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## An Analysis of Heat Transfer Effects on Surge Characteristics in Turbo Heat Pumps

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### ABSTRACT

This paper presents a new analysis of the effects of ambient temperature variations on the surge characteristics of turbo heat pumps. To this end, an analytical model capable of predicting unsteady behavior of surge in turbo heat pumps has been developed. In compression systems, surge is mostly caused by the compressor work exceeding the mechanical inertia of the system. In refrigeration systems, heat transfer in the condenser and evaporator can influence surge characteristics. The predictions demonstrate that the new model can accurately predict the limit cycle of behavior of turbo heat pumps after the surge onset. Also, predictions show that surge can be stabilized or removed by sufficient increase in the cooling water inlet temperature  $T_{cw,0}$ .

### 1. INTRODUCTION

Turbo heat pumps, also known as centrifugal chillers, are cooling and heating systems for buildings. The system uses refrigerants as the working fluid and consists of centrifugal compressor, expansion valve, and two shell-and-tube heat exchangers – condenser and evaporator. The refrigerant absorbs heat from a low temperature reservoir in the evaporator and releases heat to a high temperature reservoir in the condenser. Like other compression systems, turbo heat pumps can suffer from surge during off-design operations. Surge is accompanied by noise and violent vibration. In general compression systems, surge can cause mechanical failure of components, including impeller and bearings.

Surge is a one dimensional dynamic instability of compression systems. In Greitzer's gas turbine model (Greitzer, 1976a) surge occurs when the compressor exit pressure is higher than the design pressure. In turbo heat pumps, the abnormally high compressor exit pressure can occur for the following reason. The temperature of the building exterior (i.e. atmosphere) determines the temperature and corresponding saturation pressure in the condenser which is connected to the compressor exit. Thus, if the atmospheric temperature rises above that for which the heat pump designed, the pressure in the condenser can increase, and the compressor can encounter surge.

Much research has been conducted on surge in gas turbines which is an open-loop, single phase air compression system. Greitzer (1976) analytically and experimentally investigated surge in such compression systems. Greitzer's analytical model consists of an axial compressor, plenum, and throttle. The model is capable of accurately predicting surge in gas turbines. Hansen *et al.* (1981) showed that the Greitzer model is also applicable to describing surge in systems with centrifugal compressors. Botha *et al.* (2003) developed a surge model for both open-loop and closed-loop Brayton cycle systems by coupling the compressor and turbine. They took into account heat exchange as well as momentum effects.

Instabilities in pumping systems with liquids as working fluid have also been examined. Rothe and Runstadler (1978) developed theoretical models to explain surge in pumps. Although the fluid passing through the pump and throttle is liquid, the compressibility from the gas (air) in the compressor outlet tank yields pressure and pump mass flow rate fluctuations. They conducted experiments and validated their model. Instabilities in pumping systems with two-phase flows have also been examined. For example, Tsujimoto *et al.* (1993) has examined how cavitation affects instabilities in turbopumps.

Instabilities during phase change have been investigated. Kakaç and Bon (2008) developed analytical models of two-phase flow instabilities in tube boiling systems. The system consisted of a surge tank followed by a heater section and exit to atmosphere. They explained several types of instabilities with intersections and slopes of the characteristic curves of the tube and pump.

Transient responses of heat pumps have been investigated, but mostly from the heat transfer point of view. Chi and Didion (1981) studied transient performance of air-to-air heat pumps. They assumed a quasi-steady positive displacement compressor performance and simulated the start-up transients of the heat pump. Bendapudi *et al.* (2008) also investigated transients in a turbo heat pump. The reciprocating compressor was assumed to operate at steady-state condition, and the transient responses of heat exchangers were investigated.

Recently, Kim and Song (2010) developed an analytical surge model for turbo heat pumps based on the first principles, and showed the model can accurately predict surge. There are 9 nondimensional parameters in the turbo heat pump surge model. The important parameters which influence surge characteristics can be reduced 4 parameters,  $B$ ,  $\omega_2 / \omega_1$ ,  $H_1$ , and  $H_2$ . The effects of  $B$  and  $\omega_2 / \omega_1$  have been analyzed by Kim and Song (2010). The  $B$  parameter influences both the surge cycle shape and frequency. Also, surge frequency is increased exponentially, as  $\omega_2 / \omega_1$  is increased.

However, unlike gas turbines, turbo heat pumps also transfer heat from a low temperature reservoir to a high temperature reservoir. Therefore, not only mechanical energy from the compressor but also heat transfer in the condenser and evaporator can affect surge in turbo heat pumps. Yet such heat transfer effects have not been analyzed. This paper analyzes the effects of heat transfer on turbo heat pump surge characteristics.

## 2. MODEL DESCRIPTION

Figure 1 shows the schematic of a typical turbo heat pump system. At the compressor exit (Station a), the compressed vapor refrigerant flows into the condenser and exchange heat with the relatively low temperature cooling water passing through the tubes inside the condenser. After heat exchange, the cooling water temperature is increased and vapor refrigerant is condensed into a liquid form. The pressure and temperature of refrigerant is decreased as it passes through the expansion valve (Station b-c). The refrigerant is subsequently vaporized upon absorbing heat from the relatively high temperature chilled water in the evaporator (Station c-d) then returns to the compressor.

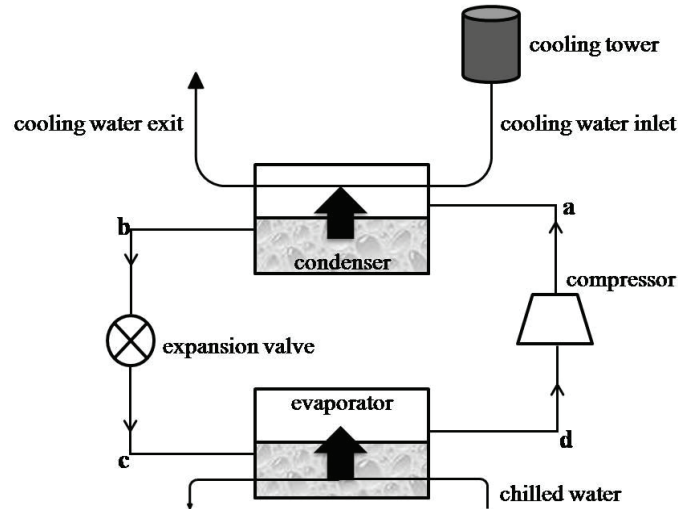


Figure 1 Schematic of turbo heat pump.

For this study, the recently developed turbo heat surge pump model of Kim and Song (2010) has been modified (APPENDIX).

The energy balances between the refrigerant and cooling water in the condenser, and between the refrigerant and chilled water in the evaporator can be expressed as

$$\dot{Q}_H = \dot{m}_{s1} h_{fg1} = g_1 \dot{m}_{cw} C_p (T_1 - T_{cw,0}) \quad (1)$$

$$\dot{Q}_L = \dot{m}_{s2} h_{fg2,x} = g_2 \dot{m}_{chw} (T_{chw,0} - T_2) \quad (2)$$

where  $g_1 = g_1(V_{1v})$  and  $g_2 = g_2(V_{2l})$  are determined by heat exchangers' geometries.

The two parameters,  $\dot{Q}_H$  and  $\dot{Q}_L$  are coupled to each other because the overall energy balance has to be satisfied. Thus, the conservation of energy gives us

$$\dot{Q}_H = \dot{Q}_L + \dot{W} \quad (3)$$

Equation A8 in the old model has been modified as

$$\tilde{m}_{s1} \tilde{h}_{fg1} = \tilde{m}_{s2} \tilde{h}_{fg2,x} + \tilde{m}_c \Delta \tilde{h} \quad (4)$$

where  $\Delta \tilde{h} = \tilde{h}_{1v} - \tilde{h}_{2v}$  is the enthalpy increase in the compressor. As heat transfer in the evaporator is calculated from system energy balance (Equation 4), nondimensional parameter in the evaporator  $H_2$  is not an input value. It is calculated by

$$H_2 = \dot{m}_{s2} \left[ g_2 \frac{(1 - \tilde{T}_2)}{\tilde{h}_{fg2,x}} \right]^{-1} \quad (5)$$

### 3. MODEL PREDICTIONS

The system of equations (Equations (A1-A9)) has been solved via a 4<sup>th</sup> order Runge Kutta method. Nondimensional input parameters, initial values, and compressor characteristic curve have been derived from experiments carried out at LS Mtron and are listed in Table 1. The refrigerant's properties are updated for fluctuating pressures using REFPROP, a commercial software program.

Table 1 Nondimensional Parameters.

$B$	0.639
$G$	6.34
$K$	16000
$\bar{\tau}$	0.0077
$\omega_2 / \omega_1$	0.2467
$Ma_1$	0.82
$Ma_2$	0.78
$H_1$	88.32
$H_2$	51.37

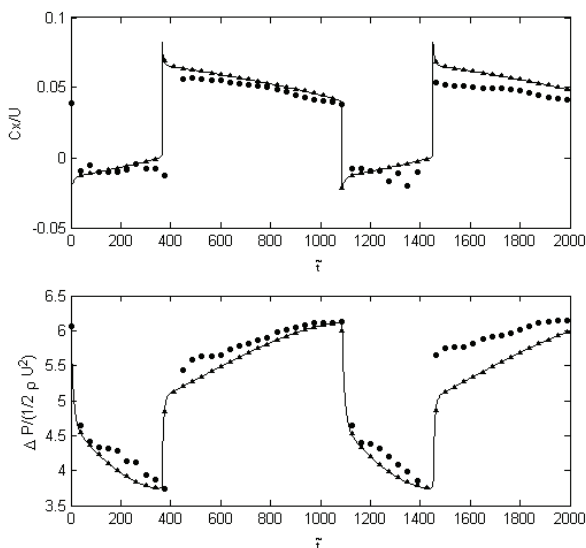


Figure 2 Nondimensional axial velocity and pressure rise.

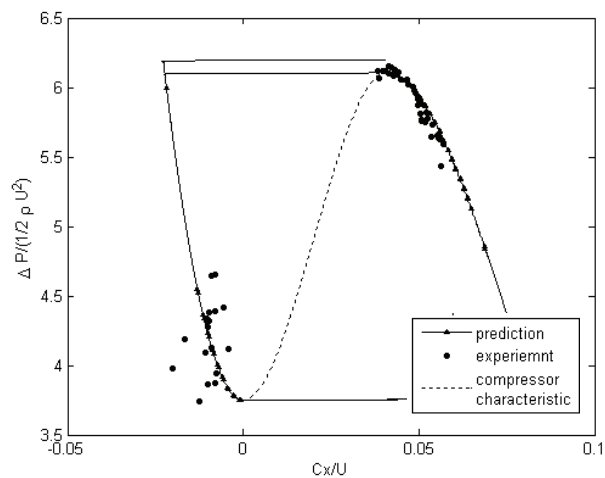


Figure 3 Pressure rise versus mass flow rate during surge.

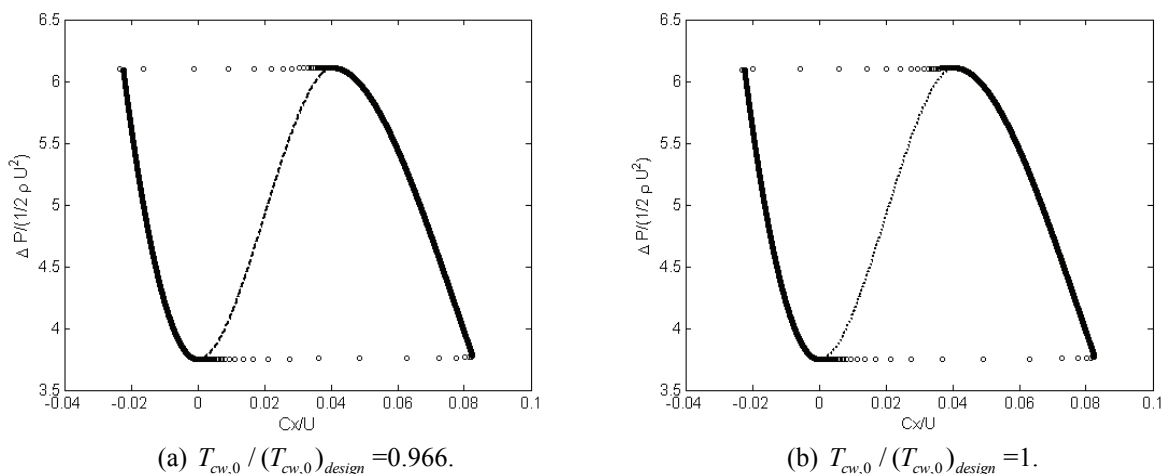
### 3.1 Design Point Operation

Figure 2 shows the graphs of predicted and measured nondimensional axial velocities at the compressor exit,  $C_x/U$ , and pressure rise through the compressor,  $\Delta \tilde{P} = \tilde{P}_1 - \tilde{P}_2$ , plotted versus nondimensional time. Figure 3 shows the predicted and measured surge cycle along with the compressor characteristic curve. Both velocity and pressure rise fluctuations are accurately predicted by the modified model.

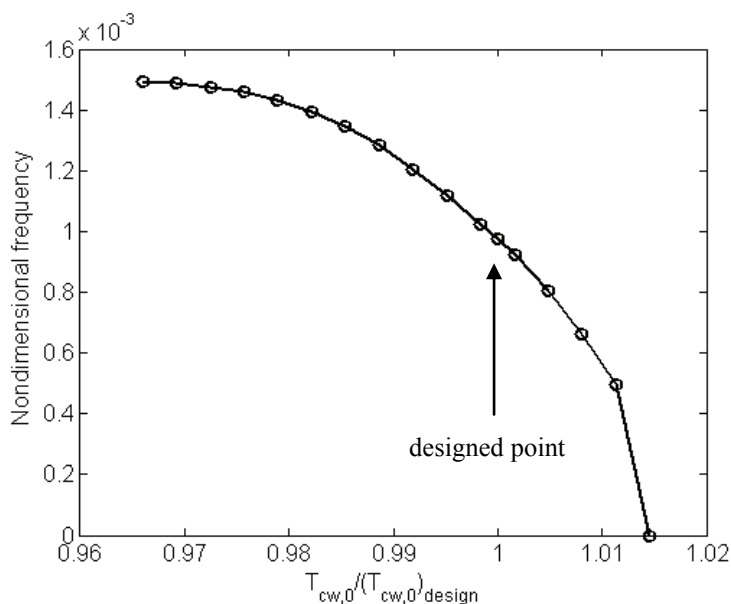
### 3.2 Heat Transfer Effects on Surge Characteristics

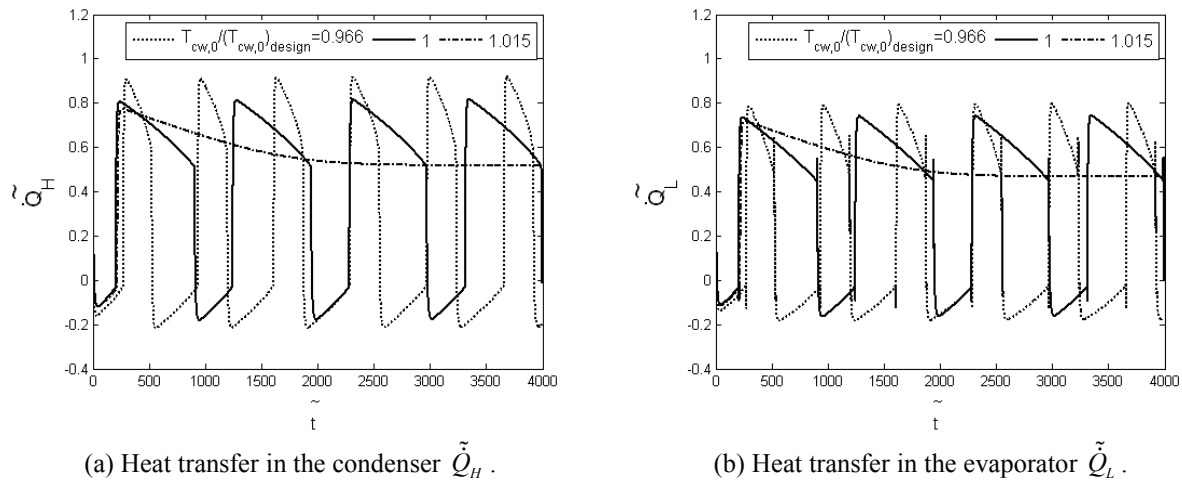
This section presents the influences of the cooling water inlet temperature  $T_{cw,0}$  on surge characteristics.

There are two types of instability; static and dynamic. Static instability is associated with the initial tendency of a system when stable operation is disturbed. Dynamic instability is characterized as the overall oscillatory motion after the stable operation is disturbed. We only consider the dynamic instability of system. The behavior of instability after onset of surge is covered here.

Figure 4  $T_{cw,0}$  effect on surge shape.

The parametric study on  $T_{cw,0}$  shows that surge shape is insensitive to the change of  $T_{cw,0}$  (Figure 4 (a) and (b)). However as  $T_{cw,0}$  influences surge frequency, Figure 5 shows the effect of cooling water inlet temperature on the nondimensional frequency of surge. Frequency is nondimensionalized by the Helmholtz resonator frequency of the condenser  $\omega_1$ . As  $T_{cw,0}$  is decreased from designed point, nondimensional frequency is slightly increased. Further decreases in  $T_{cw,0}$  do not influence the limit cycle behavior of surge. On the other hand an increase in  $T_{cw,0}$  leads to a decrease in nondimensional frequency of surge. At  $T_{cw,0} / (T_{cw,0})_{design} = 1.015$ , surge is stabilized. A further increase in  $T_{cw,0}$  results the same stable operation.

Figure 5  $T_{cw,0}$  effect on surge frequency.

Figure 6 Heat transfer characteristics with various  $T_{cw,0}$  .

Overall heat transfer characteristics upon changes in cooling water inlet temperature are shown in Figure 6. Graphs show the time variation of the nondimensionalized heat transfer in the condenser  $\tilde{Q}_H$  and evaporator  $\tilde{Q}_L$  . Surge frequency is decreased as  $T_{cw,0}$  increases. And surge is stabilized at  $T_{cw,0} / (T_{cw,0})_{design} = 1.015$ , as explained in Figure 5. The overall heat transfer  $\tilde{Q}$  is decreased as  $T_{cw,0}$  increases. It can explain why  $T_{cw,0}$  affects the surge frequency. Large value of the heat transfer  $\tilde{Q}_H$  means more condensation occurs in the condenser. That results in a small volume of vapor refrigerant in the condenser  $V_{lv}$  . Because the actual compressibility almost comes from the vapor region, not liquid region, small compressibility (energy storage ability) from small  $V_{lv}$  may result in the high surge frequency.

#### 4. CONCLUSIONS

The conclusions from this study of surge in turbo heat pumps are as follows.

- Surge has been predicted with mass and momentum conservation, refrigerant-cooling water energy balance, and system energy balance.
- Effect of the cooling water inlet temperature (ambient temperature),  $T_{cw,0}$  , on surge characteristics has been investigated.
- Surge frequency decreases as  $T_{cw,0}$  increases. And surge is stabilized above the steady state compressor characteristic curve when  $T_{cw,0}$  is sufficiently increased from the design point.

#### NOMENCLATURE

$A$	duct area	(m <sup>2</sup> )	<b>Subscripts</b>
$a$	speed of sound	(m/s)	0 inlet value
$C$	compressor pressure rise	(Pa)	1, $H$ condenser
$C_p$	specific heat of water	(J/Kg K)	2, $L$ evaporator
$F$	expansion valve pressure drop	(Pa)	$C$ compressor
$g$	coefficient of heat transfer	(-)	$chw$ chilled water
$H$	nondimensional parameter	(-)	$cw$ cooling water
$h_{fg}$	latent heat of refrigerant	(J/kg)	$l$ liquid refrigerant

$L$	duct length	(m)	$s$	phase change
$\dot{m}$	mass flow rate	(kg/s)	$T$	expansion valve
$N$	several compressor rotation	(-)	$v$	vapor refrigerant
$P$	pressure	(Pa)		
$\dot{Q}$	heat rate	(J/s)		
$R$	impeller inlet radius	(m)		
$T$	temperature	(K)		
$t$	time	(sec)		
$U$	impeller tip velocity	(m/s)		
$V$	refrigerant volume	(m <sup>3</sup> )		
$\dot{W}$	rate of work in compressor	(J/s)		
$\rho$	density	(kg/m <sup>3</sup> )		
$\tau$	time-lag constant	(sec)		
$\omega$	Helmholtz resonator frequency	(1/sec)		

**Operator**

( $\tilde{\quad}$ ) nondimensional variable

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## APPENDIX

The turbo heat pump surge model of Kim and Song (2010) consists of the following equations.

- continuity equations of each phase in the condenser

$$\frac{d\tilde{P}_1}{d\tilde{t}} = \frac{1}{B} \frac{\tilde{a}_1^2}{\tilde{V}_1} [\tilde{m}_C - (\frac{\tilde{\rho}_{1v}}{\tilde{\rho}_{1l}}) \tilde{m}_T + (\frac{\tilde{\rho}_{1v}}{\tilde{\rho}_{1l}} - 1) \tilde{m}_{s1}] \quad (A1)$$

$$\frac{d\tilde{V}_{1v}}{d\tilde{t}} = \frac{1}{B} \frac{Ma_1^2}{2} \frac{1}{\tilde{\rho}_{1l}} \frac{\tilde{a}_1^2}{\tilde{V}_1} [\frac{\tilde{V}_{1l}}{\tilde{a}_{1l}^2} \tilde{m}_C + \frac{\tilde{V}_{1v}}{\tilde{a}_{1v}^2} \tilde{m}_T - (\frac{\tilde{V}_{1l}}{\tilde{a}_{1l}^2} + \frac{\tilde{V}_{1v}}{\tilde{a}_{1v}^2}) \tilde{m}_{s1}] \quad (A2)$$

- continuity equations of each phase in the evaporator

$$\frac{d\tilde{P}_2}{d\tilde{t}} = \frac{G}{B} (\frac{\omega_2}{\omega_1})^2 \frac{\tilde{a}_2^2}{\tilde{V}_2} [-\tilde{m}_C + (\frac{\tilde{\rho}_{2v}}{\tilde{\rho}_{2l,x}}) \tilde{m}_T - (\frac{\tilde{\rho}_{2v}}{\tilde{\rho}_{2l,x}} - 1) \tilde{m}_{s2}] \quad (A3)$$

$$\frac{d\tilde{V}_{2v}}{d\tilde{t}} = \frac{G}{B} (\frac{\omega_2}{\omega_1})^2 \frac{Ma_2^2}{2} \frac{1}{\tilde{\rho}_{2l,x}} \frac{\tilde{a}_2^2}{\tilde{V}_2} [-\frac{\tilde{V}_{2l}}{\tilde{a}_{2l,x}^2} \tilde{m}_C - \frac{\tilde{V}_{2v}}{\tilde{a}_{2v}^2} \tilde{m}_T + (\frac{\tilde{V}_{2l}}{\tilde{a}_{2l,x}^2} + \frac{\tilde{V}_{2v}}{\tilde{a}_{2v}^2}) \tilde{m}_{s2}] \quad (A4)$$

- momentum conservation equations through the compressor and expansion valve

$$\frac{d\tilde{m}_C}{d\tilde{t}} = B[-\tilde{P}_1 + \tilde{P}_2 + \tilde{C}] \quad (A5)$$

$$\frac{d\tilde{m}_T}{d\tilde{t}} = \frac{B}{G} [\tilde{P}_1 - \tilde{P}_2 - K \frac{\tilde{m}_T^2}{\tilde{\rho}_{2l,x}}] \quad (A6)$$

- energy balance between the refrigerant and cooling water in condenser and the system energy balance

$$\tilde{m}_{s1} = g_1 H_1 \frac{(\tilde{T}_1 - 1)}{\tilde{h}_{fg1}} \quad (A7)$$

$$\tilde{m}_{s2} = g_2 H_2 \frac{(1 - \tilde{T}_2)}{\tilde{h}_{fg2,x}} \quad (A8)$$

- compressor hysteresis effect

$$\tilde{\tau} \frac{d\tilde{C}}{d\tilde{t}} = \tilde{C}_{ss} - \tilde{C} \quad (A9)$$

Nondimensional parameters are

$$B = \frac{U}{2\omega_1 L_C} \quad G = \frac{L_T A_C}{L_C A_T} \quad K = \frac{A_C^2}{A_s^2} \quad \tilde{\tau} = \frac{\pi N R}{L_C B} \quad \frac{\omega_2}{\omega_1} = a_{2v,0} \sqrt{\frac{A_T}{V_{2,0} L_T}} / a_{1v,0} \sqrt{\frac{A_C}{V_{1,0} L_C}}$$

$$Ma_1 = \frac{U}{a_{1v,0}} \quad Ma_2 = \frac{U}{a_{2v,0}} \quad H_1 = \frac{\tilde{m}_{cw} C_p T_{cw}}{U^2} \quad H_2 = \frac{\tilde{m}_{chw} C_p T_{chw}}{U^2}$$