Dynamic Modelling of a Closed-loop, 2-Phase Compression System for Heat Pump Applications

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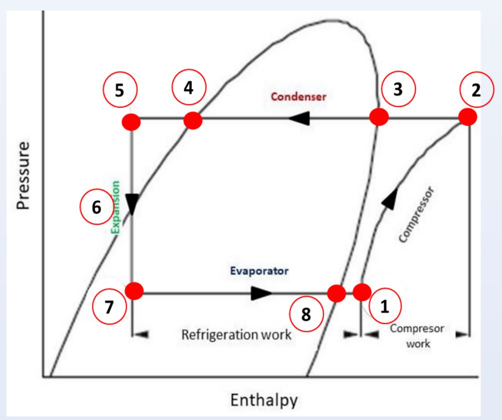
Introduction

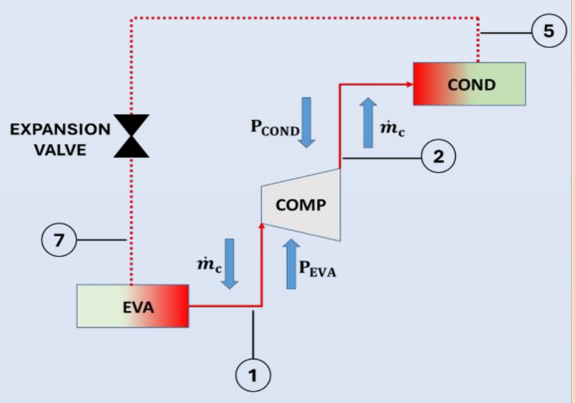
Heat pumps that use dynamic compressors, are an attractive solution to the increasing demand for highly efficient & low emission thermal requirements of residential and industrial sectors. These systems are subjected to compression system instabilities, which require special attention due to their closed loop, multi-phase, real gas behaviour. The designers of such system must know in advance when these instabilities start, how strong they would be, and how they can be mitigated. Herein lies the necessity to develop dynamic models which can predict different complex phenomenon from the perspective of the closed loop system.

Dynamic Modelling of Compressors: State of the Art

Modelling of compression systems to capture surge and stall instabilities have been explored by many researchers. The dynamic model developed by Moore and Greitzer is a widely employed technique for open compression systems, such as those for axial compressors in aircraft engines. But for heat pump systems, the problem arises due to the fact that **the compressor is placed between evaporator and condenser**, where phase transition occurs. This also means that **the inlet and outlet of the compressor face transient boundary conditions**.



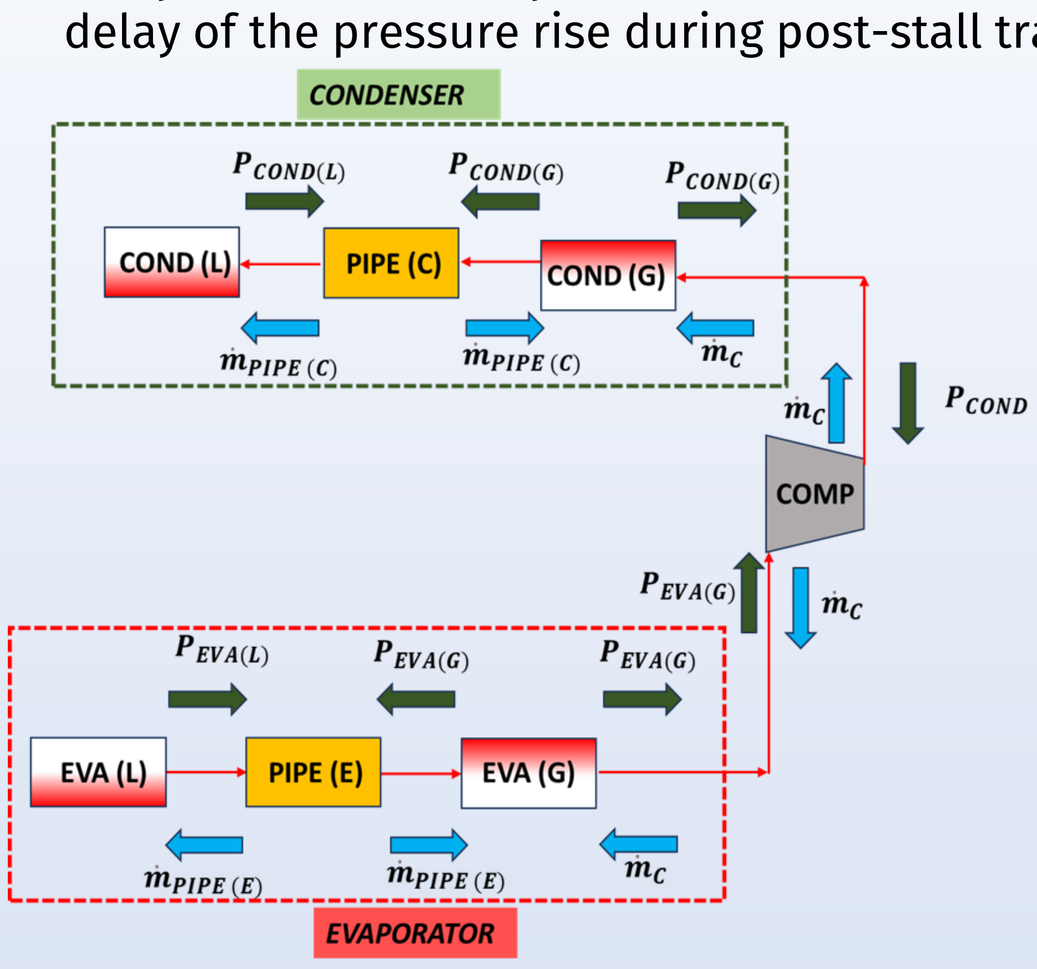
Attempts have been done to model the sensible and latent heat exchanges through purely theoretical method. But the presence of **real gas effects** in the system increases the complexity of the model.



In the model described here, **empirical data at steady state operation** of the compression system is used to represent the heat exchangers, while a **simple dynamic model**, similar to Moore – Greitzer model, is used for the compressor.

Model Schematic

Each of the heat exchangers are represented by 2 plena and a pipe. One plenum represents the liquid part of the heat exchanger, and the other plenum represents the gas spring effect. The pipe component represents the momentum in the various ducts between the heat exchanger and the compressor. The compressor solves the momentum equation, and the time delay of the pressure rise during post-stall transients.



a) The liquid plena uses **empirical heat transfer data**, as a function of **instantaneous pipe mass flow**, to calculate the **instantaneous liquid pressure** inside the plenum.

b) The gas plena uses the **instantaneous pipe mass flow rate** and **instantaneous compressor mass flow rate** to calculate **instantaneous gas pressure** information at the compressor boundary.

c) The compressor uses **instantaneous pressure information** from both plena, and **compression system characteristics** to calculate **instantaneous compressor mass flow.**

Assumptions

1) The flow in equivalent ducts is considered incompressible.
2) Assuming that a given rate of change of mass flow produces the same unsteady pressure difference in the model as in the actual duct and matching the equivalent duct area to a characteristic area, an equivalent duct geometry can be determined.
3) When the stall limit is reached, and the flow field becomes unstable, there is a time lag between the onset of instability and the fully developed stall pattern.
4) This time is equal to or longer than the order of several rotor revolutions (N_R).
5) This time delay (τ) is employed in the way the compressor pressure rise (C) is calculated from the steady-state characteristics (C_{SS}).

Model Equations

The governing equations for the model include **unsteady, inviscid, irrotational form** of **continuity, momentum and energy equations** written in forms convenient for dynamic modelling. The energy equation in liquid plena of condenser and evaporator contains a **pseudo specific heat** c_p^* which is determined from experiments at the last steady operating point of the system. The **compressor** solves a **momentum equation**, with instantaneous pressure rise (C) calculated from the **steady state characteristic curve** of the system and **time delay**, based on Moore – Greitzer model assumptions. Reasonable assumptions are made about the level of liquid in the heat exchangers at the last steady point, both with visual observations and prior experience.

COND_L:

$$\frac{d\left((m_{COND\_L}\, C_V + m_{metal\_COND}\, C_{metal})T_{COND}\right)}{dt} = \dot{m}_{PIPE\_C}(h_2 - h_5) - \dot{m}_{H2O}(C_p^*)_{COND}[T_{Ref\_sat.v@P2} - T_{H2O\_in}]$$

PIPE (C) :

$$\frac{d\dot{m}_{PIPE\_C}}{dt} = \left(\frac{A_{PIPE\_C}}{L_{PIPE\_C}}\right)[P_{COND\_G} - P_{COND\_L}]$$

COND_G:

$$\frac{dm_{COND\_G}}{dt} = [\dot{m}_C - \dot{m}_{PIPE\_C}]$$

$$\frac{d\left((m_{COND\_G}\, C_V + m_{metal\_COND}\, C_{metal})\, T_{COND\_G}\right)}{dt} = \dot{m}_C h_{COND\_G} - \dot{m}_{PIPE\_C} h_2 = \dot{m}_C C_{p\_COND\_G}\, T_{COND\_G} - \dot{m}_{PIPE\_C} C_{p2}\, T_2$$

COMP:

$$\frac{d\dot{m}_C}{dt} = \left(\frac{A_C}{L_C}\right)[P_{EVA\_G} - P_{COND\_G} + C] \qquad \frac{dC}{dt} = \frac{(C_{ss} - C)}{\tau}$$

EVA_G:

$$\frac{dm_{EVA\_G}}{dt} = [\dot{m}_{PIPE\_E} - \dot{m}_C]$$

$$\frac{d\left((m_{EVA\_G}\, C_V + m_{metal\_EVA}\, C_{metal})\, T_{EVA\_G}\right)}{dt} = \dot{m}_{PIPE\_E} h_1 - \dot{m}_C h_{EVA_G} = \dot{m}_{PIPE\_E} C_{p1}\, T_1 - \dot{m}_C C_{p\_EVA\_G}\, T_{EVA\_G}$$

PIPE (E):

$$\frac{d\dot{m}_{PIPE\_E}}{dt} = \left(\frac{A_{PIPE\_E}}{L_{PIPE\_E}}\right)[P_{EVA\_L} - P_{EVA\_G}]$$

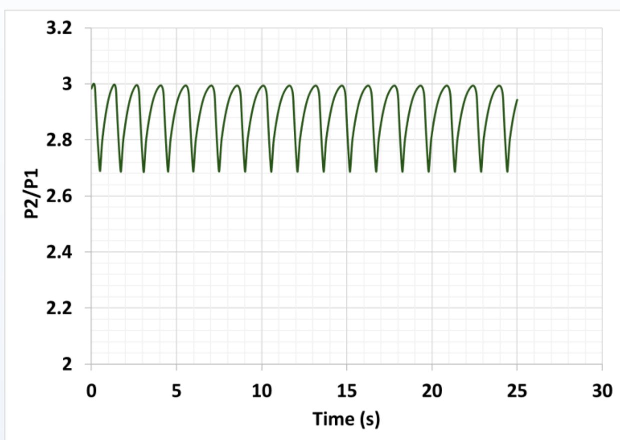
EVA_L:

$$\frac{d\left((m_{EVA\_L}\, C_V + m_{metal\_EVA}\, C_{metal})\, T_{EVA\_L}\right)}{dt} = \dot{m}_{H2O}(C_p^*)_{EVA}[T_{H2O_{in}} - T_{Ref_{sat}.v@P1}] - \dot{m}_{PIPE}(h_1 - h_7)$$

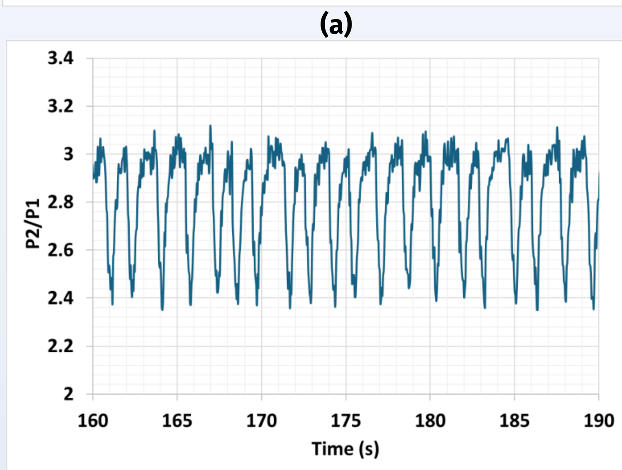
$$C_p^{*}{}_{COND} = \frac{\dot{m}_{COMP}(h_2 - h_5)}{\dot{m}_{H2O}[T_{Ref\_sat.v,2} - T_{H2O,COND,in}]}$$

$$C_p^{*}{}_{EVA} = \frac{\dot{m}_{COMP}(h_1 - h_7)}{\dot{m}_{H2O}[T_{H2O,EVA,in} - T_{Ref\_sat.v,1}]}$$

Results and Validation

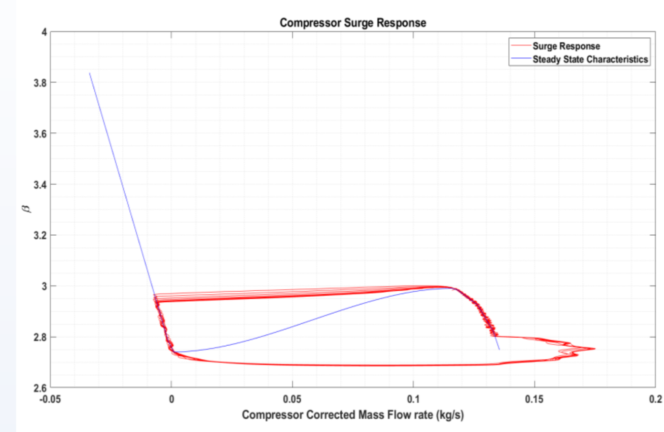


(a)

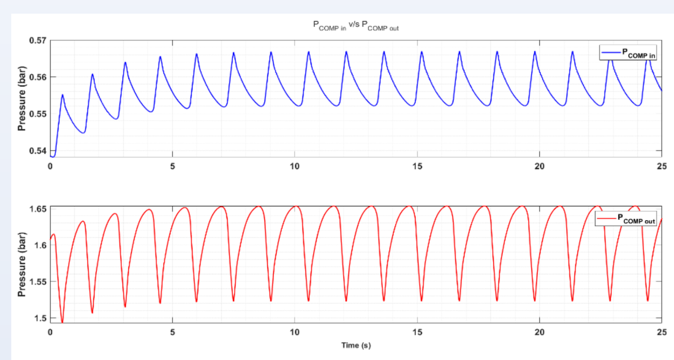


(b)

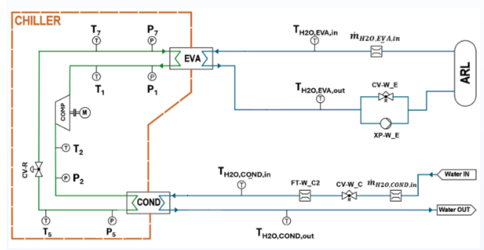
Pressure Ratio fluctuations a) Model b) Experiment (92% speed)



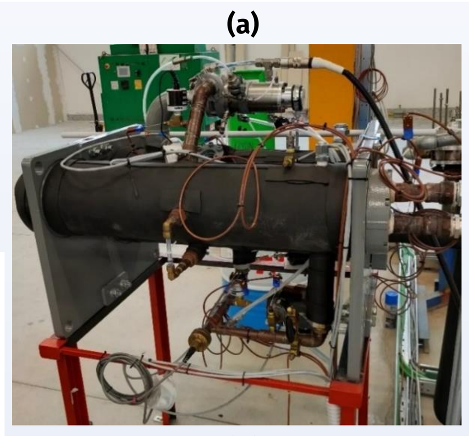
Post-stall transient response for 92% speed (Model)



Compressor Inlet & Outlet Pressures for 92% speed (model)



(a)



(b)

Experimental Test Rig (a) P & I diagram (b) Rig

Test Name	Speed	Frequency (Hz)		$\frac{(P_{2,max} - P_{2,min})}{P_{2,min}}$ (%)	
		Model	Experiment	Model	Experiment
92_19_14	92%	0.64	0.65	8.59	12.24
87_19_14	87%	0.68	0.67	8.47	12.31

With the correct tuning of liquid level parameters, the frequencies of oscillations predicted by the model were matched, and the model predicted the pressure fluctuations qualitatively and as well.

Conclusions

A closed-loop, 2 phase heat pump system with dynamic compressor is developed and validated with experiments for post-stall transient behaviour. The assumptions about the liquid levels will be replaced with actual measured values in the next step of this work.

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