PROCESS CONTROLLERS OPTIMALITY CONTROL FOR PROCESS STEAM CONDENSATE IN POWER PLANT AND UTILITIES

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ABSTRACT

The research presents process controller investigated for optimal function using process steam condensate stream of power plant and utilities of Port Harcourt refinery industrial data. Flow control institute [FCI] formula for sizing and design was applied for the studies for literature comparison. Various parameters control checks were mathematically verified for optimum in succession of 9-stepwise approach on the cavitation situation through subcritical gas streams flow to critical vapor flow; and is eventually a three-parameter-bound research of process controllers. The plots of the relationships in figures 1, 2, 5, 7 single-parameter plot and composite plots of figures 3, 4, 6, 8, 9, 10, 11, 12, 13 and 14 explicates better comparison of the parameters behaviors. From the studies the profiles shows good trends of optimality and hence, models simulations showed better performance in controllers in process streams plant operations.

Keywords: Process controller, flow control institute formula, Parameters control, process steam condensate, simulations.

INTRODUCTION

There is no process or chemical plant which can be operated without being adequately instrumented. The instrument serves to monitor flows, pressures, temperature and levels. This is very necessary in almost every process in order that the plant engineer or plant operator can see that all parts of the plants are functioning as designed [Richardson & Peacock, 2004].

Additionally, many other quantities which are more specific to the particular process can be recorded and displayed; e.g. the composition of a process stream the heat radiation produced in a crude oil heater or the humidity of a gas streams.

Process control in one form or another is an essential part of any chemical and petrochemical engineering operations. Therefore, in all processes, there arises the necessity of keeping flows, pressures, temperature, concentrations, etc. within certain limits for reasons of safety or running plant at design specifications. Therefore, process controllers are instruments that chemical and petrochemical engineers use to control and adjustment of chemical engineering systems as earlier highlighted above.

Control engineering is the engineering that is concerned with control and adjustment of systems, which need not involve a human operator. There are many kinds of process controllers in control engineering. It has become evident that automatic control is highly desirable, as manual control operation would necessitate continuous monitoring of control variable by a human operator and the efficiency of observation of the operator would inevitably fall off with time. Furthermore, fluctuations in the controlled variable may be too rapid and frequents for manual adjustment to suffice.

In the most simplest form, process controllers can be systems which are designed to carry out control of process which is most often accomplished by measuring the variable, it is required to control (i.e. known as controlled variable), and comparing this measurement with the value of which it is desired to maintain the controlled variable (i.e. the desired value or set point), and adjusting (in a prescribed way until the desired value is attained) some further variable (the manipulated variable) which has a direct effect on the controlled variables [Richardson & Peacock, 2004].

For controller design, it is frequently needed to obtain the steady-state and dynamic state relationships between the particular variables involved.

Hydraulic valves differ from process control valves in application and design. Hydraulic valves are typically used for controlling pressure and therefore have quick opening characteristics. Quick opening valves employ relatively large clearance between the plug and the seat. Alternatively these valves utilize a disc for a poppet plug. Process control valves on the other hand are used for precise control of the fluid flow rate and are of the linear or equal percentage characteristics. These types of valves usually have small clearance between the plug and the seat. Despite these differences, many of the flow phenomena in the hydraulic valve such as recirculation and jet separation and reattachment also occur in the process control valves which are not qualitatively analyzed and hence require an in depth review. Due to the fast progress of the flow visualization and measurement techniques, it becomes possible to observe flows inside a valve and to estimate the performance of a valve. Flow performance study is carried out to find the flow characteristics and flow coefficients. Numerical and experimental investigations are carried out in this field following different methods. After studying the entire flow performance, design improvements are done for design optimization.

Valves and selection criteria

Two of the most common problems facing valve designers when selecting valves for severe service conditions are cavitation and aerodynamic noise. Cavitation are a hydrodynamic flow phenomenon that, if not considered at the time of valve selection, can cause damage to valve components and the pipe network. Aerodynamic noise results from turbulent flow and is relevant only to valve handling gas flow.

[Herbert Miller, 1997] accepted fluid kinetic energy as a selection criterion for Control Valves. His work explains the criterion which involves limits on the fluid kinetic energy exiting through the valve throttling area. The selection criteria is to limit the valve throttling exit fluid kinetic energy to 70 psi (480 kPa) or less and pipe velocity to 15 m/s in order to eliminate the valve problems like unstable forces inside the valve, cavitation, erosion of critical parts, shock waves, unwanted noise and vibration and to meet the system needs. Final selection of the valve and trim type was made through experience and/or by considering one of the following parameters: pressure drop, pressure drop ratio (pressure drop divided by inlet pressure), fluid velocity or as indicated here, the fluid kinetic energy. Low pressure drops are handled with butterfly valves. As the pressure drop increases, a ball valve would be needed. A larger pressure drop would require the linear motion globe/angle type valves. The globe/angle designs incorporate many different valve trims depending upon the level of pressure drop.

Flow Simulation and Performance Characteristics

Flow performance of valves was studied by experimental flow simulation and by flow simulation using software. A control valve model was verified experimentally. The author states through this paper that control valve is predominantly ax symmetric before the plug retracts from the plane of the seat, but then the flow field makes a transition to a three-dimensional pattern after the plug retracts from the plane of the seat. The three-dimensional portions of the flow field do not appear to significantly affect the performance except at the extreme large values of valve opening.

FEEDBACK CONTROLLER

From the point of view of control engineering and chemical engineering process, the following feedback control system schemes are illustrated. Figures 2.1 a, b, c are typical temperature recorder controllers [Sinnott, 2004].

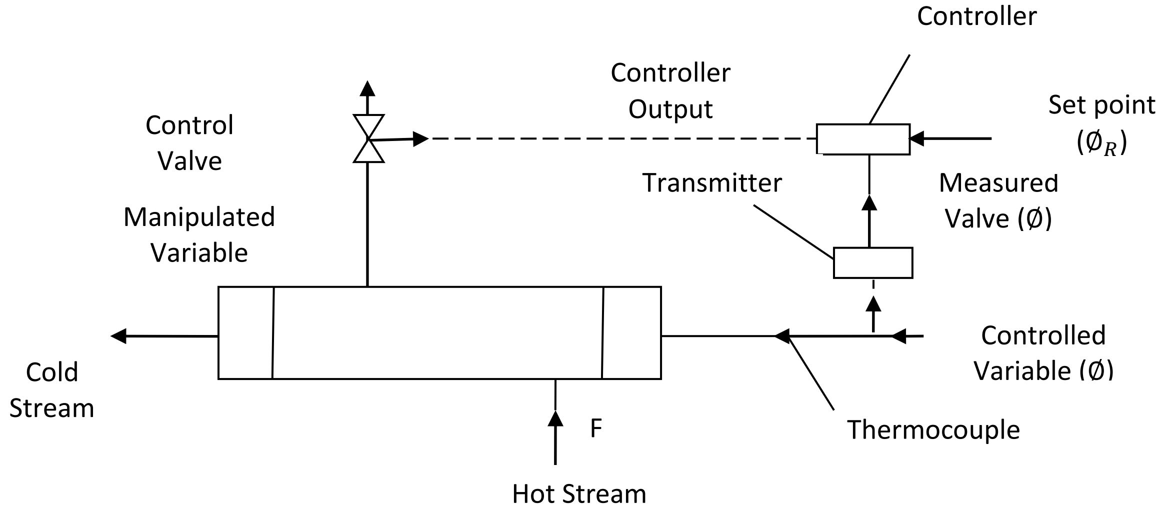

Figure 1a Simple feedback control system illustrating components

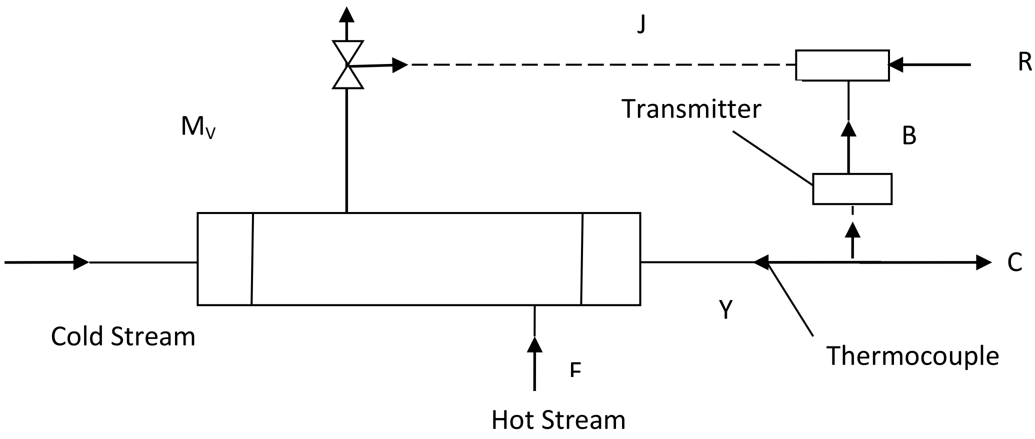

Figure 1b simple feedback control system standard nomenclature

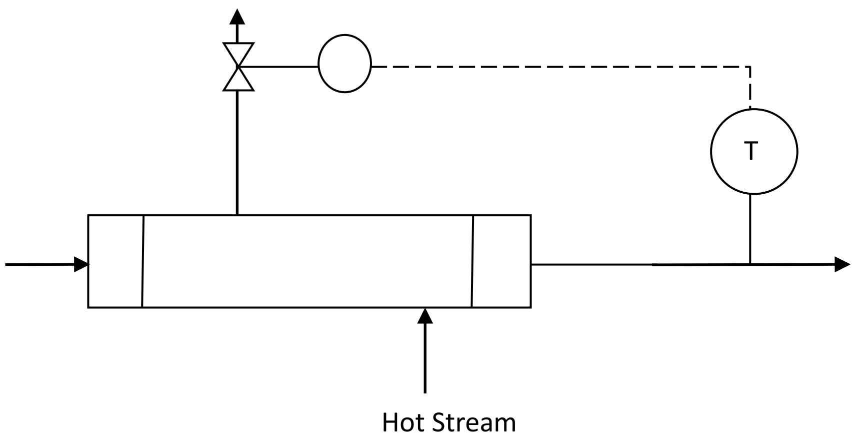

Figure 1c Simple feedback control system representation according to British Standard BS 16

The figures 2.1 a, b, c the function are to control at **Y** the temperature of the process stream leaving the heat exchanger. The temperature θ of the stream at Y (the controlled variable figure 2.1 a – usually denoted by the letter **C** as in figure 2.1b) is measured by means of a thermocouple; the output of which is fed via a suitable signal transmission system to a controller.

The controller consists of two parts. The first compares the measured temperature θ_M (i.e. the measured value – which is indicated usually by the letter B in figure 2.1b with the desired value or set point θ_R generally denoted by R. It is pertinent to note that θ and θ_M will not necessarily be identical at the same instant of time, - particularly when θ varying.
This part of the controller is termed the comparator and produces an error (ε) such that Mathematically:

$$\varepsilon = R - B = \theta_R - \theta_M$$

The second part of the controller provides the required control action. The most common types of control action (in term of fixed parameter control) and their effects upon the controlled variable are described in fixed parameter feedback control action. There are other control strategies, but this study will focus on temperature of process streams; such as the process steam condensate controller optimum function strategies maintaining some design constraints; using the FCI formulas.

It will suffice at present to observe that the controllers produce and output **J** which is a function of ε figure 2.1a and b 2.1b.

The controller output signal may be in the PID (Proportional Integrated Differential) control form of an air pressure from a pneumatically operated controller or a current or voltage supplied by an electronic controller or by a microprocessor simulating the appropriate control action. This signal is transmitted to the control valve (which is called the final control element) and is connected in such a way that the value starts to open further when ε becomes positive, i.e. when $\theta_R < \theta_M$ the control system calls for more heat to be supplied by increasing the flow rate **F** of the hot stream. When ε is negative i.e. $\theta_R > \theta_M$ than the valve starts to shut.
When θ_M is at the set point i.e. $\theta_M = \theta_R$ then R = B and $\varepsilon = 0$.
In this instance there is no control action and the positive of the value. Steam does not change.

There are two principal functions of a control loop of this kind, i.e. two reasons why a difference might occur between θ_M and θ_R thus producing an error. The first is that changes may occur in such variables as the cold or hot stream inlet temperature and cold stream inlet flow. Even **F** may vary due to reasons other than the setting of the control valve. All these are termed load changes, or are collectively described as load. Control or controlled variable in the face of variations in load is often termed the regulator problem or the load rejection case. The second reason is that we may wish to raise or lower θ for various production or operational reasons. This can be achieved by raising or lowering θ_R as desired - so creating a positive or negative error respectively. The control system will seek to minimize ε, i.e. to bring θ_M to the new value of θ_R. This is called the sevo-problem or the set point following case. It is not possible (or necessary) for θ_M or, indeed, θ to adjust too θ_R precisely under all forms of control action.

PID Controller
There are three principal types or modes of control action which are more generally employed, viz: Proportional (P), Integral (I), Derivative (D). In the first, the controller produces an output signal J which is proportional to the error, i.e.

$$J = J_o + K_{c^{\varepsilon}}$$

Where K_c is the proportional gain or sensitivity, and
J_o is the controller output when $\varepsilon = 0$.
Hence, with proportional control, the greater the magnitude of the error the larger is the corrective action applied.

It is generally assumed when considering control system dynamics that at t < 0 the control system is at a steady state process and that ε is zero. Hence, it is necessary to include the term J_o in the controller output in order to maintain the final control element (almost invariably a control valve) at its steady-state setting when $\varepsilon = 0$. The insertion of J_o in the control algorithm can be considered as setting the operating point for the controller and thus be a possible means of providing so-called bumpless transfer from manual to automatic control.

The integral and derivative modes are normally used in conjunction with the proportional mode. Integral action (or automatic reset) gives an output which is proportional to the time integral of the error. Proportional plus integral **(PI)** action may be represented mathematically:

$$J = J_o + K_{c^\varepsilon} + K_I \int_0^t \varepsilon dt$$

Where, K_I is a constant.

Derivative action (often termed rate control) gives an output which is proportional to the derivative of the error. Hence, for **PD** control:

$$J = J_o + K_{c^\varepsilon} + K_D \frac{d\varepsilon}{dt} \qquad \textbf{[4]}$$

Where K_D is a constant

Most frequently, all three modes are used together as **PID** controller, mathematically:

$$J = J_o + K_{c^\varepsilon} + K_I \int_0^t \varepsilon dt + K_D \frac{d\varepsilon}{dt}$$

The same relationships (whether in continuous or discrete form) are said to describe fixed parameter controllers when the parameters (K_c, K_I and K_D) and is left unaltered throughout the entire period of control action. Controllers in which the parameters are continually and automatically adjusted to take account of changing process conditions and dynamics are termed adaptive system control (Bennett, 1988); (Singh, Elloy, Mezencev & Munro, 1980)

MATERIALS AND METHOD

MATERIALS

The research objective is to carry out investigative studies applying the sizing formula of Fluid Control Institute formula, Inc. to size control valves for the cavitation, subcritical and critical flow situations stated below. Thereafter, show how accurate the FCI formulas predictions performed; while utilizing process data of the power plant & utilities of the refinery process steam condensate data for testing the formulas for optimum function.

Adopting a 9–point criterion for investigations on the cavitation, occurring situation through subcritical gas streams flow to critical vapor flow; and is eventually a three-parameter-bound research of process controllers [Nicholas, 22004].

POWER PLANT & UTILITIES PROCESS DATA

Cavitation: Select a control valve for a situation where cavitation may occur.

The fluid is process steam condensate; inlet pressure P_1 is 167psia (1151.5kPa);

ΔP is 105lb/in^2 (724.0kPa);

Inlet temperature T_1 is 180^{O}F (82.2^{O}C);

Vapour pressure, $\boldsymbol{P_v}$ is 7.5 Psia (51.7kPa).

Sub-critical gas flow: determine the gas capacity required at these conditions;

Fluid is air;

Flow Qg is 160,000std ft^3/h (1.3std m^3/s);

Inlet pressure P_1 is 275psia (1896kPa);

ΔP is 90lb/in^2 (620.4kPa);

Gas temperature T_1 is 60^{O}F (15.6^{O}C)

Critical vapour flow: a heavy-duty angle valve is suggested for a steam pressure-reducing application. Determine the capacity required and compare and alternate valve type.

The fluid is saturated steam;

Flow W is 78,000lb/h (9.8kg/s);

Inlet pressure P1 is 1260psia (8688kPa);

And outlet pressure P_2 is 300psia (2068.5kPa).

Method
Theoretical concepts
Setting out criteria for calculations:
Choose the valve type and determine its critical-flow factor for the cavitation situation to take place or not.

1. Compute the maximum allowable pressure differentiate for the valve.
2. Select another valve and repeat the cavitation calculation (i.e. confirmatory).
3. Apply FCI formula for sub-critical flow determination.
4. Compute C_v using the unified gas-sizing formula.
5. Determine C_f for critical vapour flow.
6. Compute critical pressure drop in the valve.
7. Determine the value of C_v.
8. Refer to table for some data for the work.
9. Plot fraction of critical flow rate against Y.

The research method is Fluid Control Institute [FCI] formulas [Henry, 2002].

Sizing / Design of Controllers:
The step-wise procedures are adequately maintained for the design criteria.
Step1:
Compute the valve flow coefficient (C_v)
The value flow coefficient C_v, is a function of the maximum steam flow rate through the value and the pressure drop that occurs at this flow rate. When choosing a control valve for a process control system, the usual procedure is to assume a maximum flow rate for the value based on a considered judgment of the overload the system will carry. This value will not exceed 25 percent overload.

$$Overload(P) = 0.20\dot{m} \qquad [1]$$

Where $\dot{m}$ = maximum overload in kg/s.
Thus the rated capacity of the value is given by:

$$Q = \dot{m} + P = \dot{m}(1.20) = 1.20\,\dot{m} \qquad [2]$$

Where Q = Rated capacity
The pressure drop across a steam control value is a function of the value design, size and flow rate.

$$\Delta P = f(valve\ design, size\ and\ flow\ rate) \qquad [3]$$

Where ΔP = pressure drop; KP_a
The most accurate pressure drop estimate that is usually available is that given in the valve manufacturer's engineering data for a specific valve size, type and steam flow rate.
Thus the valve flow coefficient C_v is given mathematically as:

$$C_v = \frac{WK}{3(\Delta PP_2)^{0.5}} = \frac{WK}{3}(\Delta PP_2)^{-0.5} \qquad [4]$$

Where; W = steam flow rate, in Kg/S
$K = 1 + (0.0007F\ superheat\ of\ the\ steam)$
ΔP = Pressure drop P_2 = Control-value outlet pressure at maximum steam flow rate, kg/m^2 (N/m^2)
Step 2: Compute low-load steam flow rate:
This is mathematically stated as:

$$W = \frac{3(C_v\Delta PP_2)^{0.5}}{K} \qquad [5]$$

Again the Nominal diameter of the value analyzed as above is given as:

$d = \left[\frac{C_v}{10}\right]^{0.5}$ **[6]**

Again, to sized control values for liquids, a similar procedure and relation adopted as thus:

$C_v = {}^{VG}/_{\Delta P}$ **[7]**

Where;

V = Flow rate through the valves, $\left[\frac{m^3}{s}\right]$

$\Delta P =$ Pressure drop across the value at maximum flow rate, kg/m^2

G = Specific gravity of the liquid.

To size control valves for gases, this relation below is adopted

$C_v = \frac{Q[GT_G]^{0.5}}{[1360\ (\Delta PP_2)^{0.5}]}$ **[8]**

Where;

$C_v =$ Value flow coefficient

Q = Gas flow rate, $\left[\frac{m^3}{s}\right]$ at 1 atm of 15.6^{0}C

T_G = Temperature of the flowing gas, in ^{0}C

Note when the valve outlet pressure, P_2, is less than $0.5P_1$ i.e.

If $P_2 < 0.5P_1$ **[9]**

Where;

$P_1 =$ The valve inlet pressure, P_a

Thus, $(\Delta PP_2)^{0.5} = \left[\Delta P\left({}^{P_1}/_2\right)\right]^{0.5} = [0.5^*\Delta PP_1]^{0.5} = 0.707\sqrt{\Delta PP_1}$

$(\Delta PP_2)^{0.5} = 0.7071^*\sqrt{\Delta PP_1}$ **[10])**

Again, to size valve for vapor, other than steam, the following relations are applied:

$C_v = \frac{W}{63.4}\ (V_2/\Delta P)^{0.5}$ **[11]**

Where;

W = vapor flow rate [kg/s]

$V_2 =$ Specific volume of the vapour at the outlet pressure, P_2, [m^3/kg].

When the control valve handles a flashing mixture, water and steam, compute C_v using equation [7] above

The allowable pressure drop = ΔPR **[12]**

Where;

R = pressure drop connection factor which is the function of the difference between the temperature and actual one.

Step 3: Control valve characteristics and range-ability

The design of a control valve installed in a process system in which flow varies, say 100 to 20 percent and pressure drop rises from say 5 to 80% within the system must be characterized and have range with the design steps below:

Step 1:

Compute the Required Valve Range-ability

This is gotten with the design equation as thus:

$R = \left(\frac{Q_1}{Q_2}\right)\left(\frac{\Delta P_2}{\Delta P_1}\right)^{0.5}$ **[13]**

Where;

R = the valve range-ability

$Q_1 =$ Valve initial flow in the % of the total flow

$Q_2 =$ Valve final flow, in % of total flow

P_1 = % final pressure drop across the valve

Step 2:
Select the Type of valve characteristics to use
Table 1 Control Valve Characteristics

Valve type	Typical flow range-ability	Stem movement
Linear	12-1	Equal steam movement for equal flow change
Equal %	30-1 to 50-1	Equal stem movement for equal % flow chart
On-off	Linear for first 25% of travel; on-off there after	Same as linear up to on-off range

Table1 shows equal % valve must be used if a range-ability of 20% is required. Such a valve have equal stem movements for equal-percentage changes in flow at a constant pressure drop based on the flow occurring just before the change is made.
Step 3:
Show valve effective characteristics related to pressure drop
If a valve is to operate at a constant load without changes in the flow rate, the characteristic of the valve is not important, since only one operating point of the valve is used.
There is an economic compromise in the selection of every control valve. Where possible, the valve pressure drop should be as high as needed to give good control.
Cavitation, Sub-Critical and Critical Flow Consideration in Controller Selection
Using the sizing formulas of the Fluid Controls Institute [FCI] size control valves for the cavitation, subcritical and critical flow situation are made thus:

- **Cavitations:** Select a control-valve where cavitations may occur with given conditions of temperature, pressure, vapour pressure (Po) and Pressure drop.
- **Critical Valve Flow:** A heavy angle valve is suggested for a steam pressure-reducing application. Determine the capacity required and compare on alternative valve type.

The following steps are procedures used for the design and calculation of cavitation, subcritical and critical consideration in controller selection.
Step 1:
Choose the valve type and determine its critical flow factor for the cavitation situation.
A butterfly control valve is acceptable on a steam condensate application.
Thus:
$C_f = 0.68$ For a butterfly valve with 60^0 operation.
Where;
$C_f =$ Critical flow factor
Step 2:
Compute the maximum allowable pressure differential for the value.
The relation: $\Delta P_m = C_f^2 \, (P_1 - P_v)$ **[14]**
Is used for such computation, where;
$\Delta P_m =$ Maximum allowable pressure differential, kg/m^2
$C_f =$ Critical flow factor
$P_1 =$ Inlet pressure, KPa
$P_2 =$ Vapour pressure, KPa
Equation [14] can only applicable if the actual pressure drop exceeds the allowable pressure drop, hence cavitation occurs.
Step 3:
Select another valve and repeat the cavitation calculation

Now, $C_f = 0.68, 0.72, \dots 0.95$ to get the various ΔP_m this situation can only arise when single-port top guided value, double port value, etc. are also selected for the computation. For the two valves, the single-port top guided valve offers lower seat leakage but the double-port top guided valve offers the possibility of a more economical actuator, especially in larger valve sizes.

Step 4: Apply the FCI formula for subcritical flow.

This is given by:

$$C_v = \frac{Q_g}{\left[1360 \left(\frac{\Delta P}{GT}\right)^{0.5}\right]\left[\frac{(P_1+P_2)}{2}\right]^{0.5}} \qquad [15]$$

Where;

$C_v =$ Valve flow coefficient

$Q_g =$ Gas flow, [m^3/3]

$\Delta P =$ Pressure differential, [kg/m^2]

G = Specific gravity of gas at 1 atm and 15.6^{0}C

T = Absolute temperature of the gas, k.

Step 5:

Compute C_v using the unified gas-sizing formula for greater accuracy, many engineers use the unified gas sizing formula.

In case a single-port top-guided valve installed open to flow:

$C_f = 0.90$

$$y = \left(\frac{1.63}{C_f}\right)\left(\frac{\Delta P}{P_1}\right)^{0.5} \qquad [16]$$

Where;

y = defined by the equation, unit less.

$$C_v = \frac{Q_g\,(GT)^{0.5}}{[834\, C_f\,(y - 0.148y^3)\,]} \qquad [17]$$

The error should not exceed 10%, hence good estimates.

Step 6:

Determine $C_f = 0.55$ for such case.

Step 7:

Compute the critical pressure drop in the valve

The design equation is given as:

$$\Delta P_c = 0.5\,(C_f)^2\, P_1 \qquad [18]$$

Where;

$\Delta P_c =$ Critical pressure drop, kPa

Step 8:

Determine the value of C_v

The design equation is given by:

$$C_v = \frac{W}{1.83\, C_f P_1} \qquad [19]$$

For a more economical choice, a single-port top guided valve is installed.

RESULTS AND DISCUSSION

Program calculation cum analysis of design model

$\dot{m}_1 = 0.9kg/s$ for steam

$P_1 = 551.6kP_{ag}$steam

$\dot{m}_2 = 0.038kg/s$
$P_2 = 275.8\ kP_{ag}$
$P_o = 689.5kP_{ag}$ of steam is available for heating
Assuming 5-155 of ΔP
$C_v = \frac{WK}{3(\Delta P\ P_2)} 0.5$
i.e. $\Delta P = 10\%\ of\ 551.6$
K = 1, i.e. $1 + (0.0007 \times F\ super\ heat)$
$P_R = 94.7 = 652.9565\ kP_{ag}$
$w = 0.19$
$W = \frac{3(C_v \Delta P P_2)^{0.5}}{K}$
$\Delta P = 80 - 40P_{sig} = 275.8\ kP_{ag}$
$W = 0.075kg/s.l$
Hence, 5% - 95%, C_v will be given as range.

d (nominal diameter calculation):

$d = \left({C_v}/{10} \right)^{0.5} \times 0.025\ m.$ ie 1inch = 2.5cm = 0.025m.

To size a C_v

For liquid

1. $C_v = v\ (G/\Delta P)$
   G = specific gravity of the liquid
   Where $V = {m}/{\rho}$
   $C_v = \frac{\dot{m}}{\rho}\ (G/\Delta P)$

2 **For Gases**

$C_v = \frac{Q\ (GT_a)^{0.5}}{1360\ (\Delta P P_2)^{0.5}}$

Where Q = gas flow rate in ${m^3}/{s}$

$T_a = 15.6^0 C\ (60°_F)$
$P_2 = 101.4\ kp_a(14.7P_{sia})$
$NB\colon {P_1}/{2} = (\Delta P P_2)^{0.5}\colon if\ P_2 < 0.5P_1$

2. **C_v for vapor not steam**

$C_v = \frac{W \left({V_2}/{\Delta P} \right)^{0.5}}{63.4}$
$If\ P_2 < 0.5\ P_1$

Then, ${P_1}/{2}$ replaces ΔP in the above equation [ ]

Control valve characteristics and rangeability

$Q_1 = 100\%$
$Q_2 = 20\%$
$\Delta P = 5\%\ to\ 80\%$

$R = \left({Q_1}/{Q_2} \right) \left({\Delta P_2}/{\Delta P_1} \right)^{0.5}$ Range ability

$= \left({100}/{20} \right) \left({80}/{5} \right)^{0.5}$

= 20
Valve characteristics depends on the range ability
Cavitation, Subcritical and critical flow
Fluid type steam condensate

$$\left.\begin{aligned} P_1 &= 115.5\ kp_a \\ \Delta P &= 724.0\ kp_a \\ T_1 &= 82.2^{\circ}C \\ P_v &= 51.7\ kp_a \end{aligned}\right\} \text{ Cavitations conditions}$$

For subcritical gas flow conditions

$Q_g = 1.3\, {}^{m^3}/_{s}$

$P_1 = 1896\ kp_a : \Delta P = 620.4 kp_a, T_1 = 15.6^{\circ}C$

Critical vapor flow conditions
Suggestion: Heavy duty angle value is needed
The fluid is saturated
$W = 9.8 kg/s$
$P_1 = 8688\ kp_a$
$P_2 = 2068.5\ \ kp_a$
Procedure:
$C_f = 0.68$ for a butterfly value with 60^0 operation

1. $\Delta P_m = C_f^2\ (P_1 - P_v)$
$(0.68)^2\ (1157.5 - 51.7) = 510.2 kp_a$

Since actual pressure drop is 724.0 kPa exceeds the allowable drop, $510.2 kPa$ cavitations will occurs
i.e. cavitation will occurs if $\Delta P < \Delta P_m$ (maximum allowable pressure differential for value).
Take: $C_f = 0.00: 0.02: 1.00$
For single – port top-guided value with flow to open plus
$\left(C_f (P_1 - P_v) = \Delta P_m\right)$
Same values $C_f = 0.90$
For subcritical gas flow

2. $C_v = \frac{Q_g}{[1360(\Delta P/GT)^{0.5}]\left[{}^{P_1+P_2}/_{2}\right]^{0.5}} = \frac{160{,}000}{[1360(90/520)^{0.5}]\ [275+185]^{0.5}}$
$\Delta P = 10: 10: 100 \qquad = \qquad 18.6$
Compute for C_v using unified gas-sizing formula

3. $y = \left(\frac{1.63}{C_f}\right) \left({}^{\Delta P}/_{P_1}\right)^{0.5}$
$C_f = 0.90$: → Single-port top-guided value
$y = \left(\frac{1.63}{0.90}\right) \left({}^{90}/_{275}\right)^{0.5} = 1.04$

$C_f = 0.68$ → Butterfly control value
$Y = \left(\frac{1.63}{0.68}\right)\left({}^{90}/_{275}\right)^{0.5}\ 1.37$
$C_f = 0.02: 2: 100$

4. $C_v = \frac{Q_g (GT)^{0.5}}{[834\ C_f \times (y - 0.148 y^3) P_1]}$

For instance $C_v = \frac{160{,}000(520)^{0.5}}{[834(0.90)(0.87)275]} = 20.4$

5. $\Delta P_c = 0.5\left(C_f\right)^2 P_1$

$C_f = 0.55$
$\Delta P_c = 0.5(0.55)^{0.5} \times 8688 = 1316.9\ kPa$
6. $C_f = \frac{W}{1.83 C_f P_1} = \frac{78{,}000}{1.83(0.90)(1260)} = 37.6$

Summary of Program Results

Table 1 Results for valve flow coefficient with inlet and outlet Pressures, P_1 and P_2, Flow rate, W, Pressure drop ΔP, K and Percentage flow (%).

%	P_1	ΔP	P_2	W	K	C_v
0.1	80	8	94.7	1500	1	18.16561
0.2	80	16	94.7	1500	1	12.84503
0.3	80	24	94.7	1500	1	10.48792
0.4	80	32	94.7	1500	1	9.082806
0.5	80	40	94.7	1500	1	8.123908
0.6	80	48	94.7	1500	1	7.41608
0.7	80	56	94.7	1500	1	6.865956
0.8	80	64	94.7	1500	1	6.422514
0.9	80	72	94.7	1500	1	6.055204
1	80	80	94.7	1500	1	5.744471

Table 2 Results of nominal diameter, d varying with C_v and flow rate, W

K	C_v	ΔP	P2	W	d(m)
1	18.16561	40	54.7	0.075426	0.033695
1	12.84503	40	54.7	0.063426	0.028334
1	10.48792	40	54.7	0.057312	0.025603
1	9.082806	40	54.7	0.053335	0.023826
1	8.123908	40	54.7	0.050441	0.022533
1	7.41608	40	54.7	0.048193	0.021529
1	6.865956	40	54.7	0.046371	0.020715
1	6.422514	40	54.7	0.044849	0.020035
1	6.055204	40	54.7	0.043547	0.019454
1	5.744471	40	54.7	0.042415	0.018948

Table 3 Results control valves for liquids and gases,
C_{VL} and C_{VG} varying with pressure drop for $P_2 > 0.5P_1$

ΔP	P2	A	V	T_a	C_{VL}	C_{VG}
8	94.7	27.52453	180	520	22.5	2.783118
16	94.7	38.92557	180	520	11.25	1.967962

24	94.7	47.67389	180	520	7.5	1.606834
32	94.7	55.04907	180	520	5.625	1.391559
40	94.7	61.54673	180	520	4.5	1.244648
48	94.7	67.42106	180	520	3.75	1.136203
56	94.7	72.82307	180	520	3.214286	1.05192
64	94.7	77.85114	180	520	2.8125	0.983981
72	94.7	82.5736	180	520	2.5	0.927706
80	94.7	87.04022	180	520	2.25	0.880099

Table 4 For $P_2 < 0.5P_1, the\ constant\ a\ is\ replace\ \ 0.5P_1$

P_1	$P_1/_2$	V	T_a	ΔP	C_{VL}	C_{VG}
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.65375	0.754528
80	40	180	520	108.8435	1.653751	0.754528

Table 5 When $P_2 > 0.5P_1$, C_{VV} is obtained and $P_2 < 0.5P_1, C_{VV1}\ is\ obtained$

W	V_2	ΔP	C_{VV}	P_2	b=(P1)^2/(4*P2)	C_{VV1}
1800	0.0125	4	1.587114	14.7	108.8435374	0.30425444
1800	0.025	8	1.587114	14.7	108.8435374	0.43028076
1800	0.0375	16	1.374481	14.7	108.8435374	0.52698415
1800	0.05	24	1.295874	14.7	108.8435374	0.60850889
1800	0.075	32	1.374481	14.7	108.8435374	0.74526814
1800	0.1125	40	1.505669	14.7	108.8435374	0.91276333
1800	0.1625	48	1.651921	14.7	108.8435374	1.097005
1800	0.2375	56	1.848932	14.7	108.8435374	1.32621437
1800	0.35	64	2.099555	14.7	108.8435374	1.60996319
1800	0.5125	72	2.395322	14.7	108.8435374	1.948179
1800	0.75	80	2.748963	14.7	108.8435374	2.35674479

Note: When $P_2 > 0.5P_1$, C_{VV} is obtained and $P_2 < 0.5P_1, C_{VV1}\ is\ obtained$

Table 6 Summary of Excel Results of Control Valve Characteristics and Range ability, R = 20

C_f	P_1	P_v	ΔP_m	ΔP	$C_{v,S}$	y	$C_{v,y}$	ΔP_c	C_v
0.02	1151.5	51.7	0.43992	10	55.93953	46.62432	-0.05319	1.7376	1691.387
0.04	1151.5	51.7	1.75968	20	39.55522	23.31216	-0.21478	6.9504	845.6935
0.06	1151.5	51.7	3.95928	30	32.2967	15.54144	-0.49097	15.6384	563.7956

0.08	1151.5	51.7	7.03872	40	27.96977	11.65608	-0.89283	27.8016	422.8467
0.1	1151.5	51.7	10.998	50	25.01692	9.324864	-1.43736	43.44	338.2774
0.12	1151.5	51.7	15.83712	60	22.83722	7.77072	-2.14948	62.5536	281.8978
0.14	1151.5	51.7	21.55608	70	21.14316	6.660617	-3.06513	85.1424	241.6267
0.16	1151.5	51.7	28.15488	80	19.77761	5.82804	-4.23644	111.2064	211.4234
0.18	1151.5	51.7	35.63352	90	18.64651	5.18048	-5.74039	140.7456	187.9319
0.2	1151.5	51.7	43.992	100	17.68963	4.662432	-7.69419	173.76	169.1387
0.22	1151.5	51.7	53.23032	110	16.8664	4.238575	-10.284	210.2496	153.7624
0.24	1151.5	51.7	63.34848	120	16.14835	3.88536	-13.8226	250.2144	140.9489
0.26	1151.5	51.7	74.34648	130	15.51483	3.586486	-18.8779	293.6544	130.1067
0.28	1151.5	51.7	86.22432	140	14.95047	3.330309	-26.5956	340.5696	120.8134
0.3	1151.5	51.7	98.982	150	14.44353	3.108288	-39.6842	390.96	112.7591
0.32	1151.5	51.7	112.6195	160	13.98488	2.91402	-66.4477	444.8256	105.7117
0.34	1151.5	51.7	127.1369	170	13.56733	2.742607	-150.654	502.1664	99.49335
0.36	1151.5	51.7	142.5341	180	13.18507	2.59024	2431.187	562.9824	93.96594
0.38	1151.5	51.7	158.8111	190	12.83341	2.453912	156.8149	627.2736	89.02037
0.4	1151.5	51.7	175.968	200	12.50846	2.331216	87.18169	695.04	84.56935
0.42	1151.5	51.7	194.0047	210	12.20701	2.220206	63.07755	766.2816	80.54224
0.44	1151.5	51.7	212.9213	220	11.92635	2.119287	50.88362	840.9984	76.88122
0.46	1151.5	51.7	232.7177	230	11.6642	2.027144	43.54037	919.1904	73.53856
0.48	1151.5	51.7	253.3939	240	11.41861	1.94268	38.64574	1000.858	70.47446
0.5	1151.5	51.7	274.95	250	11.18791	1.864973	35.15814	1086	67.65548
0.52	1151.5	51.7	297.3859	260	10.97064	1.793243	32.5528	1174.618	65.05334
0.54	1151.5	51.7	320.7017	270	10.76557	1.726827	30.53667	1266.71	62.64396
0.56	1151.5	51.7	344.8973	280	10.57158	1.665154	28.93326	1362.278	60.40668
0.58	1151.5	51.7	369.9727	290	10.38771	1.607735	27.62996	1461.322	58.32369
0.6	1151.5	51.7	395.928	300	10.21311	1.554144	26.55153	1563.84	56.37956
0.62	1151.5	51.7	422.7631	310	10.04704	1.50401	25.64585	1669.834	54.56087
0.64	1151.5	51.7	450.4781	320	9.888806	1.45701	24.87563	1779.302	52.85584
0.66	1151.5	51.7	479.0729	330	9.737823	1.412858	24.21353	1892.246	51.25415
0.68	1151.5	51.7	508.5475	340	9.593551	1.371304	23.63904	2008.666	49.74667
0.7	1151.5	51.7	538.902	350	9.455507	1.332123	23.13647	2128.56	48.32534
0.72	1151.5	51.7	570.1363	360	9.323255	1.29512	22.69366	2251.93	46.98297
0.74	1151.5	51.7	602.2505	370	9.196402	1.260117	22.30097	2378.774	45.71316
0.76	1151.5	51.7	635.2445	380	9.07459	1.226956	21.95072	2509.094	44.51018
0.78	1151.5	51.7	669.1183	390	8.957494	1.195495	21.63672	2642.89	43.3689
0.8	1151.5	51.7	703.872	400	8.844817	1.165608	21.35387	2780.16	42.28467
0.82	1151.5	51.7	739.5055	410	8.736287	1.137179	21.098	2920.906	41.25334
0.84	1151.5	51.7	776.0189	420	8.631657	1.110103	20.86562	3065.126	40.27112
0.86	1151.5	51.7	813.4121	430	8.530699	1.084287	20.65382	3212.822	39.33458
0.88	1151.5	51.7	851.6851	440	8.433202	1.059644	20.46014	3363.994	38.44061
0.9	1151.5	51.7	890.838	450	8.338973	1.036096	20.28248	3518.64	37.58638
0.92	1151.5	51.7	930.8707	460	8.247834	1.013572	20.11905	3676.762	36.76928
0.94	1151.5	51.7	971.7833	470	8.159619	0.992007	19.96831	3838.358	35.98696

0.96	1151.5	51.7	1013.576	480	8.074176	0.97134	19.82893	4003.43	35.23723
0.98	1151.5	51.7	1056.248	490	7.991362	0.951517	19.69976	4171.978	34.5181
1	1151.5	51.7	1099.8	500	7.911044	0.932486	19.57979	4344	33.82774

Table 7 Summary Results for Fraction of Critical Flow Rate C_f and y

Y	$y-0.148y^3$	C_f
0	0	0
0.2	0.198816	0.2
0.4	0.390528	0.4
0.6	0.568032	0.575
0.8	0.724224	0.7
1	0.852	0.85
1.2	0.944256	0.95
1.4	0.993888	1

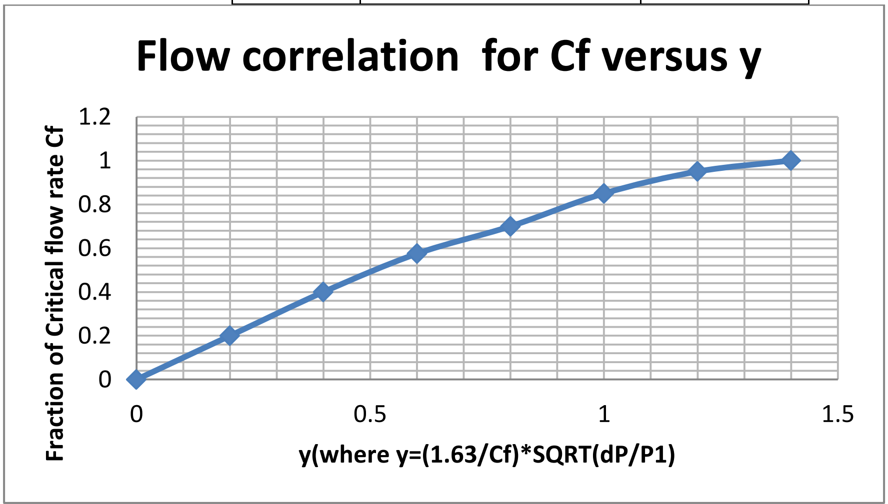

Figure 1 Flow correlations established from actual data for many valve configurations at maximum valve opening.

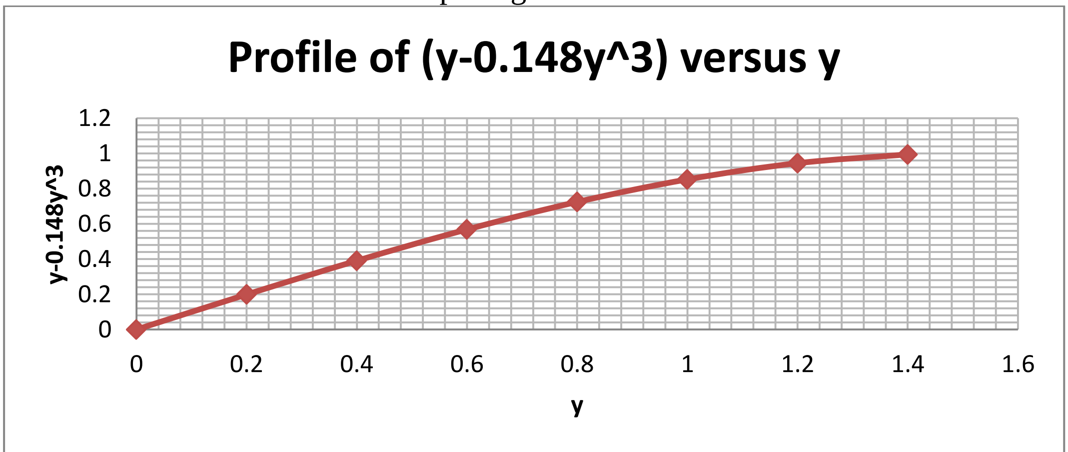

Figure 2 Plot of correlation–factor values

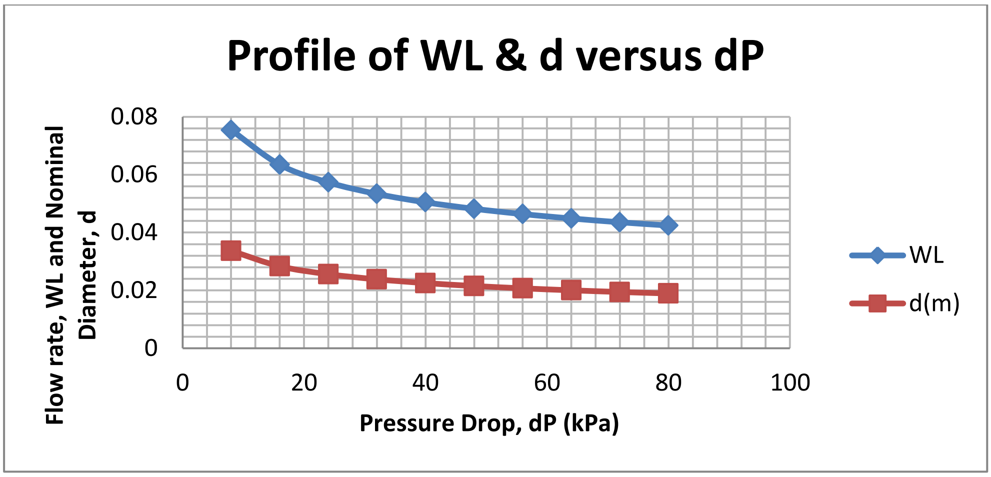

Figure 3 Plot of flow rates, WL & nominal diameter, d varies with differential pressure

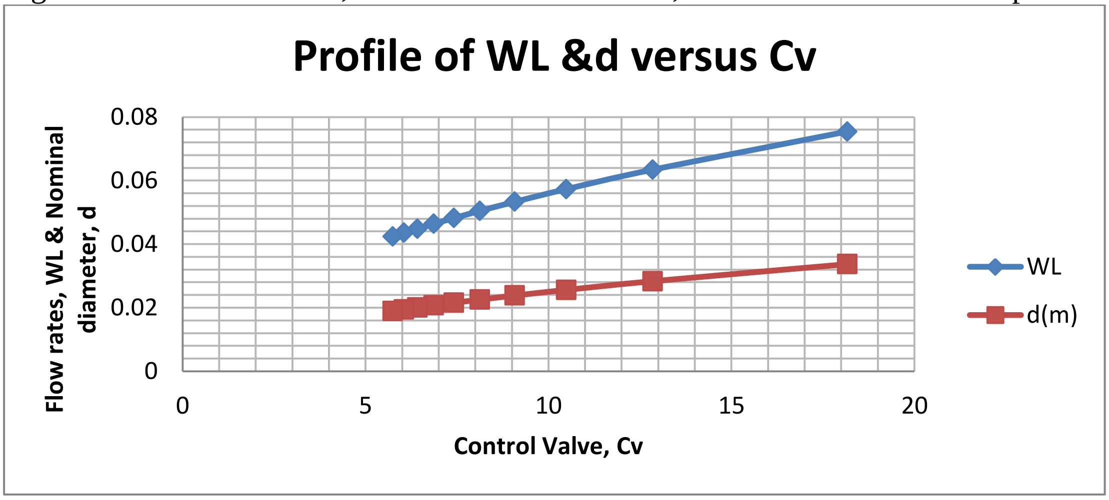

Figure 4 plot of flow rates & nominal diameter against control valve

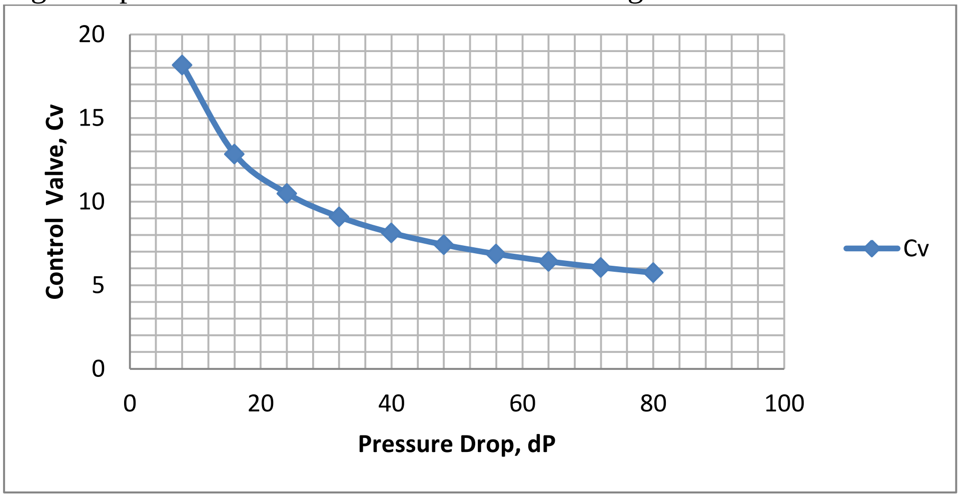

Figure 5 Plot of control valve varying with pressure drop

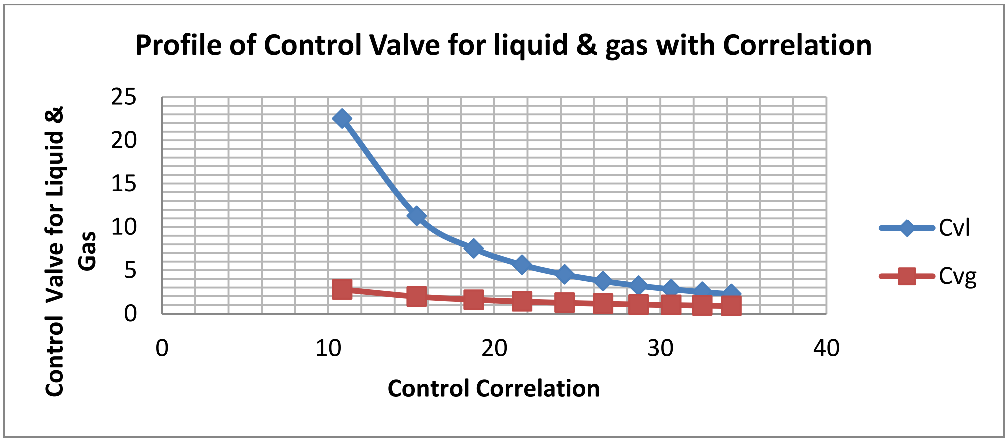

Figure 6 Plot of control valve for liquid & gas (C_{vl} & C_{vg}) against control correlation y.

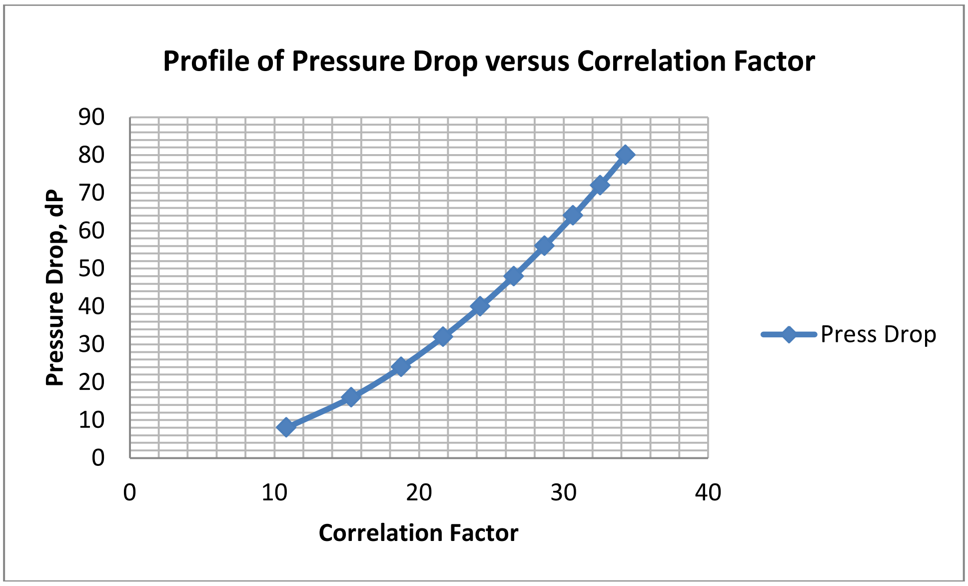

Figure 7 Plot of pressure drop, ΔP against correlation factor

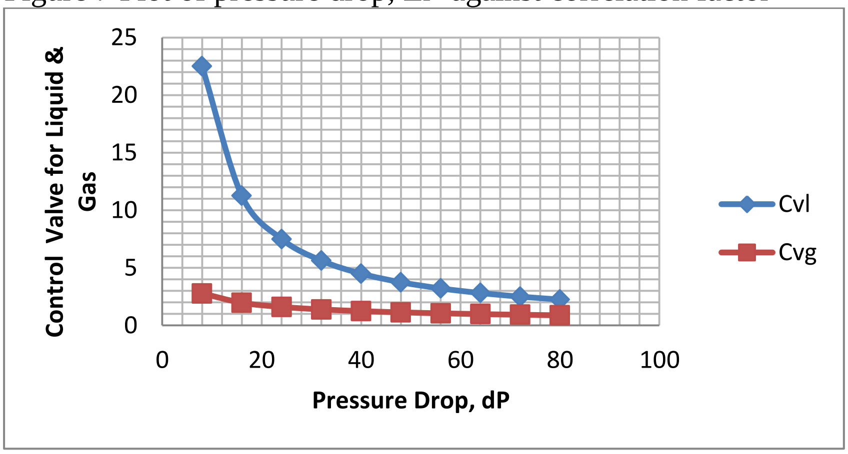

Figure 8 Plot of control valve for liquid & gas against pressure drop

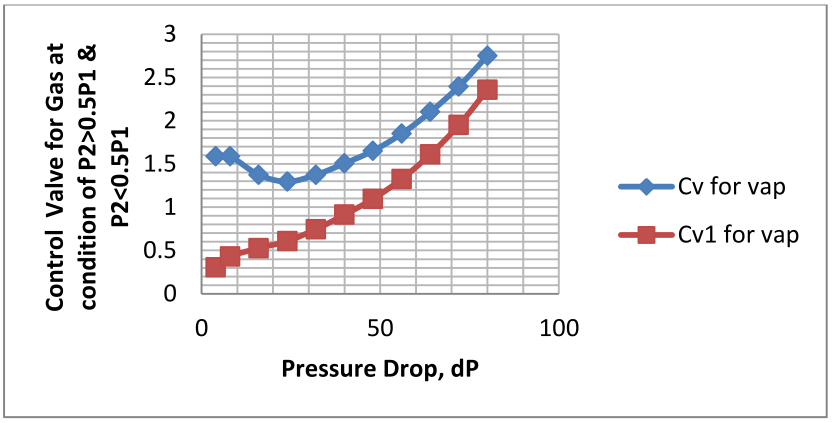

Figure 9 Plot of control valve at different pressure conditions i.e. $P_2 > 0.5P_1$ & $P_2 < 0.5P_1$ against pressure drop, ΔP

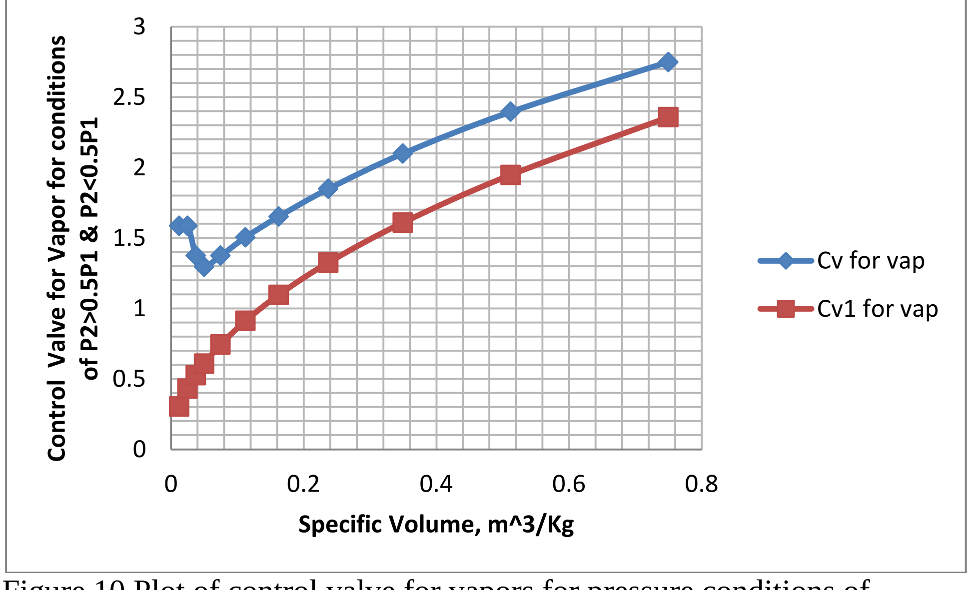

Figure 10 Plot of control valve for vapors for pressure conditions of $P_2 > 0.5P_1$ & $P_2 < 0.5P_1$ against specific volume

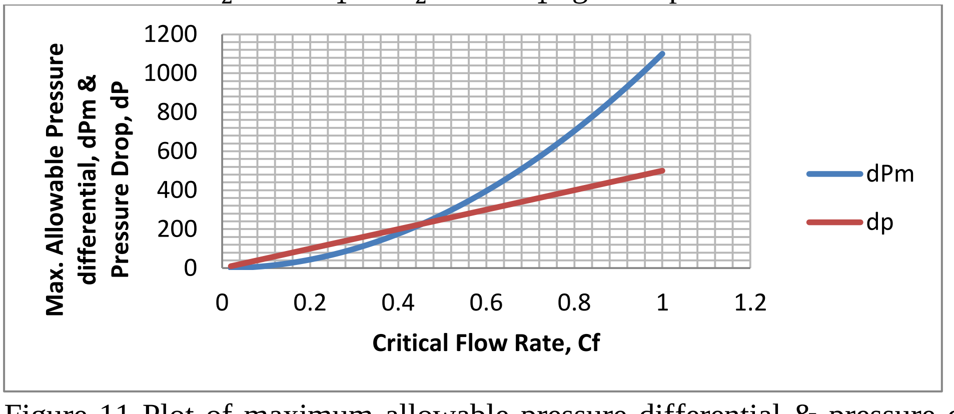

Figure 11 Plot of maximum allowable pressure differential & pressure drop against critical flow factor

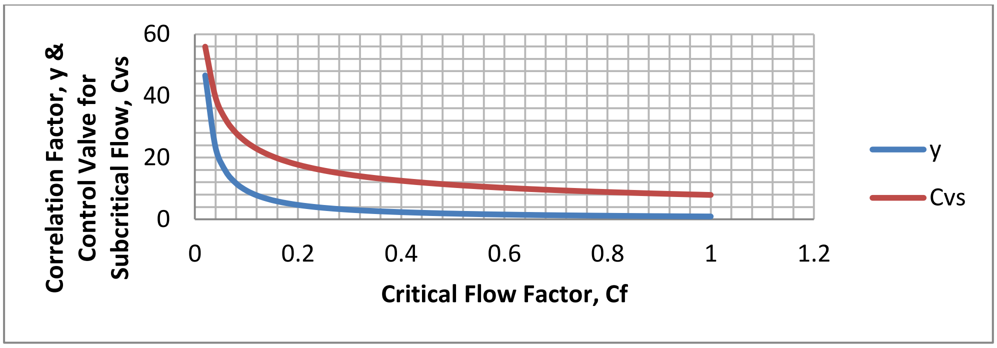

Figure 12 Plot of correlation factor, y & control valve for subcritical flow, $C_{v,s}$ against critical flow factor

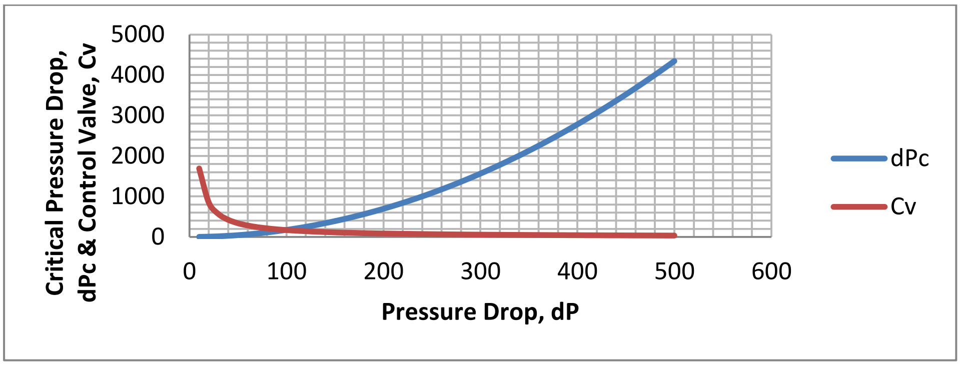

Figure 13 Plot of critical pressure drop, ΔP_C& control valve, C_v against pressure drop, ΔP

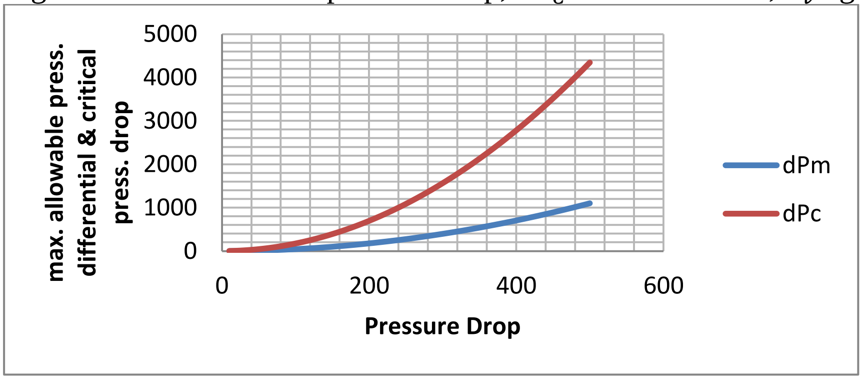

Figure 14 Plot of maximum allowable pressure differential, ΔP_m & critical pressure drop, ΔP_c against pressure drop, ΔP

DISCUSSIONS

Variation of critical flow rate against flow correlation

Figure 1 shows the flow correlation function from actual data for many valves configurations at maximum valve opening. This is seen as critical flow rate, and directly proportional to the correlation flow as shown in the figure 1. But at higher values of y, almost becomes exponential increased.

Variation of Correlation factor values against flow correlation

Figure 2 shows profile plot of correlation factor values (y-0.0148y^3) versus correlation flow, y. as shown in the figure 2, the profile is a projectile. That the correlation factor value is related to the flow correlation, y

Profile of flow rate versus nominal diameter

Figure 3 depicts the variation of flow rate in (kg/s) and Nominal diameter, d (m) Pressure Drop. Both curves decreases exponentially by a factor as the pressure Drop.

For instance, at a pressure drop of 10kPa,

$d \propto e^{dP}$ **[20]**

$W_L \propto e^{-dP}$ **[21]**

$d \simeq 0.035m$ & $W_l = 0.075kg/s$ and as pressure drop increases to dP = 80kPa the nominal diameter and flow rate decreases exponentially to respectively d = 0.015m and $W_l = 0.045m$

Profile of flow rate, nominal diameter against control valve

Figure 4 shows the profile of flow rate, W_L and Nominal Diameter, d versus control valve, C_v increases from 5 to 18, d = 0.015m to d = 0.025m and W_L increases from W_L =0.04kg/s to W_L = 0.08kg/s. Figure 4 further exhibits a throttling effect of valve performance.

i.e

$d \propto C_v$ **[22]**

$W_L \propto C_v$

$\Rightarrow d = KC_v$

$W_L = K_i C_v$

Where: $K_i \simeq 0.035kg/s$; K = 0.015m

Variation of Control valve, $C_v$versus Pressure Drop, dP

Figure 5 depicts control valve C_V decreases exponentially with pressure drop, $\boldsymbol{dP}$.

$C_V \propto e^{-dP}$ **[23]**

$C_{v=K_2 e^{-dP}}$ **[24]**

Where: K_2 = 18

From dP= 10kPa, to dP = 80kPa; $C_v$decreases exponentially from C_v = 18 to C_v =5

Plot of Control valve versus flow correlation

Figure 6 depicts the profiles of control valves for liquids and gases with control correlation y. The control valves characteristics for fluids (Liquids and gases) decreases exponentially as the control correlation factor increases i.e.

$C_{v_i} \propto e^{-y}$ **[25]**

$C_{v_i} = K_i e^{-y}$ **[26]**

Where: $i = l, \& g$.

$C_{v,l} = 22e^{-y}$ **[27]**

Where K_l=22 and $K_g = 3$.

From the plot, control for liquids is higher than that for gases as the correlation control factor varies.

Plot of Pressure drop, dP against correlation factor

Figure 7 shows an exponential increase of pressure drop from the threshold value of dP = 8kPa to maximum of dP = kPa with the correlation factor, y as it increases from y = 11 to y = 34.

i.e. $dp \propto e^y$

$dP \propto 8e^y$ **[28]**

Where K = 8 is the constant of proportionality.

Profile of Control Valves characteristics for fluid against Pressure drop

Similarly figure 6 and figure 8 indicates the variation of control valves characteristics for fluids with pressure drop. It shows vividly that the control valves for fluids decreases exponentially as pressure drop increases i.e.

$C_{V_i} \propto e^{-dP}$ **[29]**

Where i = l & g.

From the same figure, the exponential decrease for liquids is higher than that for gases. This shows that it is more convenient to control liquid than gas as pressure drop increases.

Variation of Control valves of vapor other than steam against Pressure drop

Figure 9 indicates profiles of control valves of vapor varying with pressure drop at different pressure conditions. As shown in the figure above, when outlet pressure P_2 is greater than half the initial pressure ($P_2 > 0.5P_1$), then control valve for vapor, other than steam suddenly drops from $C_v = 1.56\ to\ C_V = 1.3$ and then increases exponentially to $C_V = 2.7$ as pressure drop increases from OkPa to 25kPa and then to dP = 80kPa.

Similarly, the control value exponentially increases from $C_V = 0.4\ to\ C_V = 2.4$ along incremental pressure drop when the outlet pressure, P_2 is less than half the inlet pressure, $^1/_2\, P_1 (5P_1 < 0.5P_1)$.

Profile of control valves for fluids against specific volume

Conversely, figure 10 shows the variation of control valves for fluids with specific volume, v at different pressure conditions of $P_2 > 0.5P_1\ and\ P_2 < 0.5P_1$.

When the condition $P_2 > 0.5P_1$, curve 1 (C_v) shows a sudden drop in C_v value from C_V = 1.3 and then increases almost linearly from $C_v = 1.3$ to $C_v = 2.6$ as specific volume of the vapour, other than steam increases from 0 to 0.7 $\frac{m^3}{Kg}$.

When, $P_2 < 0.5P_1$ exponential increase of the control valve value from $C_V = 0.4\ to\ C_V = 2.4$ as the specific volume increases from V = 0m^3/kg to V = 0.7m^3/kg.

Variation of maximum allowable pressure differential & pressure drop against critical flow factor

Figure 11 shows plots of maximum allowable differential pressure dPm and pressure drop, dP against critical flow factor, C_f

The maximum allowable pressure differential have sharp exponential increase from dPm = 0kPa to dPm = 1050kPa and pressure drop have linear increment from dP = OkPa to dP = 500kPa as critical flow rate C_f increases from $C_f = 0\ to\ C_f = 1$

This shows that at higher value of C_f, the maximum allowable differential pressure becomes highest values as compared to the value of pressure drop which increases linearly.

i.e. $dPm \propto e^{c_f}$ **[30]**

$dP \propto C_f$ **[31]**

Another important point, you must note from graph is that at $C_f = 0.45$ dPm and dP is the same i.e.

At C_f = 0.45, $dPm = dP$ = 240kPa. This is the pressure where maximum allowable pressure differential is the same with the pressure drop.

Plot of correlation factor & control valve against critical flow

Figure 12 depicts the profiles of correlation factor, y and control valve, C_V against critical flow factor, C_f. Both profiles show exponential decrease as the critical flow factor above.

The subcritical control valve, C_{V_S} values are higher at low values of C_f and almost become steady at higher values of C_f. Similarly, for the correlation factor, the values for y at lower values of C_f and vice versa.

Variation of critical pressure drop & control valve versus pressure drop

Figure 13 depicts profiles of critical pressure drop, ΔP_c & Control valve, C_v against Pressure drop, ΔP. As the pressure drop increases, ΔP_c increase sharply and exponentially while C_v decreases slowly and exponentially till it reaches zero values and becomes steady. At pressure drop of 120kPa, $\Delta P_c = C_v$ = 200, is the point of intersection of the control valve and the critical pressure drop.

Plot of maximum allowable pressure drop & critical pressure drop versus pressure drop

Figure 14 shows the variation of maximum allowable pressure drop, ΔP_m and critical pressure drop, ΔP_c against pressure drop, dP. At incremental pressure drop, dP, the exponential increase of ΔP_m is far lower than that of ΔP_c i.e.
$\Delta P_c >>> \Delta P_m$, as $dP \rightarrow \infty$. This proves that the critical values of pressure drop are highest comparable to maximum allowable pressure differential.

CONCLUSION

This research thrives to investigate control valve selection for process steam condensate control of power plant and utilities of a refinery process plant. The parameters for sizing of the controller are cavitation, subcritical and critical flow situation plus other sub-routine necessary calculations.

The design model specific for the controllers algebraic in nature were evaluated using excel program. The various results were plotted in the profiles of figures 1 to 14. Fundamentally, fluid control institute formula for sizing and modeling of control valves and its characteristics as the validating literature.

Process controllers' selections for optimality study are presented. The application of mass balance and thermodynamic data were used for the sizing of control valves characteristics for control parameters such as cavitation, subcritical and critical flow of fluids (liquids and gases). A butterfly-valve type and single-port-top-guided valve with flow to open plug at $C_f = 0.90$ was analyzed and compared for the types to determine the maximum allowable pressure, pressure drop and critical pressure drop with flow correlation factor carried out and generalized profiles were achieved as shown in figures 1 - 14 above.

The research is most acceptable and adopted in the process control units of most oil and gas industries controllers revalidating process.

NOMENCLATURE

FCT:	Fluid control institute, Inc.
θ_R :	Set point
θ:	controlled variables
M_V	Measured valve
θ_M	Measured variable
ε	Error
C_V	Control valve
J	Output
PID	Proportional integrated derivative controller
F	Feedback response
J_O	Controller output
K_C	Proportional gain or sensitivity
T	Time of response
PI	Proportional integral action
K_d	Constant of PD-controller
K_i	Constant of integration representing lost
PD:	Proportional derivative controller.

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