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ABSTRACT

A simple power plant steam header model has been developed for normal operation and operational transient simulations The starting basis has been the the time-response analysis of the power plant steam header and its related plant components The analysis of the time-constants confronted against the user requirements led to the conclusion that a simple header model based on the quasi-static assumptions can be adequately used in most of the calculation needs where a time frequency requirement for information processing does not exceeds few Hertz The header model has been solved with the Newton-Raphson technique and its fast convergence warranted an economic program run in personal microcomputers

SIMULAÇÃO NUMÉRICA DO COLETOR DE VAPOR

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RESUMO

Foi desenvolvido um programa de computação de um modelo simplificado de coletor de vapor para operação normal e transientes operacionais O ponto de partida foram as análises de tempos de resposta do coletor e dos principais componentes a ele ligados A análise dos tempos de resposta confrontados contra os requisitos dos usuários levou à conclusão que um modelo simples de coletor baseado em hipótese quase-estática pode ser adequadamente usado na maioria dos cálculos onde a frequência de processamento de informações não excede alguns Hertz O modelo do coletor foi resolvido pela técnica de Newton-Raphson e sua rápida convergência permitiu uma economia na execução no ambiente de microcomputador

1 INTRODUCTION

In a steam supply system fed by multiple boilers the efficient use of steam may rely on the appropriate steam header design, such that the plant performance is kept close to its optimum operating conditions Moreover, when steam from many boilers is shared between two turbines, generally at the distinct demand levels, the pressure differences along the connecting tubes play a key role in the mass flow distribution in the steam tube network

The boilers pressure level is the primary driver for the steam flow pattern, but, it is in turn, primarily influenced by the steam outflow level, responding to it with a short time constant that is, much faster than to the temperature change driven by heat transfer phenomena

On the other extremity of the tube network the steam flows are determined by the levels of power demand imposed on each one of the turbines In general, a turbine controller is not very sensitive to the steam pressure if its minimum level in not trespassed thus the steam demand at those points can be determined Even if the condition is not satisfied a steam consumption versus pressure curve can readily be built

As previously noted, the steam tube network, steam header included, plays the driving role in determining the steam outflow levels from the boilers It is therefore the subject of our investigation in the present work, the objective being the development of a reliable and computationally inexpensive steam header model which might be implemented in a full nuclear plant simulator

The time-response capability of the steam header model must be compatible with the frequency-response requirements demanded by the the plant simulator users In average, the frequency response in a range of few Hertz is the expected requirement, derived from the commercial plant analysis and the plant control simulations The requirement also fulfills the hardware availability, since the simulator is written for personal microcomputers

The steam header simulation program to be developed in this work must be as simple as possible, but, uncompromisingly, the quality requirement must be preserved

Complex models are, in fact, very powerful but they are excessively and inherently time consuming, as they deal with and generate a great deal of wasteful detailed state variables Thus, simple models which could assess the most relevant variables within acceptable accuracy and, at the same time, were fast and economic would be of valuable help

2 STEAM HEADER MODEL DESCRIPTION

An example of steam header network which links the boilers to the turbines is shown in Figure 1 The local pressure and steam flows are defined by P and W, respectively, with the subscripts indicating the locations

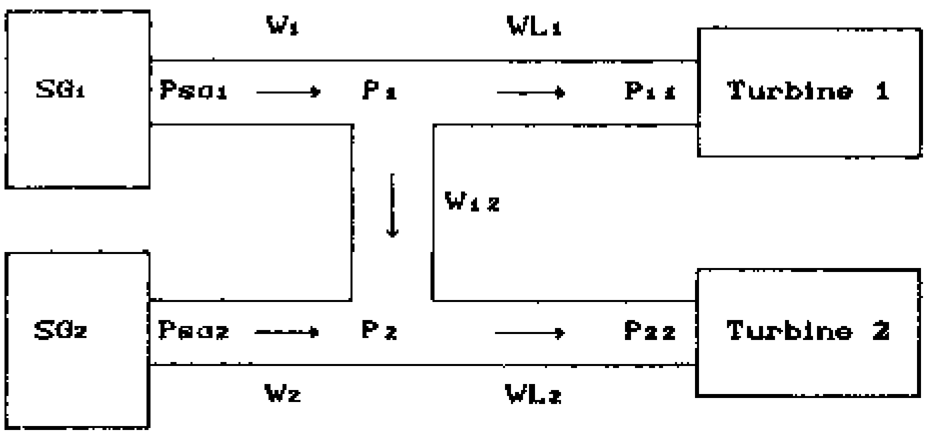


Figure 1 - *Steam Header Sketch*

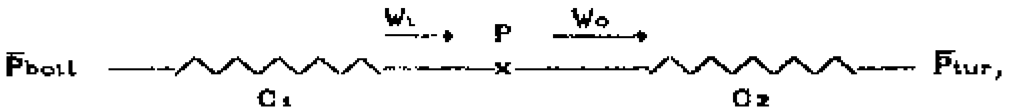
The power demand levels on each turbine may independently be controlled, may even be shut off, so that the W_{12} steam flow direction can either be positive or negative, depending upon the boiler pressure and demand unbalancing degree

In normal operating condition the value of W_{12} is close to zero and the average steam transit time from the boiler to the turbine is around half second

2.1 STEAM HEADER TIME-CONSTANT ESTIMATE

In this section a discussion on the steam header time-response characteristics is performed The relevant phenomena and time dependence is treated in detail

Considering a point of interest on the steam tube the local pressure P will be estimated following the simplified resistive network sketched below The end pressures, $\bar{P}_{boil}$ and $\bar{P}_{tur}$, are fixed and they are determined by the boilers and the turbines, respectively The network impedances are represented by C_1 and C_2



The steam flow rates, W_i and W_o, depend on the pressure differences according to the relationships

$$\bar{P}_{boil} - P = C_1 \, W_i^2 \qquad \text{(Eq 1)}$$

and

$$P - P_{tur} = C_2 \, W_o^2 \qquad \text{(Eq 2)}$$

The mass conservation equation applied on the steam header is given as

$$V \frac{d}{dt} \rho = W_i - W_o \qquad \text{(Eq 3)}$$

where

ρ is the steam density and
V is the header volume

and its expansion in terms of pressure and enthalpy results

$$V \left\{ \frac{\partial}{\partial P} \rho \bigg|_h \frac{d}{dt} P + \frac{\partial}{\partial h} \rho \bigg|_P \frac{d}{dt} h \right\} = W_i - W_o$$

(Eq 4)

The approximate figure for $\frac{\partial}{\partial P} \rho \big|_h$ is about 0 11kg/m^3/bar for liquid and 0 6kg/m^3/bar for vapor

Meanwhile the figure for $\frac{\partial}{\partial h} \rho \big|_P$ is about −0 15kg/m^3/(J/kg) for liquid and −0 00002kg/m^3/(J/kg) for vapor

With the figure of $\frac{d}{dP}Hv$, on the saturation line, which is about -320(J/kg)/bar we estimate an enthalpy variation in the order of 320J/kg for vapor if 1 bar of pressure variation occurred in the boiler steam dome The average header enthalpy variation can be estimated from the energy conservation equation

$$\frac{d}{dt}h = (h_i - h) W_i/(V\rho) - (h_o - h) W_o/(V\rho) ,$$

where h_i and h_o are the incoming and outgoing steam flow enthalpy, respectively

In the instant mixture approximation, i e , $h_o = h$, the equation becomes

$$\frac{d}{dt}h = (h_i - h) W_i/(V\rho) \qquad \text{(Eq 5)}$$

The incoming enthalpy, h_i, is the boiler enthalpy Thus, as an exercise, if $W_i \simeq 10kg/s$, $\rho \simeq 20kg/m^3$ and $V \simeq 0\,7m^3$, an instant variation of 1bar in the boiler would yield an enthalpy variation rate of 230(J/kg)/s in the steam header, with a time constant of 1 4s If $W_o \simeq 3kg/s$ the header time constant would be of the order of 0 5s

Assuming, as an example, a rapid pressure increase of 1 bar in 1 second, the first term on the left side of Equation 4 would be estimated to be 0 6kg/m^3 while the second term would be estimated as 0 0046kg/m^3 That is, the later is about 100 times smaller than the former, negligible for an overall estimate

Therefore, in the header energy equation, the accumulation term is relatively small, in the first approximation it can be neglected The result is

$$\boxed{\frac{d}{dt}P = A (W_i - W_o)} \qquad \text{(Eq 6)}$$

where

$$A = 1/ \left\{ V \frac{\partial}{\partial P}\rho \bigg|_h \right\}$$

Thus, for a typical power plant the figure for A is of the order of

$A = 14 bar/kg$	for liquid, and
$A = 2.5 bar/kg$	for vapor

Substitution of W_i and W_o given by Equation 1 and 2 into the left side of Equation 6 gives

$$\frac{d}{dt} P(t) = f(t,P(t)) \qquad \text{(Eq 7)}$$

where

$$f(t,P(t)) = \frac{A}{\sqrt{C_1}} \sqrt{P_1 - P} - \frac{A}{\sqrt{C_2}} \sqrt{P - P_2}$$

The behavior of the pressure P is therefore nonlinear The estimative of its Lipschitz/1/ constant, L, is

$$L = \max \left| \frac{\partial f(t,P(t))}{\partial P} \right|$$

or

$$L = \max \left| \frac{A}{2\sqrt{C_1}\sqrt{P_{boil} - P}} + \frac{A}{2\sqrt{C_2}\sqrt{P - P_{tur}}} \right| \qquad \text{(Eq 8)}$$

As an exercise with $C1 \simeq C2 \simeq 0.007 bar/(kg/s)^2$, $(P_1 - P) \simeq (P - P_2) \simeq 0.5 bar$, and $A \simeq 2.5 bar/kg$, one estimates a figure for L as

$$L \simeq 40/\text{second}$$

By approximating the left side of Equation 7 with a linear expansion around a reference point t_o, one gets

$$\frac{d}{dt} P(t) = f(t,P(t_o)) + \frac{\partial}{\partial P} f(t,P(t)) \Big|_{t_o} (P(t) - P(t_o))$$

(Eq 9)

If the reference point t_o were fixed at the time when an stable equilibrium between W_i and W_o still existed, a step increase in the pressure in the pressure $P(t_1)$ would give

$$\boxed{P(t) = P(t_o) + [P(t_1) - P(t_o)]\, e^{-Lt}} \qquad \text{(Eq 10)}$$

The steam header time constant τ, can therefore be estimated as

$$\boxed{\tau \approx 1/L \simeq 0.075 \text{second}}$$

In the other hand the Equation 6 can be interpreted as an equation of the response of an controller(integrator) driven by an error $(W_i - W_o)$, where A is the gain of the integrator/2/ And once the gain of an integrator is inversely proportional to its time-constant, one can have it adjusted to have the controller operational only in a limited frequency range

In the present example the particular dependency of the steam flow relative to the pressure difference between the header extremities resulted in a clear cutoff in the frequency domain By comparing Equation 6 to Equation 10 and noticing that a step pressure increase $P(t_1)$ corresponds to an error $(W_i - W_o)$ and $P(t_o)$ corresponds to the null error one can rewrite the Equation 10 defining $(P(t_1) - P(t_o))$ as an error ε as

$$\delta P = \varepsilon\, e^{-Lt} \qquad \text{(Eq 11)}$$

where

$$\delta P = P(t) - P(t_o)$$

which by the Laplace transform becomes

$$\overline{\delta P} = \frac{1}{s + L}\,\bar{\varepsilon}$$

Therefore steam header is effectively a low-pass filter with the threshold frequency around L, i e , of the order of 40Hz

Thus within few tenths of a second the pressure in the header reaches equilibrium state And, consequently, phenomenon whose frequency lies above L will not be reproduced by the header response For further header response analysis it is necessary to assess the time-response characteristics of all components connected to the header

If a connecting component time-response were found to be very large a correspondingly very low frequency input would be fed to the header and obviously the header response would follow this low frequency But if the header cutoff frequency were found to be too high, comparatively to the input frequencies of its connecting components it would mean that header cutoff frequency, thus the integrator gain could have been lowered without any harm to the header response Further, this relaxation would also bring the advantage in the integration procedure since the time-step size could be enlarged on account of the slower response characteristics of such an idealized header model

The artificially enlarged header time-constant, τ_{req} can be chosen as the inverse of the new cutoff frequency and the smaller integrator gain, A, can be derived for the Equation 6 by satisfying the relationship

$$1/\tau_{req} = \frac{A}{2\sqrt{C_1}\,\sqrt{\overline{P}_{boil} - P}} + \frac{A}{2\sqrt{C_2}\,\sqrt{P - \overline{P}_{tur}}} \qquad \text{(Eq 12)}$$

for all possible values which maximizes the denominators A simplified expression for C_1 and C_2 above, would be

$$A = 0\,084\,\sqrt{\Delta P(\text{bar})} \;/\; \tau_{req}(s) \quad (\text{bar/kg}) \qquad \text{(Eq 13)}$$

As example, in order to get $\tau_{req} \simeq 0.1s$ the value of A for ΔP=0.5bar is

$$\boxed{A = 0.60 bar/kg} \quad (\tau_{req} \simeq 0.1 second)$$

In comparison with the value obtained from Equation 6 the integrator gain is about four times smaller. Therefore in this example, the steam header time-response is of the order of 0.1 second reaching an equilibrium in about 0.3 second for a step error input.

In case of the turbines whose steam flow rates are not very sensitive the header pressure level the steam demand variation is dictated mostly by the power demand of the plant controller. Is this case the second term on the right side of the Lipschitz expression Equation 8, can be neglected and the resulting L value is

$$\boxed{L = 20/second}$$

therefore the header time-constant for the steam header becomes

$$\boxed{\tau = 0.05 second}$$

2.2 - BOILER PRESSURE TIME-CONSTANT

In a simplified assumption the boiler pressure reacts to two main driving inputs: the steam production rate variation from the heat transfer phenomenon and the steam flow rate variation demanded by the turbines. The time-constants involved are an order of magnitude apart from each other and the dominant phenomenon in a short time lag, which is the present objective, is the second one.

A simple estimation of steam flow dependency of the boiler pressure can be devised from the gas law approximation as

$$P_{boil} = ZRT\rho_{boil}$$

where Z is the compressibility factor assumed constant. And for a boiler pressure of 35bar the ZRT is about 22bar m^3/kg

The pressure variation rate as function of the steam flow rates can therefore be given as

$$\frac{d}{dt}P_{boil} = ZRT\frac{d}{dt}\rho_{boil}$$

which with the utilization of mass conservation equation, Equation 9 can be rewritten as

$$\boxed{\frac{d}{dt}P_{boil} = B\,(W_{boili} - W_{boilo})} \qquad \text{(Eq 14)}$$

where

$B = ZRT/V_{boil}$,

V_{boil} is the boiler steam dome volume

W_{boili} and W_{boilo} as production and extraction flow rates, respectively

With a typical V_{boil} value of 13m^3, B could be of the order of

$$\boxed{B = 17\ \text{(bar/kg)}}$$

The largest unbalance of W_{boili} and W_{boilo} to be considered in the present time-response study can be taken by assuming the largest steam rate variation demanded by the turbines. Assuming a commercial turbine with about 6kg/s in steam consumption, a sudden turbine trip could yield an instant pressure variation rate of

$$\frac{d}{dt}P_{boil} = 10\ \text{bar/s}$$

On the left hand side of Equation 14 the value of W_{boili} is practically independent of the boiler pressure, but the value of W_{boilo} is given by

$$P_{boil} - P = C_2 W_{boilo}^2 \qquad \text{(Eq 15)}$$

Substituting the W_{boilo} of Equation 15 into Equation 14, and carrying through a procedure similar to that of the steam header, a Lipschitz constant is obtained as

$$L = \max \left| \frac{B}{2\sqrt{C_1}\ \sqrt{P_{boil} - P}} \right|$$

which, for a pressure difference of 0 5bar, yields

$$L = 14/\text{second}$$

Therefore the steam header time-constant in a step variation of the steam outflow is of the order of 0 07 second

2 3 - COMPARISON OF THE TIME-CONSTANTS

The sound velocity in a steam, about 550m/s, permits the pressure transmission in a steam header in less than 0 05 second while the physical velocity is about 40m/s which results in about 0 5 second to transport the enthalpy

The boiler time-constant was estimated as 0 07 second, and the header time-constant as 0 05 second The elapsed time to reach an asymptotic state is about three time-constants, that is 0 2 second and 0 15 second, respectively

In a constant speed steam turbine the steam demand is controlled by a centrifugal device which drives the valve aperture in a fraction of a second An experimental time of 0 1 second is believed to be representative for most commercial turbines Therefore the boiler steam extraction rate responds to the turbine demand in about 0 1 second

Comparison of the above figures leads to the conclusion that the header time-constant is sufficiently small to absorb the steam flow variations coming from both the boiler and the turbine Moreover, if the frequency of the information extraction is smaller than 10Hz the header can be treated quasi-statically

Therefore the steam header can be assumed as if it had reached an equilibrium state after a variation of the boiler and the turbine states occurred The header can be assumed to be at equilibrium at each time-step

The energy equation can be straitfully considered as the enthalpy is transported in less than four time-steps of size 0.1second

3 - QUASI-STATIC MODEL FOR THE STEAM HEADER

The quasi-static model of the steam header can be further simplified since the right portion of the header network is swept by steam flow rates WL_1 and WL_2 which are quite independent from the header pressure, as explained they are mainly dictated by the turbine demand levels

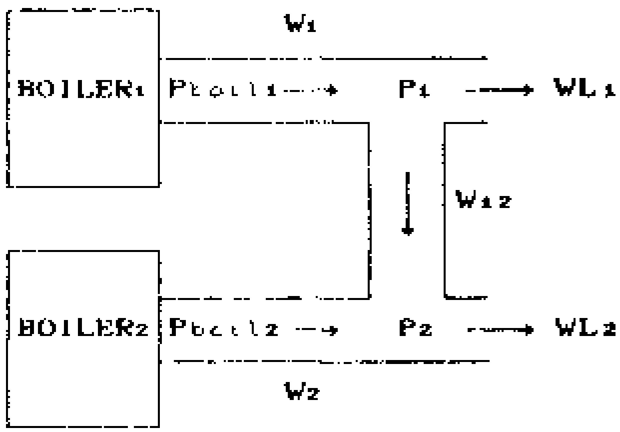


Figure 2 - Header quasi-static model

The equations which govern the state variables in this model are

$$\begin{cases} P_{boil1} - P_1 = C_1 W_1^2 \\ P_{boil2} - P_2 = C_2 W_2^2 \\ P_1 - P_2 = C_{12} W_{12}^2 \\ W_1 = WL_1 + W_{12} \\ W_2 = WL_2 - W_{12} \end{cases} \quad \text{(Eq 16)}$$

where

$$C_i = \frac{k_i}{2 \rho_i A_i^2}$$

$$k_i = f\ L_i / D_i + \sum K_i \quad \text{, with } i = 1,\ 2 \text{ and } 12$$

The unknowns, P_1 P_2 W_1 W_2 and W_{12}, are determined by the Newton-Raphson method

By eliminating P_2, W_1 e W_2 from the two first equations in the above system the result is

$$f(\underline{x}) = \underline{0} \quad \text{(Eq 17)}$$

where

$$f(\underline{x}) = \begin{bmatrix} P_{boil1} - P_1 - C_1 W_{12}^2 - 2C_1 WL_1 W_{12} - C_1 WL_1^2 \\ P_{boil2} - P_1 - (C_1 - C_{12}) W_{12}^2 + 2C_2 WL_2 W_{12} - C_2 WL_2^2 \end{bmatrix}$$

and

$$\underline{x} = \begin{pmatrix} P_1 \\ W_{12} \end{pmatrix}$$

The Newton-Raphson method is written in a simplified notation as

$$\underline{x}^{(n+1)} = \underline{x}^{(n)} - \underline{J}_f^{-1}\ \underline{f}(\underline{x}^{(n)}) \quad \text{(Eq 18)}$$

where

$$J' = \begin{pmatrix} -1 & -2C_1W_{12} - 2C_1WL_1 \\ -1 & -2(C_2-C_{12})W_{12} + 2C_2WL_2 \end{pmatrix}$$

4 CONCLUSIONS

The present time-response analysis carried over for the components connected to the steam header led to the conclusion that a quasi-static header model might be qualitatively accurate in a plant simulation program if the information extraction frequency is not greater than about one tenth of a second

A complex header model to capture the sound effect is recommended for more detailed use but most of the plant simulations required in the project and operation stage can be fulfilled by the simpler model presently analysed

Computationally the program is very inexpensive and the Newton-Raphson method converges to the solution invariably in less than 3 iterations

5 REFERENCES

1 LAMBERT J D Computational Methods in Ordinary Differential Equation New York, John Wiley, 1973

2 SHINSKEY, F G Process-Control Systems New York, McGraw-Hill 1979

APENDIX

SIMPLIFIED HEADER SIMULATION SOURCE LISTING

AND

EXAMPLE RUN FOR A TURBINE TRIP TRANSIENT

```
{                                                             }
{ PROGRAMA PARA SIMULAR COLETOR COM NEWTON-RAPHSON            }
{          VERSAO   DE 29-10-92           HN                    }
{   ___                                                         }
{                                                          ___  }
{                                                               }
{  'GV1  =-- W1 ---= (P1)-= WL1 =(P11)-  Wtu1 =-(P111) Tur1     }
{                                                          ___  }
{   ___                            Wtc1                         }
{                                  _\/                          }
{                                                               }
{   ___               \/ W12       Tc1                          }
{                                  ___                     ___  }
{                                                               }
{   GV2   -  W2 -- -=(P2)-  WL2 -(P22)-- Wtu2 ==(P222) Tur2     }
{                                   :                      ___  }
{   ___                            Wtc2                         }
{                                  _\/_                         }
{                                                               }
{                                  Tc2                          }
{                                  ___                          }
{                                                               }

Program COLETOR;
uses crt;



const Pi = 3.14159; c=0.00001
var
   i,  sinal              : integer;

   tempo, tempoSim, tempoTrans, W10  W20   temp, deltat,


   Hgv1, Hgv2, H0, H1, H2, H12, HL1  HL2, Hent,

   fator, Pin, W12n, eps, eps1, eps2,
   L1  LL1, Ltu1  L12, f1,
   L2, LL2, Ltu2,
   C1  , CL1, Ctu1, k1   kL1  ktu1, D1   ro1   A1 ,
   C2  , CL2  Ctu2, k2 , kL2, ktu2  D2   ro2 , A2 ,
   C12   k12, D12, ro12, A12 ,
   Pgv1  W1 , P1 , P11 , P111, WL1 , Wtu1  Wtc1  W12,
   Pgv2  W2 , P2 , P22   P222  WL2   Wtu2, Wtc2                : double;
   arq1  arq2, arq3, arq4    text;
label denovo;
```

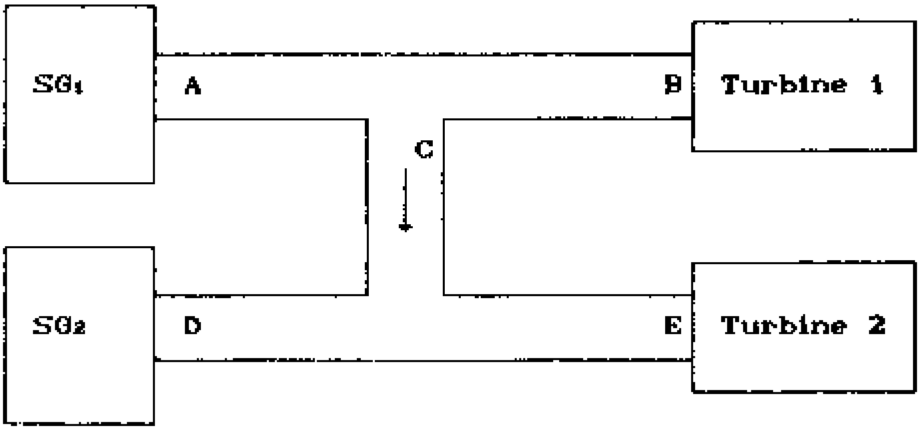
```
FUNCTION Fx1(Pgv1,x,y double)   double,
begin
  Fx1 := Pgv1   x   C1*y*abs(y)   2 0*C1*WL1*y   C1*WL1*abs(WL1);
end,

FUNCTION Fx2(Pgv2,x,y double)   double,
begin
  Fx2  - Pgv2   x   (C2-C12)*y*abs(y) + 2 0*C2*WL2*y   C2*WL2*abs(WL2);
end,

FUNCTION Jfx11inv(y double)   double,
begin
  Jfx11inv = - 2 0*(C2-C12)*y + 2 0*C2*WL2
end,

FUNCTION Jfx12inv(y double)   double,
begin
  Jfx12inv - 2 0*C1*y + 2 0*C1*WL1,
end,

FUNCTION del(y double)   double,
begin
  del  - 2 0*( (C2 C1 C12)*y   C1*WL1   C2*WL2 ),
end,

BEGIN
assign (arq1, C \UTIL\GRAPHER\COLETOR1 DAT ),
rewrite(arq1),
assign (arq2, C \UTIL\GRAPHER\COLETOR2 DAT ),
rewrite(arq2),

{ dados geométricos do coletor de vapor }
     ro1  = 19 30 ,
     ro2  - 19 30 ,
     ro12 - 19 30 ,      ff  - 0 03,

     L1    := 8 00,   LL1 := 8 0, Ltu1  = 0 1,
     L2       8 00,   LL2 := 8 0, Ltu2    0 1,
     L12    - 5 0,
     D1     = 6 0*0 0254,
     k1  =(3 0 + ff*L1/D1), kL1 - (1 0+ff*LL1/D1), ktu1:- (0 00+ff*Ltu1/D1);
     D2     = 6 0*0 0254,
     k2  - k1,              kL2   kL1,                ktu2:= ktu1;
     D12    = 4 0*0 0254,
     k12 =(3 0 + ff*L12/D12);

     A1     = 0 25*Pi*D1*D1,
```

```
    A2     - 0 25*Pi*D2*D2,
    A12    = 0 25*Pi*D12*D12,
    C1     0 5*k1  /(ro1 * A1*A1  ) ,
    C2   = 0 5*k2  /(ro2 * A2*A2  ) ,
    C12  - 0 5*k12 /(ro12* A12*A12) ,

{ condição de contorno inicial }
    Pgv1   = 38 50JE+5,
    Pgv2   = 38 5E+5,

    Wtu1   = 8 999, Wtc1  = 0 0,
    Wtu2   = 9 0  , Wtc2 := 0 0,

    WL1    - Wtu1 + Wtc1,
    WL2    - Wtu2 + Wtc2,
    W12   :- WL1    Wtu1,

    H0     2 816E+6,
    H1     H0,
    H2   - H0,
    Hgv1   H0,
    Hgv2   H0,
    H12  = H0,
    HL1  - H0,
    HL2  - H0,

{ valores de chute inicial }
    P1     :  Pgv1,
    P2        Pgv2,

    clrscr,
    W1     WL1 + W12,   W10  - W1,
    W2   - WL2   W12    W20 :- W2,

    tempo    0 0,

    deltat  - 0 02,

    TEMPOSIM  -  1 0,   tempoTrans  = 0 4,

DENOVO
    tempo    tempo + deltat,

   Pgv1  = Pgv1 + 1 7/c*(W10   W1 )*deltat,
   Pgv2  = Pgv2 + 1 7/c*(W20 - W2 )*deltat,

   temp    9 00,
   if (tempo>tempoTrans)  then                    {TRANSIENTE NA SAIDA DO TUBO   2}
     temp    9 0   7 0*(1 0 - exp( (tempo TempoTrans)/0 02));
```

```
Wtul:  temp,

WL1   :  Wtu1 + Wtc1;
WL2    = Wtu2 + Wtc2;

Hgv1   H0*temp/9 00,


  eps1    1 0,  eps2  - 1 0,  i:= 0,

  While (eps1 > 0 1) or (eps2 > 0 00001) do
  begin
    i - i+1,

    P1n  := P1   (Jfx1linv(W12)*Fx1(Pgv1,P1 W12) + Jfx12inv(W12)*Fx2(Pgv2,P1,W12))/de)(W12);

    W12n  = W12  (              fx1(Pgv1,P1,W12) -        Fx2(Pgv2,P1,W12))/de)(W12),

    eps1  - Abs(P1    P1n ),
    eps2  = Abs(W12   W12n),
    P1      P1n ;
    W12  - W12n,
    If W12<0 0 then C12  =   Abs(C12),
    If W12>0 0 then C12  - + Abs(C12),
    W1  := WL1 + W12,
    W2  := WL2   W12,
    If W1 <0 0 then C1   -   Abs(C1 ),
    If W2 <0 0 then C2  :-   Abs(C2 ),

    If i > 20 then begin
      Writeln( MOTIVO DA PARADA  Newton-Raphson do COLETOR nao convergiu );
      sound(700), delay(200), nosound,
      Halt; end,

  end  { while eps }

  CL1     0 5*L1 /(ro1 * A1*A1  ) ;
  CL2  - 0 5*L2 /(ro2 * A2*A2  ) ;
  Ctu1 - 0 5*tu1/(ro1 * A1*A1  ) ,
  Ctu2 - 0 5*tu2/(ro2 * A2*A2  ) ,

  P2  :- P1   C12*W12*Abs(W12),
  P11  - P1 - CL1*WL1*Abs(WL1),
  P22  = P2   CL2*WL2*Abs(WL2),
  P111 = P11  Ctu1*Wtu1*Abs(Wtu1),
  P222 - P22  Ctu2*Wtu2*Abs(Wtu2),

(calculando entalpias com modelo de mistura instantanea }
  H1   - H1 + deltat*(Hgv1   H1 )/W1/(A1*L1*ro1);
  H2  := H2 + deltat*(Hgv2   H2 )/W2/(A2*L2*ro2),
```

```
If W12> 0 0 then Hent   H1  else Hent - H2,
If Abs(W12)>1 0E-7 then H12    H12+ deltat*(Hent - H12)/Abs(W12)/(A12*L12*ro12),

If W12> 0 0 then Hent - H1 else Hent   (W1*H1   WL2*HL2)/WL1,
HL1   - HL1+ deltat*(Hent   HL1)/WL1/(A1*(LL1+Ltu1)*ro1),
If W12< 0 0 then Hent - H2 else Hent - (W2*H2 + WL2*HL2)/WL2,
HL2    HL2+ deltat*(Hent   HL2)/WL2/(A2*(LL2+Ltu2)*ro2),



Writeln(  - --   -  =- -   tempo  - - , tempo 6 2,   == com iteracoes= ,i:3,  *** ),

Writeln( Pgv1   P1   P11   P111   ,
        Pgv1*c 12 8, P1*c 12 8, P11*c 12 8, P111*c 12 8);
Writeln( Pgv2   P2   P22   P222 - ,
        Pgv2*c 12 8, P2*c 12 8, P22*c 12 8, P222*c 12 8),
Writeln(  W1   WL1   Wtu1      - , W1 12 6,               , WL1 12 6, Wtu1 12 6);

Writeln(    W12                              , W12 12 6),
Writeln(  W2   WL2 - Wtu2     -   W2 12 6              , WL2 12 6, Wtu2 12 6);
Writeln(  -    ----     --     -    -    -    --  - -  ------  --  ),
{ Writeln(  H1   HL1             , H1 12 0,               , HL1 12 0),
Writeln(    H12                              , H12 12 0),
Writeln(  H2   HL2               H2 12 0,               , HL2 12 0);
}
Writeln(arq1  tempo,    , W1 12 6,               ,WL1 12 6,
                        ,         W12 12 6
                        , W2 12 6,               ,WL2 12 6 ),
Writeln(arq2, tempo,    , Pgv1*c 12 6,      P1*c 12 8,    ,P111*c 12 8
                  ,      Pgv2*c 12 8,    , P2*c 12 8,    ,P222*c 12 8),
{readln,}

If tempo < tempoSim  then goto DENOVO,

Writeln(  Fim da Rodada  ),
close(arq1),
close(arq2),

END  { PRINCIPAL }
```

TURBINE TRIP TRANSIENT SIMULATION RESULTS



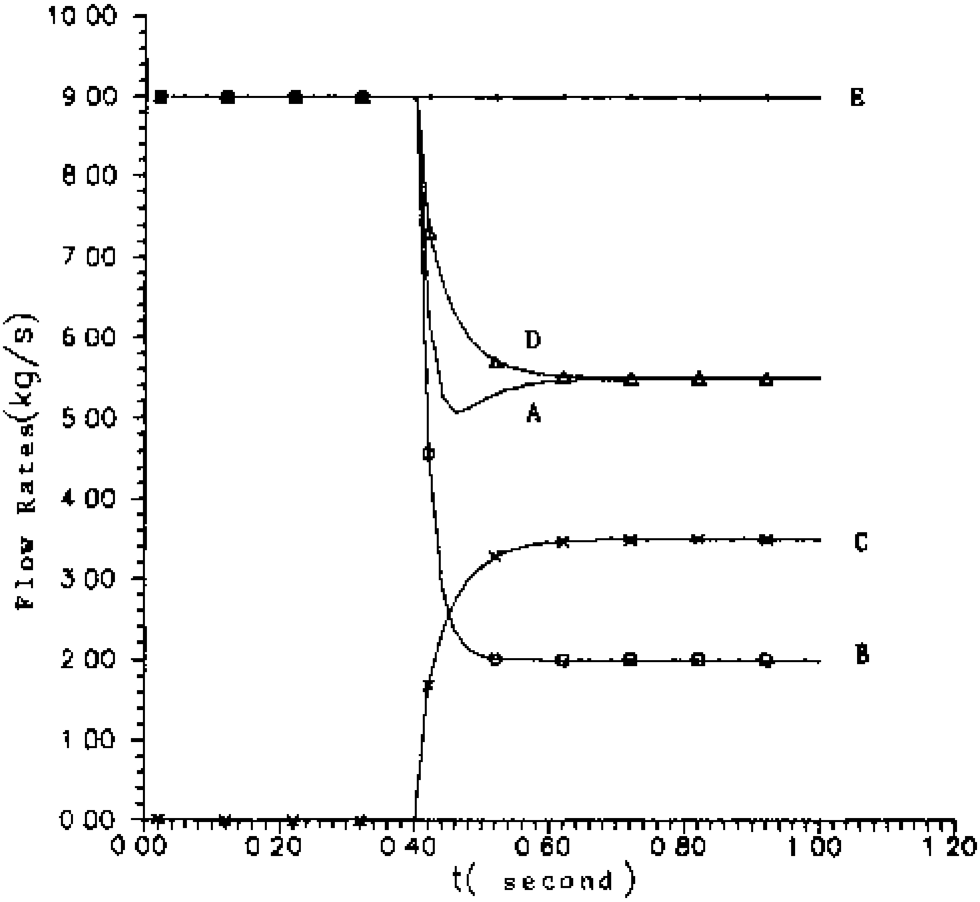


FIGURE A 1 Mass flow rate variations at points A B C D and E for the trip in turbine 1