1986

Influence of the Expansion Valve on the Evaporator Performance

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INFLUENCE OF THE EXPANSION VALVE ON THE EVAPORATOR PERFORMANCE.

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INTRODUCTION

The possibility of micro-electronics most probably will lead to a new generation of expansion valves, the properties of which will not be limited by the mechanical constructions as it is the case with expansion valves of usual design. Which improvement in superheat control do we seek, which improvements could be reached?

Many investigators already mentioned the possible influence of the setting and the operation of a thermostatic expansion valve on evaporator performance. The expansion valve of usual design has properties that can be compared with those of a proportional controller. There is a proportional band, the superheat will always differ from the set value, the difference being dependent on the positioning of the valve needle (or evaporator load).

By introducing an integrating action, this difference can be reduced substantially.

By using a mathematical model, the influence of the superheat setting for an expansion valve of usual design and one with integrating action was investigated.

The model comprised air cooler, expansion valve, a capacity controlled compressor and a water-cooled condenser.

MODEL

Using a mathematical model instead of an experimental stand to study special phenomena of a refrigerating cycle has some advantages:

- the process conditions can be adjusted instantaneously and exactly to the desired values,
- the process conditions can be measured exactly.

Controlling the process conditions in a real test stand can be a rather time consuming task, or might require a test stand better equipped than the one available, especially when the influence of only one parameter has to be studied and all others have to stay constant. For instance the possibility of keeping the evaporator outlet air temperature at a constant value as in the measurements in this study is a rare one and will normally not be available.

The same applies for instance to measuring the exact superheat in the suction pipe of an evaporator with the intention to use it for the control of the massflow. This requires the measurement of the suction pressure and the transformation of the signal into one describing the saturation temperature. In a mathematical model the information can be obtained without any special equipment.

The results of the model were backed up by experiments in a test room.

The evaporator is a standard air cooler with a nominal capacity of 8 Kw at an evaporation temperature of -10 °C and an inlet air temperature of 0 °C. It has six parallel circuits with the pipes in in-line formation. Each circuit has a length of 15.25 m.

The air in the test room was dried with a second evaporator/air cooler in order to obtain a dew point below the surface of the test evaporator. As a result no frost formation or condensation took place during the test so it was not necessary for the model to account for these effects.

Experiments however were necessary in order to validate the model and also for the determination of coefficients in the mathematical model which could not be obtained sufficiently reliable from theory. Such coefficients are for instance the heat transfer coefficient and the pressure drop of the two phase flow region. Therefore they had been calculated out of measurements on a broad range of operating conditions, which included heat loads from 1 to 10 Kw. These values have been used to modify existing correlations found in literature [1],[2].

These modified correlations proved to account for the model a good superheat temperature and maximum possible heat load at given conditions.
The necessity of modification is a disadvantage for the use of the model, but proved inevitable because of the complexity of non adiabatic two phase flow in a series of horizontal pipes connected with u-bends.
To study the possibility of a non uniform distribution of air or refrigerant along the parallel pipes of the evaporator, the model could be split up into two parts, each describing a set of parallel pipes. The distribution of the air along these has to be inputed, the distribution of the refrigerant could be calculated assuming equal qualities of the refrigerant flows to each circuit. The refrigerant flows then get such values that the pressure drops along all evaporator circuits are equal.
The water cooled condenser is able to provide a constant condenser pressure and subcooling temperature, to avoid secondary effects in the evaporator performance.

SIMULATION RESULTS WITH INTEGRATING ACTION

Simulations with the mathematical model of the evaporator equipped with a thermostatic expansion valve with integrating action were made, for several constant setpoints of the superheat. Figure 1 shows in case of an uniform air and refrigerant flow around the parallel circuits, the necessary evaporation temperature and the length of the evaporator used for evaporation, as a function of the demanded heat load.

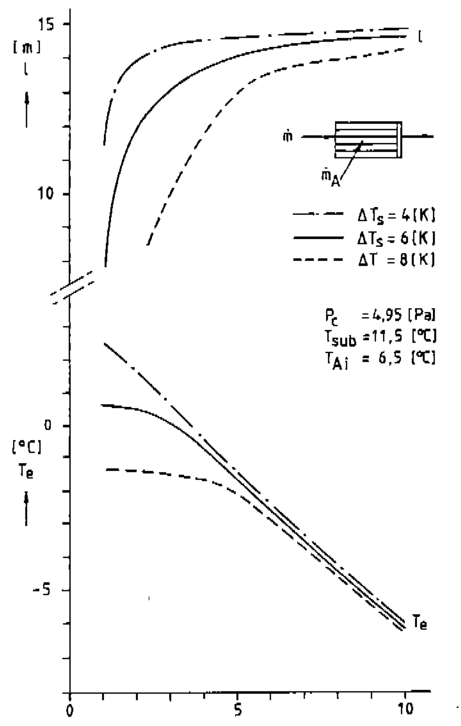

figure 1. Evaporation temperature and length of the evaporation region as a function of the heat load. Simulation results of an evaporator with a TEV with integrative action for three different setpoints.

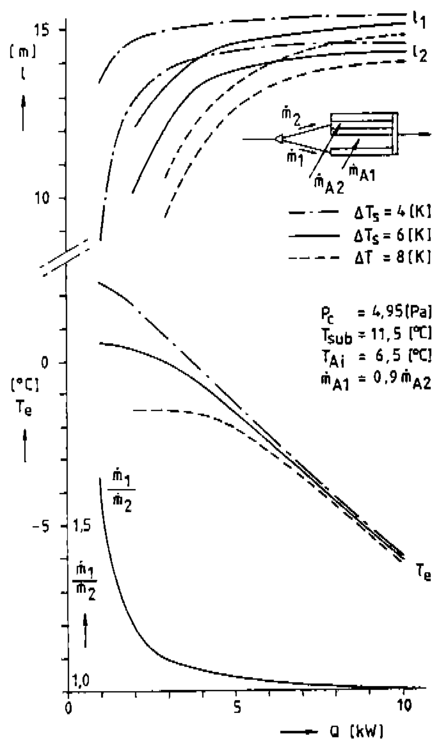

figure 2. Evaporation temperature and length of the two series of evaporation regions and, for the case of 4 K superheat, the ratio of the refrigerant massflows as a function of the heat load. Simulation results of an evaporator with a TEV with integrative action for three different setpoints.

The evaporator temperature indicates the influence of the setpoint on the compressor work. With a lower temperature the pressure ratio across the compressor will be higher, and so the compressor work.
The available length of the evaporator tube that remains for evaporation is the direct cause of this evaporation temperature, because of the higher heat transfer in the evaporation region compared with the superheat region. Therefor this length of the evaporation region should not become smaller then necessary. On the other hand when the evaporation region approaches the evaporator outlet too close, liquid refrigerant might quit the evaporator together with the superheated vapour and disable the measured signal of this superheated vapour temperature [3]. The expansion valve would become instable unless safety was sacrificed for the cause of stability by increasing the time constant of the controller. Consequently the control of the expansion valve should not decrease the length of the superheat region below a certain value.
Assuming a necessary length of the superheat region of at least 0.6 m, the setpoint of 4 K in figure 1 proves to be useful when the heat load stays less then 6 Kw. Both the setpoints of 6 and 8 K ensure the required distance between the evaporator region and the evaporator outlet. On the other hand below heat loads of respectively 3 and 5 Kw the remaining lengths for evaporation decrease quickly resulting in a lower nécessary temperature and an increase in the compressor work of respectively 3.3 and 15.3% compared with a superheat of 4 K at a heat load of 3 Kw.
Resulting the setpoint of 6 K seems to be a reasonable choice, specially when the heat load does not sink below 3 Kw.

Figure 2 shows the same results as figure 1 but the evaporator now has no equal distribution of the air and the refrigerant flow along the parallel pipes of the evaporator. The lower pipes have a 10% lower massflow of air while at lower heat load the refigerant distribution becomes uneven too.
The setpoint of 4 K now can provide only the savety zone of 0.6 m at a heat load lower then 2 Kw. For the same reason the heat load with 6 K should not exceed 5 Kw, while at 3 Kw the evaporator gets a non effective filling.
With a setpoint of 8 K which had no consequences at the high side, the evaporator will be used ineffectively below 5 Kw.
The compressor work increases at 3 Kw with 12.2% compared with a setpoint of 6 K (a comparison with 4 K is not possible because then the evaporator would not work stable at 3 Kw).
Resulting, the setpoint of 8 K will be the only usefull one of the three possibilities, however with no good properties when the load decreases below 5 Kw.

SIMULATION RESULTS WITHOUT INTEGRATING ACTION

Controlling the evaporator with an integrating TEV with a constant superheat as setpoint proved, according the calculated length of the superheat region and the evaporation temperature necessary to obtain the demanded heat load, not to be a good method.
For the cause of stability this agrees with the conclusion of Huelle [4] that the minimum superheat that could be used as stable control signal for a TEV did not have a constant value as a function of the heat load.
Thereby also for the case of keeping an evaporator filled in an effective way with evaporating refrigerant, the wish to realize a special superheat was found to have a negative effect, specially when lowering the heat loads.
That a conventional TEV is not able to realize a constant superheat at different heat loads can therefor even be an advantage.
This can be seen in figure 3 an 4 where the evaporation temperature, the length of the superheat region and the realized superheat are plotted for an evaporator with a conventional TEV, for three different settings of the static superheat. Note, the number indicating the setpoint is the position of the adjusting screw of the TEV, and not the value of the static superheat, which is not constant.
In the case of uniform air and refrigerant flows, figure 3, setpoint 2.66 gives a good result over the whole working range. Setpoint 3.00 gives a little better filled evaporator with circa 1% lower compressor work, but will lead to hunting when the heat load decreases below 5 Kw.
With the non uniform flows in figure 4 only setpoint 2.33 can work stable, but with a reasonable filling of the evaporator on the whole range of heat loads.

So a conventional TEV can, because of its changing "setpoint", have a broader working range then a TEV with an integrating action used to keep the superheat constant at all heat loads.
The problem however is that chosing the optimal setpoint for a range of process conditions requires an analyses, for instance with a mathematical model of the evaporator and its expansion valve.

To make an integrating action useful there should be the possibility to change the superheat according the process conditions and to have the necessary information to know what is the optimal superheat at each process condition. Another theoretical possibility as using for instance the presence of liquid refrigerant nearby the evaporator outlet or even an admissible amount of liquid refrigerant leaving the evaporator as a signal to control the expansion valve [3] would keep the evaporator optimally filled, independent of, and thus without the necessity of information about, the heat load or any other possible influence. Such feed control would require no hand setting at all.

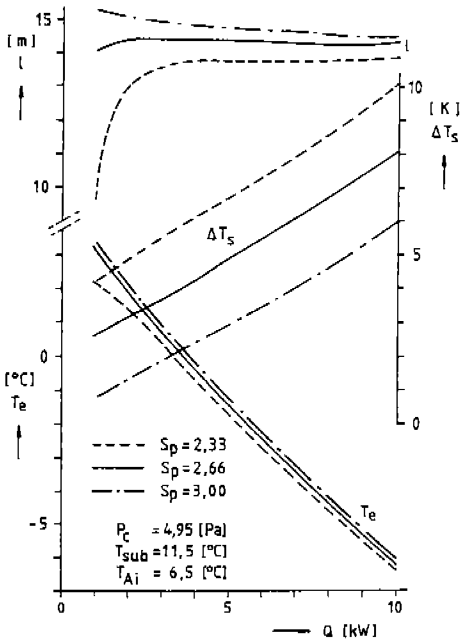

figure 3. Evaporation temperature and length of the evaporation region and superheat as a function of the heat load. Simulation results of an evaporator with a conventional TEV for three different setpoints.

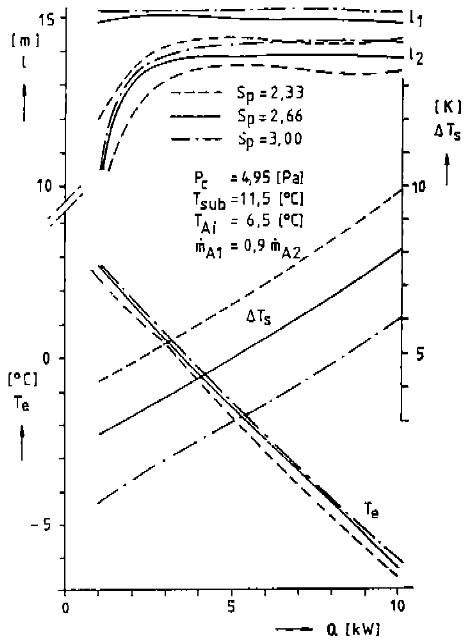

figure 4. Evaporation temperature and length of the two series of evaporation regions and superheat as a function of the heat load. Simulation results of an evaporator with a conventional TEV for three different setpoints.

ABOUT USING A CONSTANT MINIMUM LENGTH OF THE SUPERHEAT REGION

The necessary length of the superheat region to prevent liquid refrigerant waves to leave the evaporator has been set on 0.6 m. However about this subject no exact information is available, but based on the results of Wedekind [7] the assumption of the constantness can be chequed.
Wedekind who studied the oscillations of the end position of the evaporation region, described the amplitude $\Delta l(t)$ of these oscillations as :

$$\Delta l(t) = \frac{(1-\alpha_i)\,\dot{m}\,r}{(1-\bar{\alpha}_o)\,\dot{q}\,\pi\,d}\,\bar{x}_1(t) \qquad (1)$$

With
α_i = void fraction at the evaporator inlet [-]
$\bar{\alpha}_o$ = mean nonoscillatory void fraction [-]
$\dot{m}$ = total inlet massflow of refrigerant [-]
r = enthalpy difference vapour-liquid refrigerant [kJ/kg]
$\dot{q}$ = evaporator heat flux [kW/m²]
d = inside diameter evaporator pipe [m]
$\bar{x}_1(t)$ = mean flow quality perturbation [-]

where the mean flow quality perturbation was thought only to be influenced by the evaporator geometry.
Because the oscillations are only of interest when they reach the outlet of the evaporator, the equation can be simplified for our case. The energy balance of the evaporating refrigerant:

$$\dot{m}\,r\,(1-x_i) = \dot{q}\,\pi\,d\,l_e \qquad (2)$$

with
x_i = inlet flow quality [-]
l_e = length of the evaporation region [m]

gives combined with equation (1):

$$\Delta l(t) = \frac{(1-\alpha_i)}{(1-\bar{\alpha}_o)(1-x_i)}\,\bar{x}_1(t) \qquad (3)$$

With $\bar{x}_1(t)$ constant as a function of the evaporator configuration and the ratio $(1-\alpha_i)/(1-\bar{\alpha}_o)$ according figure 6 in [7], the biggest oscillations of the mixture vapour transition point on the range of 1-10 Kw are found at the smaller heat loads. At maximum heat load the amplitude of the oscillations is then only a 12% lower. This is in our case less then 0.5% of the length of the evaporator.

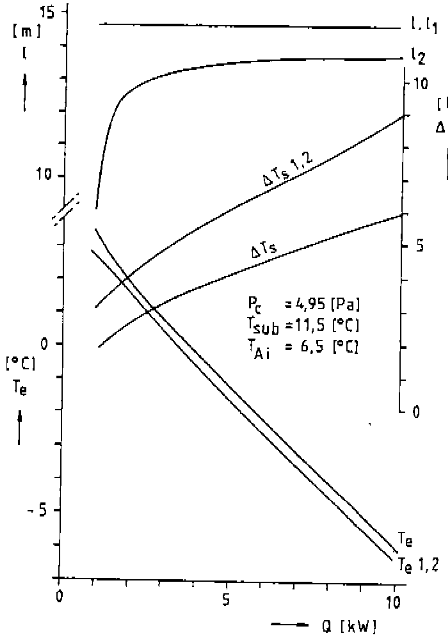

figure 5. Evaporation temperature, length of the evaporation region(s) and superheat as a function of the heat load. With and without uniform refrigerant and air flow along the evaporator pipes. Simulation results of an evaporator controlled in such a way that the length of the (shortest) superheat region is kept at a constant value of 0.6 m.

The change of the superheat region necessary to prevent liquid to leave the evaporator, in steady conditions, is only of importance when the evaporator inlet quality approaches to zero.
In that case the amplitude of the oscillations could almost be doubled.
Above an inlet quality of 0.05 however as was the case in our measurements and simulations, using a constant value as the minimum allowed length of the superheat region was quite acceptable.
In figure 5 then for both uniform and non uniform flows the evaporation temperatures, the superheats and the lengthes of the evaporation region are plotted in the case of a constant length of the (shortest) superheat region of 0.6 m.
These superheats would, based on the previous assumptions, be the optimal setpoints for a TEV with integrating action.

INFLUENCE OF THE SUPERHEAT ON THE COP

In the case of chosing the setpoint of the superheat control, with or without integrating action, an interesting fact to know would be which values the evaporation temperature should have to keep the heat load constant for different values of the superheat. Thanks to a special possibility of the evaporator test stand this effect could be studied in practice as well as with the simulation model.
Namely the possibility of automatic compressor speed control based on a thermocouple signal.

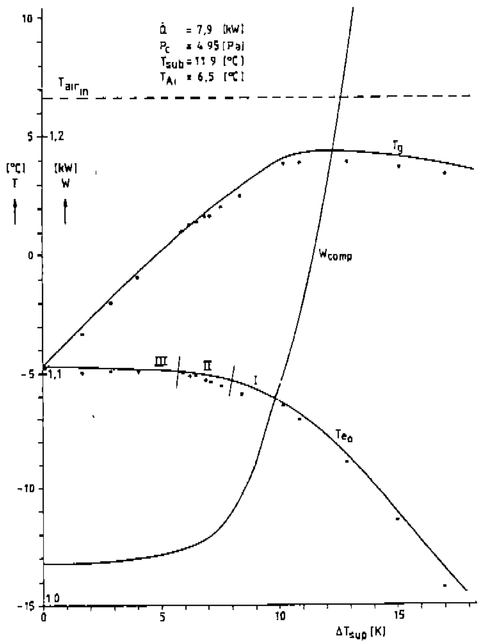

figure 6. Saturation and vapour temperature at the outlet of the evaporator measured (points) and simulated (lines), and the calculated compressor work, necessary to keep the heat load constant, as a function of the superheat.

Using this to control the air temperature difference across the evaporator and with a separate control of the heaters in the test room to keep the air temperature at the inlet of the evaporator constant, and so the massflow of air through the evaporator, the heat load could be controlled.
With constant refrigerant conditions before the expansion valve, the test stand was useful to study the influence of purely the superheat on the COP.
Figure 6 shows the saturation and the vapour temperature at the outlet of the evaporator necessary to realize the heat load of 7.9 Kw for a series of superheats.

The points in region I and II could be obtained by changing the static superheat of the TEV. In region III the TEV could not work stable, so here a hand controlled expansion valve has been used.
It was found that in region I the refrigerant left the evaporator as superheated vapour. In region II the vapour was accompanied by droplets as could be noticed on the measured vapour temperature, while in region III liquid waves started to leave the evaporator.
Both simulations and measurements show with increasing superheat first a slow but above 10 K a quick decreasing evaporation temperature. Consequently the compressor work starts to increase slowly too and later on rather quickly.
As a result decreasing the superheat from 10 to 8 K reduces the compressor work with 5.4%, from 8 to 6 K with 1.6%, and from 6 to 4 K only with 0.4% .
This agrees with one of the conclusions of [4], that the heat load of the evaporator would not be increased considerably with decreasing the superheat below the minimum stable signal.
So a superheat lower then 6 K, which was here the minimum superheat with a stable signal useful for the expansion valve, does not give a better energetic performance of the evaporator. On the contrary, with a lower superheat the amount of refrigerant that leaves the evaporator unevaporated as liquid and, in the good case, evaporates later on in the circuit, has to be compressed too.
Reichelt [5] reported a massflow of 4.5% of liquid leaving the evaporator at 1 K. What means that bringing the superheat back from 6 to 1 K will not lead to a decrease of the compressor work with 0.5% but to an increase of 4% .
The effects of changing the superheat on the compressor work with respect to the influence of unevaporated refrigerant depent on the amount of it in the superheated vapour. [5] and [6] seem to indicate that the mass percentage of the liquid droplets, when no waves are present any more in the suction vapour flow, is less then 0.3% . In that case this influence is according figure 6 less then that of the evaporation pressure and superheat temperature and thus a higher superheat will have a negative effect on the ratio heat load/ compressor work. Resulting a superheat close, from the upper side, to the one that still gives the (minimum) stable superheat will be the best solution with respect to the COP.

CONCLUSIONS

- A TEV equipped with an integrating action with a constant setpoint will not control an evaporator in an optimal way with respect to stability and efficiency in case of non constant process conditions.
- In case of a well chosen combination of a conventional TEV with setpoint the evaporator can even better be controlled.

Use of a mathematical model will help to find such a combination, but catalog data as provided for in the usual way by expansion valve manufacturers are insufficient for this purpose.
- The optimal superheat with respect to the energy, was found to be the smallest superheat that still supplied a stable control signal.
- A lower superheat, apart from the problem of stability, will give a lower COP because of the unevaporated refrigerant that leaves the evaporator.
- A higher superheat first gives a slow, later on a quick decrease of the COP, resultng from the uneffective use of the evaporator surface.
- To make an optimal use of micro-electronics a precise knowledge of the physical processes to be handled is necessary.

ACKNOWLEDGEMENT

This research at the Delft University of Technology has been sponsored by the "Stichting Technische Wetenschappen".

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INFLUENCE DU DETENDEUR A LA PERFORMANCE DE L'EVAPORATEUR.

Résumé.
On a étudié l'influence du détendeur thermostatique sur la performance de l'évaporateur. Dans cet étude l'évaporateur était un refroidisseur à air. Un détendeur usuel (qui ressemble à un régulateur à action proportional) a été comparé avec un étendeur disposant d'un action intégratif.
L'action intégratif permet le détendeur d'entretenir une surchauffé constante indépendent de changements instantanés de la capacité frigorifique.
On a envisioné des situations avec et sans distribution égale de l'air et du fluide frigorigène dans l'évaporateur.
Enfin on a étudié l'influence du réglage de la surchauffé sur l'efficacité de la machine frigorifique, en cas d'une valeur donnée constante de la capacité frigorifique. Pour ce cas, on a comparé les résultats de simulation à l'aide d'ordinateur avec des résultats d'expériments pratiques.

INFLUENCE OF THE EXPANSION VALVE ON THE EVAPORATOR PERFORMANCE.

Summary.
A study has been made on the influence of a thermostatic expansion valve on the performance of an evaporator. The evaporator was an air cooler.
An expansion valve of usual design (which looks like a proportional controller) has been compared with an expansion valve with an integrating action. With the integrating action the evaporator can, independend of changes in its heat load, be controlled at a constant superheat.
Situations with and without equal distribution of the air and the refrigerant in the evaporator have been envisioned.
Finally the influence of the superheat on the C.O.P. has been studied, in case of a constant heat load. In this case the results of the computer simulation have been compared with experimental results.