THE START-UP BEHAVIOUR OF A GAS TURBINE - EJECTOR - LONG STACK SYSTEM

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INTRODUCTION

Emergency power generating units of power plants are usually driven by Diesel engines or gas turbines, which discharge their exhaust gases via short (6-7m high) stacks. The Commission's Loy Yang A Power Station emergency gas turbine units were to be installed between two boiler houses. For safety reasons, this necessitated ducting of the high temperature exhaust gases above the boiler house roof via a 120 m high 200 m long stack.

The general arrangement of a unit is shown in Figure 1. Two gas turbines drive the generator through a common gear box, all of which is housed in an acoustic enclosure. The exhaust from each turbine discharges into an ejector which draws cooling air from the enclosure and attemperates the exhaust gases. Cooling air enters the generator enclosure via filters and passes into the gas turbine enclosure through the generator and the oil cooler radiator fans. The outflow from the two ejectors pass to a common silencer and then to the long exhaust stack. On start-up the turbines achieve full speed in approximately 30 seconds with very rapidly increasing mass-flow at the end of this interval (Figure 4). The exhaust temperature peaks at about 20s and after a trough it rises again as more fuel is injected according to the loading of the generator.

Concern was expressed that the back pressure caused by the long stack would be too great to allow the ejectors to operate satisfactorily and there was a possibility of backflow of hot exhaust gases into the turbine enclosure. Calculations showed that for an acceptable stack diameter the system would operate satisfactorily under steady state operating conditions. To determine if the system would operate satisfactorily during start-up, a model was developed to describe the flow through the system of filters, fans, gas turbines, ejectors, silencer, and stack.

NOMENCLATURE

A	area	s	length
C	pressure loss coefficient	t	time
f	friction coefficient	T	temperature
g	gravitational acceleration	v	average velocity
h	height	v(r)	local velocity
m	rate of mass-flow	ρ	average density
p	pressure	$\rho(r)$	local density

Subscripts:

c	cooling air	t	turbine
e	enclosure	x	ejector outlet
f	filter	o	ambient
i	ejector inlet		

MATHEMATICAL MODELS

During the development, several models were formulated and programmed for solution on a PDP11/34 computer. A preliminary model was developed for a single pipe with known rate of inlet massflow. The system of two partial differential equations (momentum equation and continuity) was solved for velocity and pressure by finite difference techniques.

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The results did not preclude the possibility of satisfactory start-up operation and provided insight into what aspects could be neglected in a detailed model. In the second step described here the aim was to include more elements of the complex ducting system, whilst keeping the model simple, in accordance with the reliability and availability of input data. The most essential output from the model is the predicted cooling air flow rate which is treated as an iteration variable. At time t we assume that everything is known and at the new time t+dt we know the turbine exhaust mass-flow and temperature. It is assumed that the exhaust flow characteristics of the turbines (Figure 4) are not influenced by the system (e.g. by back-pressure). Starting with an estimated cooling air mass-flow (that can be negative), the pressure in the generator enclosure is -

$$p_e = p_o - \frac{\rho_o}{2} C_f \; v_f \; |v_f| \tag{1}$$

As the radiator fans in the enclosure are only switched on when the turbines are nearly up to speed, the pressure in the turbine enclosure can be regarded as being the same as in the generator enclosure.

The ejector inlet pressure for positive and negative cooling air flow is -

$$\begin{aligned} p_i &= p_e - (1 + C_i) \frac{\rho_o}{2} v_c^2 \qquad && for \;\; m_c > 0 \\ p_i &= p_e && for \;\; m_c \leq 0 \end{aligned} \tag{2}$$

and is regarded as average for the whole inlet cross section. The pressure increase through the ejector can be calculated from the integral form of the momentum equation for the control volume shown in Figure 2. Neglecting the unsteady term and the wall friction (the ejector is two orders of magnitude shorter than the stack), and assuming top-hat velocity profiles, the average pressure at the outlet for positive or negative cooling air flow is -

$$p_x = p_i + \frac{m_t v_t + m_c v_c - m_x v_x}{A_x} \tag{3}$$

Since the inlet/outlet velocity profiles are very much like those shown in the bottom of Figure 3, all of the momentum terms should be corrected by appropriate profile form-factors of the form -

$$\frac{\iint \rho(r) \, v^2(r) \, dA}{\rho \, v^2 A} \tag{4}$$

These form-factors are calculated from steady state velocity survey data and are considered constants during the start-up. The average temperature of the turbine exhaust - cooling air mix at the ejector outlet is -

$$T_x = \frac{m_t T_t + m_c T_c}{m_t + m_c} \tag{5}$$

There is no need to introduce temperature profile form-factors. The cooling air temperature is equal to the enclosure temperature in the case of positive cooling air flow, and is equal to the turbine exhaust temperature in the case of backflow. The latter assumption provides quick convergence of the iterations in the case of backflow.

Starting with the silencer, the stack is divided into small elements in which the density can be regarded constant. Accounting for the inertia forces, friction, and buoyancy, the pressure difference between neighbouring nodes is -

$$p_j - p_{j-1} = -\frac{dm_x}{dt}\frac{\Delta s}{A} - \frac{\varrho}{2} f \frac{\Delta s}{d} v|v| - \varrho g \Delta h \qquad (6)$$

The friction factor contains the pipe friction as well as the losses of elbows, etc. It is assumed that since the changes are relatively slow, losses, etc, can be calculated as if we were dealing with a succession of steady state flows. The local density and local velocity are calculated on the basis of the local temperature. Temperatures are calculated from a simplified energy equation -

$$\frac{\partial T}{\partial t} = -v \frac{\partial T}{\partial s} \qquad (7)$$

In neglecting heat losses to the walls of the ducting this equation overestimates the temperature. The error, however, is not thought to be very significant (the critical period is only about 35 seconds) and, since in our particular case a temperature drop would result in nearly the same drop in buoyancy as in friction losses, the accuracy of the solution is still commensurate with that of the input data.

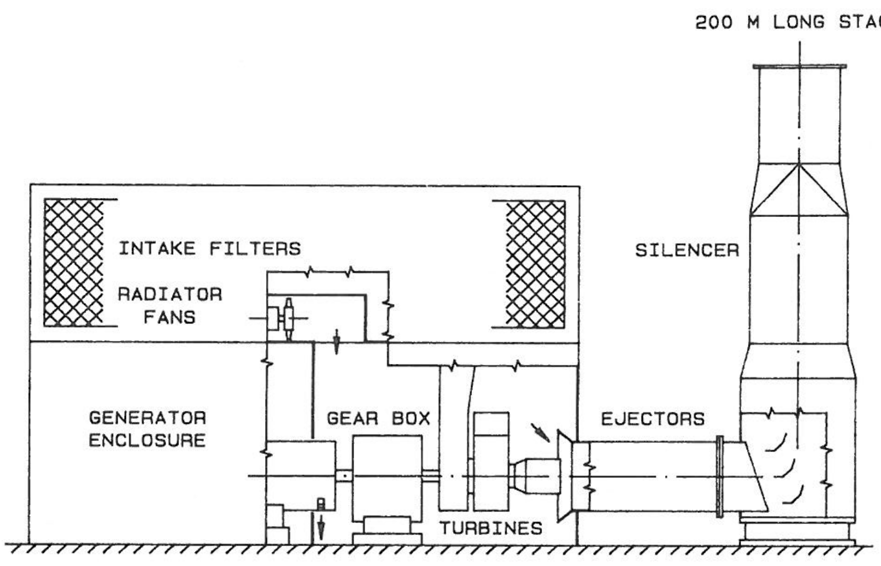
The advantage of this form of the energy equation is that it can be given a geometrical meaning (Figure 3). If the "old" temperature distribution is known, we can create the "new" distribution by shifting each of the old nodes a distance that is the products of the local velocity and the time-step (see the arrows leading to the new points P1, P2, ... in Figure 3). The point P0 is the mixture temperature at the ejector outlet calculated from equation 5. We can now fit a spline through the points P0, P1, ..., and obtain the new temperature values at the old nodes. In this way essentialy bigger steps can be employed both in space and time; 20-30 nodes for the 200 m long stack, and time steps of 0.1-0.25 s. (This made possible the use of the PDP11/34 computer with software simulated floating point processor.)

With the repetitive use of equation 6 the pressure at the stack outlet can be calculated. If it is not equal to the ambient pressure at that level, the cooling air flow should be modified. Usually, the balancing cooling air flow can be found in 3 steps; more steps (5-6) are needed when the mass-flow is small and in the case of backflow.

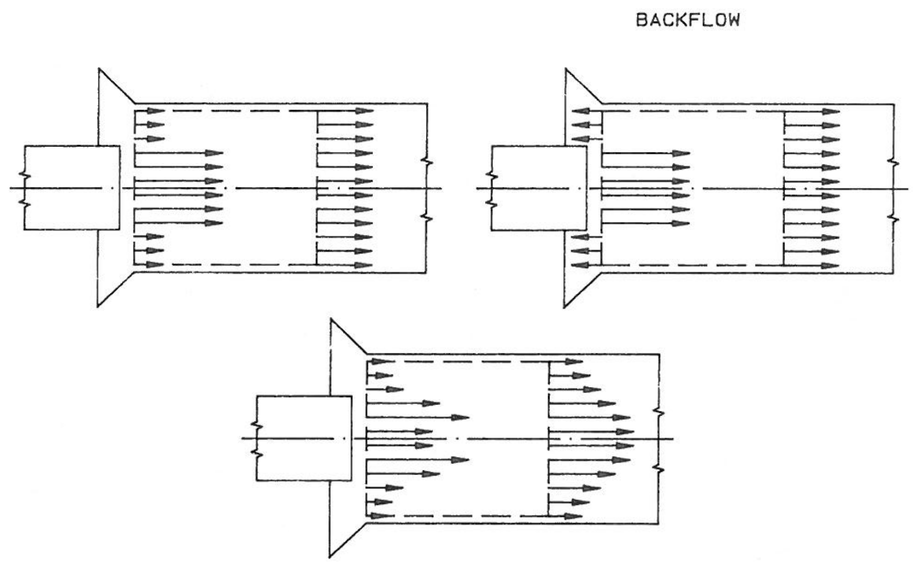
RESULTS AND CONCLUSIONS

The computer program written to solve the governing equations generates intermediate nodes along the stack, fits splines on the input mass-flow and temperature points. On the output side it prints/plots all the primary variables along the stack at predefined times (see a typical temperature plot in Figure 5). The main purpose of the model is to predict the cooling air flow. The most adverse situation in practice is when start-up is with a cold stack. In such a case an initial backflow is predicted, followed by positive cooling air flow with a sharp dip at the time when the turbine mass-flow increases very rapidly towards the end of the run-up to speed (Figure 4). Depending on system parameters (e.g. filter pressure drop) backflow may occur at this stage of the start-up. Starting with a "preheated stack" (e.g. start after a trip-off), the resulting draft is enough to offset the backflows.

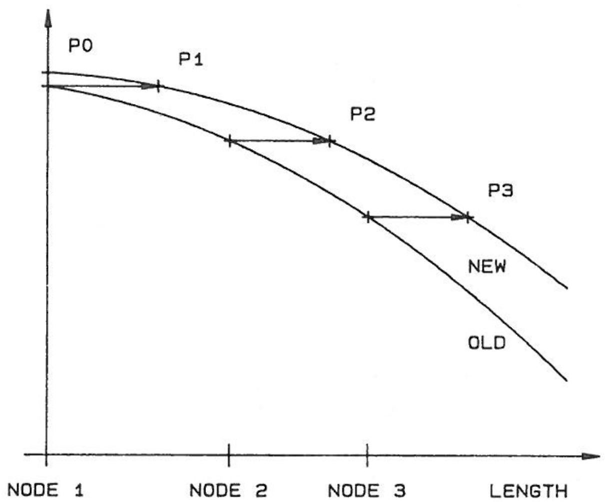
The system has been installed and commissioned but at the time of writing this paper there are no detailed measurements available with the long stack. However, observations confirmed that on cold winter days when starting from cold, positive enclosure pressure occurs in the first 5-10 seconds, and there appeared to be some backflow for a very short period of time at the end of the turbine acceleration period. Nevertheless, there has been no report of any damage due to overheating within the turbine enclosure. Under warmer atmospheric conditions or with a start following trip-off, backflows into the enclosure are most likely avoided.



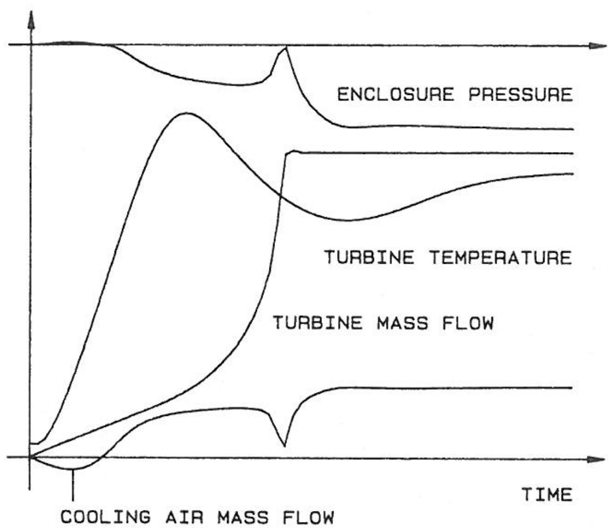
GENERAL LAYOUT
FIGURE 1



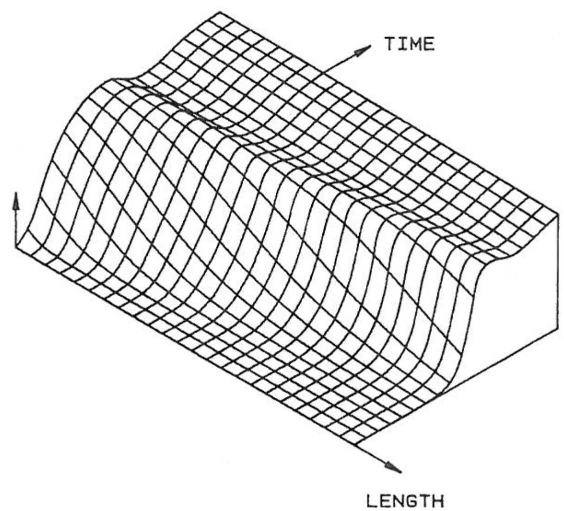
EJECTOR VELOCITY DISTRIBUTIONS
FIGURE 2



THE CALCULATION OF A NEW TEMPERATURE CURVE
FIGURE 3



CHARACTERISTIC CURVES
FIGURE 4



TEMPERATURE DISTRIBUTION ALONG THE STACK
FIGURE 5