abruptly after $X = 0$ had been achieved.
This phenomena could not occur in a full refrigeration cycle.

5. DESIGN DISCUSSION

5.1 ReCompression built-in volume ratio (Vi_{RC})

The V_i in the recompression, 1.5, was designed to fit the integration with an existing screw compressor intermediate pressure port. The resulting very large discharge port increases the sensitivity to rotor speed and ReCompression pressure. Improvements can be made regarding the Vi_{RC}.

5.2 Expansion built-in volume ratio (Vi_e)

It has been known to screw expander designers for over 40 years that the V_i has to be significantly smaller than the flange-to-flange volume ratio of the working media, see Platell (1993). (This fact has recently become patented by Smith and Stocik (1998). This is particularly important when the flange-to-flange-volume ratio of the media is large. (Above 6).

The tested Phase Separator has a variable built-in expansion volume ratio Vi_e in a range from 3 to 11 as described in Öhman (2000). Also the swept volume of the expander Vdpe is variable in a range from 0.04 – 0.15 l/rev.
The results shown in this report cannot be used to determine "optimal" built in volume ratio of the expansion due to significant throttle losses in the Vi-variation mechanism.

5.3 Potential efficiencies of a Phase Separator Expansion

Liquid condensate expansion with screw machines has only been studied using a very limited number of designs why the conclusions so far are questionable. However, refrigerant gas screw expanders with liquid condensate injection have been used in the industry. Typical peak efficiencies with R134a and pressure ratios below 4 is 87%. (Isentropic expansion efficiency of Screw Expander M31, see fig. 7).
Simulation models for liquid condensate expansion indicate peak efficiencies of 85 % as possible to achieve with screw expanders. (This will however be strongly dependent on the method of maintaining the proper, and changing, liquid flow to the evaporator as a function of the heat load).

5.4 ReCompression

Low pressure ratio screw compressors generally have a low peak efficiency below 80 %. The combination of remains of liquid and a relatively low tip-speed further reduce the peak efficiency.
As an engineering estimate a peak isentropic efficiency of 70% can be seen as reasonable.

5.6 Combined effects

In the case of ReCompressing flash gas to the condensor pressure Öhman (2000) and Brasz (2001) the combined efficiency is η_e x η_{RC} so a peak efficiency of 60% should be achievable. Using a lower ReCompression pressure the result becomes more complex. In fig. 8 COP improvements for R134a has been simulated with the above peak efficiencies.

5.6 Potential impact on refrigeration system fundamentals

Apart from the obvious benefits of better COP and higher cooling capacity for a given compressor size the Phase Separator technology permits the use of novelty type evaporators. Micro-channel heat exchangers often suffer from reduced efficiency when the refrigerant distribution is unbalanced. The flash gas produced in a conventional throttle valve constitute a severe limitation in this respect. Using the gas-free liquid produced by a Phase Separator significantly simplify the use of micro-channel evaporators leading to an even more interesting potential for improving the refrigeration system.

REFERENCES

Brasz, J., 2001, Single Rotor Expressor as Two-Phase Flow Throttle Valve Replacement, US- Patent No.: 6.185.956

Brasz, J., 2003, Screw-expressor testing on an R-134a chiller-efficiency liquid carry-over, and chiller benefit, *IMechE Conference Transactions of the International conference on compressors and their systems,* City University, London, p. 325 – 338.

Lysholm, A., 1938, Rotory Compressor. US-Patent No.: 2.111.568.

McKay, R., 1982, *Helical Screw Expander Evaluation Report,* JPL Publication 82-5, Pasadena, p. 186.

Olofsson, H., 1993, Machine For a Gaseous Medium, US-Patent No.: 5.192.199.

Platell, P., 1993, *Displacement expanders for small scale cogeneration*, Licentiate thesis, Dept. of Machine Design, Royal Institute of Technology, Stockholm, p. 107.

Smith, I. K., Martin, P.R. 1989, The Trilateral Flash Cycle for Power Recovery from Hot Dry Rock. *Int. Conf. on Hot Dry Rock Geothermal Energy, Camborne, Cornwall, UK,* Robertson Scientific Publ.

Smith, I. K., 1999, Review and Overview of Two-Phase Screw Expanders, *IMechE Conference Transactions of the International conference on compressors and their Systems,* City University, London, p. 95 – 104.

Smith, I., Stocik, N., 1998, Deriving Mechanical Power by Expanding a Liquid to its Vapour, US-Patent no.: 5.833,446.

Öhman, H., 2002, Cooling Apparatus and Method For Increasing Cooling Capacity, *PCT-Gazette*, Dec.-00, p. 87.

NOMENCLATURE

s'_1	saturated liquid entropy, condensor
s''_2	saturated gas entropy, evaporator
s'_2	saturated liquid entropy, condensor
h'_1	saturated liquid enthalpy, condensor
h'_2	saturated liquid enthalpy, evaporator
h''_2	saturated gas enthalpy, evaporator
ηRC	adiabatic efficiency, Compression
$\dot{m}_{tot}$	total massflow, condensor
Gas	Refrigerant used
hRC.s	enthalpy difference across isentropic Re-Compression
he.s	enthalpy difference across isentropic Expansion
Gas	refrigerant type
P2	pressure, evaporator
P1	pressure, condensor
CAP	cooling capacity, conventional cycle.
$\dot{m}R_c$	mass flow being ReCompressed
Δh_c	enthalpy difference, real main compressor
Δh_{RC}	enthalpy difference, real ReCompression
COP	cooling capacity/main compressor power.
T_{INC}	main compressor inlet temperature
η_C	adiabatic efficiency of main compressor
n	rotational speed
η_{vol}	volumetric efficiency
f	filling factor, Expansion
ρ'_1	density, saturated liquid, condensor pressure
ρ''_2	density, saturated gas, evaporator pressure
$X_{MEASURED}$	Measured leaving real Phase Separator for evaporator
$\dot{m}_{compressor}$	mass flow through main compressor
Δh_2	$h''_2 - h'_1$
COPmeas	measured COP

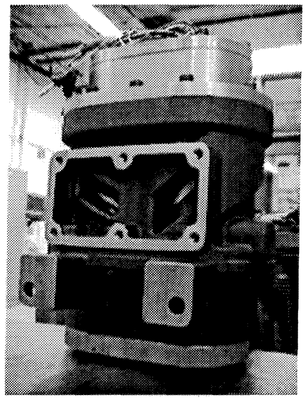


Fig 1.

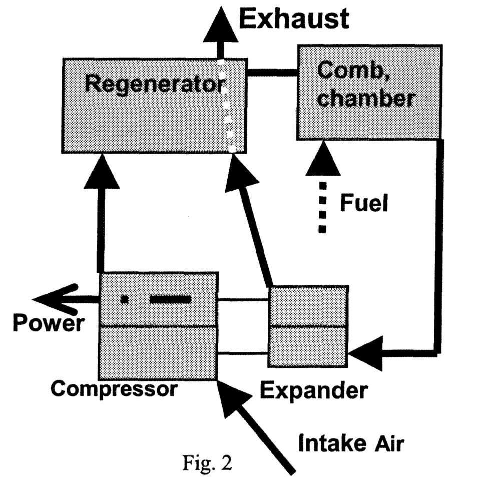


Fig. 2

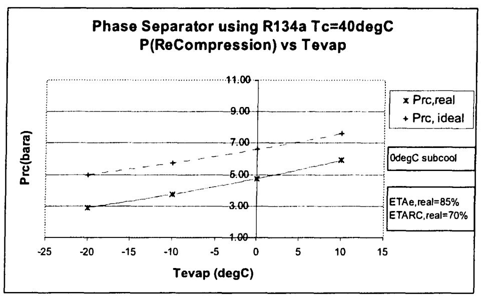
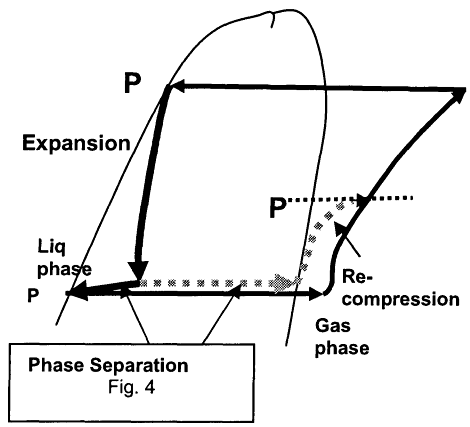


Fig. 3



Phase Separation
Fig. 4

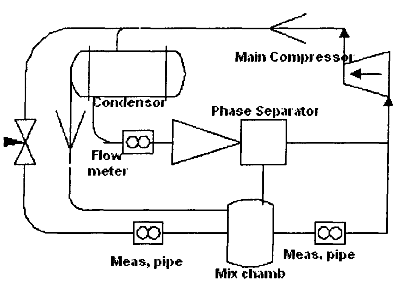


Fig. 5

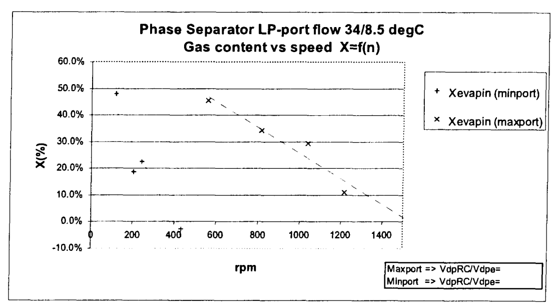


Fig. 6

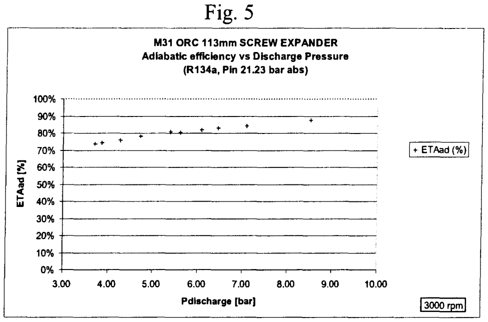


Fig. 7

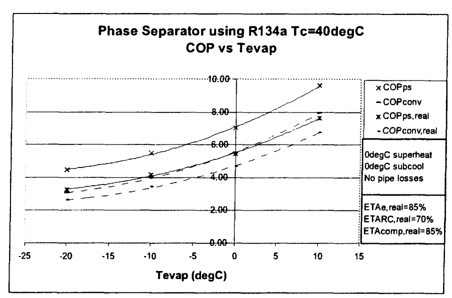


Fig. 8