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CALCULATION NO. QDC-2300-M-0082 PROJECT NO.: N/A	PAGE NO.: 1
SAFETY RELATED REGULATORY RELATED	NON-SAFETY RELATED
<u>CALCULATION TITLE:</u> Pressure Locking Calculation for HPCI System Valve MOV 1-2	301-8
STATION/UNIT: Quad Cities/1 Sys	TEM ABBREVIATION: HPCI
EQUIPMENT NO.: (F APPL.) MOV 1-2301-8	
REV: Ø STATUS: QA SERIAL NO. OR CHRON NO. N/A	DATE:
PREPARED BY: <u>K. Higgins</u> REVISION SUMMARY: Original Issue DO ANY ASSUMPTIONS IN THIS CALCULATION REQUIRE LATER VERIFICATION YES = NO SEE ASSLAND. 4	DATE: 11/9/9
REVIEWED BY: <u>J. Kelly</u> Jol 7 Reft	11/9/95
REVIEW METHOD: Detailed review	COMMENTS (C OR NC): <u>LC</u>
APPROVED BY: ACTOR WORATOR 11/1/95	
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COMMONWEALTH EDISON COMPANY CALCULATION REVISION PAGE

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COMMONWEALTH EDISON COMPANY

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N/A

I. PURPOSE/OBJECTIVE

Valve MOV 1-2301-8, which is installed in the HPCI System at Quad Cities, has been determined to be susceptible to the pressure locking phenomena. The purpose of this calculation is to determine that the valve will open when called upon by design during a pressure locking event by calculating the force necessary to overcome pressure locking and the available Motor/Gearing Capability (MGC) of the MOV.

II. METHODOLOGY AND ACCEPTANCE CRITERIA

The methodology for calculating the thrust required to open the MOVs under the pressure locking scenario is based on the Reference 1 (Roark's) engineering handbook. This methodology has been previously applied by ComEd in the References 2 and 10 calculations. The methodology determines the total force required to open the valve under a pressure locking scenario by solving for the four components to this required force. The four components of the force are the Pressure Locking Component, the static unseating component, the piston effect component, and the "reverse piston effect" component. These components are determined using the following steps.

Pressure Locking Component of Force Required to Open the Valve

The valve disk is modeled as two plates attached at the center by a hub which is concentric with the valve disk. A plane of symmetry is assumed between the valve disks. This plane of symmetry is considered fixed in the analysis.



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The pressure force is assumed to act uniformly upon the inner surface of the disk between the hub diameter and the outer disk diameter. The outer edge of the disk is assumed to be unimpeded and allowed to deflect away from the pressure force. In addition, the disk hub is allowed to stretch. The total displacement at the outer edge of the valve disk due to shear and bending and due to hub stretch are calculated using the reference 1 equations.



An evenly distributed force is assumed to act between the valve seat and the outer edge of the valve disk. This force acts to deflect the outer diameter of the valve disk inward and to compress the disk hub. The pressure force is reacted to by an increase in this contact force between the valve disk and seats. The valve body seats are conservatively assumed to be fixed. Therefore, the deflection due to the known pressure load must be balanced by the deflection due to the unknown seat load. The deflection due to the pressure force is first calculated. Then, the reference 1 equations are used to determine the contact force between the seat and disk which results in a deflection which is equal and opposite to the deflection due to the pressure force. Because the disk varies in thickness, the average thickness will be used for purposes of this calculation.

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The coefficient of friction between the seat and disk is determined based on the open valve factor from a DP test. The stem force required to overcome the contact load between the seat and disk which opposes the pressure force is equal to:

(seat load) x [(seat mu) cos(seat angle) - sin(seat angle)] x 2 (for two disk faces).

Static Unseating Force

The static unseating force represent the open packing load and pullout force due to wedging of the valve disk during closure. These loads are superimposed on the loads due to the pressure forces which occur during pressure locking. The value for this load is based on static test data for the MOVs.

Piston Effect

The piston effect due to valve internal pressure exceeding outside pressure is calculated using the standard industry equation. This force assists movement of the valve stem in the open direction.

$$F_{\text{pirmulant}} = \frac{.s^{r}}{4} \times D_{\text{rem}}^{2} \times (P_{\text{rem}} - P_{\text{rem}})$$

"Reverse Piston Effect"(F_{vert})

The reverse piston effect is the term used in this calculation to refer to the pressure force acting downward against the valve disk. This force is equal to the differential pressure across the valve disk times the area of the valve disk times the sine of the seat angle times 2 (for two disk faces).



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Total Force Required to Overcome Pressure Locking

As mentioned previously, the total stem force (tension) required to overcome pressure locking is the sum of the four components discussed above. All of the terms are positive with the exception of the piston effect component.

Next the motor gearing capability available to overcome static unseating forces is determined using the statically measured stem factor, the pullout efficiency, the temperature factor, and the ComEd motor test data (for breakdown torque and voltage factor), Reference 5.

Voltage, MR Breakdone × Temp Factor × OAR × Eff million × MGC 👝 = -Stem Factor

Determination of Open Valve Factor

The open valve factor (VF) for the purpose of this calculation is obtained from ComEd MOV White Paper MOV-WP-160, Rev. 0, 10/5/95. This value is consistent with calculated VFs that utilize differential pressure pull out and running thrusts.

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Acceptance Criteria

The margin between one time structural limit and required thrust is calculated. A margin of 15% or greater is required. This margin accounts for equipment inaccuracy and degradation.

The margin between MGC and required thrust is calculated. A margin of 20% or greater is required. This margin accounts for equipment inaccuracy and degradation.

If the enhanced capability evaluation is applied, a margin of 40% or greater is required. This margin accounts for voltage, current, and thrust measurement inaccuracies from the static test and a possible reduction in overall MOV efficiency from the static condition to the estimated pullout condition.

III. ASSUMPTIONS

- 1. The valve disk is assumed to act as two ideal disks connected by a hub. The equations in reference 1 are assumed to conservatively model the actual load due to pressure forces. This assumption is considered conservative since inspection of the disk drawings show large fillets between the disk hub and seats which should make the valve disk stiffer than assumed in the reference 1 equations.
- 2. The coefficient of friction between the valve seat and disk is assumed to be the same under pressure locking conditions as it is under DP conditions. This assumption (in combination with assumption 1) is considered to be justified based on bench marking of the calculation against ComEd and EPRI pressure locking test data for similar flex-wedge gate valves.
- 3. The bonnet pressure value is based on a scenario in which the MOV 1-2301-8 valve bonnet is pressurized to reactor feedwater pressure (1085 psig) by leakage past upstream check valve 1-2301-7. This value is conservative because it does not consider any feedwater piping line losses. The upstream and downstream pressure (both considered to be 0 psig) are based on a scenario in which a LOCA occurs, the feedwater header is depressurized and check valve 1-1-0220-58B does not allow any back leakage from the Reactor into the feedwater line. Again, the upstream and downstream assumed values are conservative as it is very unlikely that the feedwater line would decay to 0 psig before HPCI injection occurs. The bonnet pressure value is obtained from the Quad Cities UFSAR. See Reference 6.

4. A valve temperature of 250 °F is assumed for the purpose of this calculation. This temperature is based on conduction from feedwater at its normal operating temperature of 340 °F and MSIV room temperature of 150 °F. This assumed temperature will require verification. Feedwater and MSIV room temperatures are based on UFSAR Tbl 1.2-3 and Zone 10 during a LOCA per Bechtel Spec. 13524-069-N201, Rev. 1, respectively.

IV. DESIGN INPUTS

1. Valve Disk Geometry information is based on the Reference 4, Faxes from the Crane-Aloyco Valve Company. (Attachment A)

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CALC	CALCULATION NO. QDC-2300-M-0082				PROJECT N	10. N/A		PAGE NO. 9
	2. Motor Data is taken from the Reference 5 report and RSMDS, Reference 7.							
v .	V. REFERENCES							
	1.	Sixth Edit	ion of Roark's	Formulas for	Stress and St	rain		
	2.	MPR Calc Load", day Function c	ulations 101-0 ted 3/23/95; ar of Bonnet Press	13-1, "Effect of nd 101-013-4, sure", dated 3/	of Bonnet Pre "Estimate of 23/95	essure on 1 Valve Un	Disc (seatir	to Seat Contact 1g Force as
	3.	NMAC Re	port NP-6660-	D, " Applicat	ion Guide Fo	r Motor C)pera	ted Valves"
	4.	Crane-Alo 10/25/95,	yco Telecopy f Attachment A.	rom Bruce Ha	irry to Ken H	iggins (Be	chtel) dated
	5.	ComEd W 10/05/95.	hite Papers M(OV-WP-125, 1	Rev. 2, 10/4/	95 and M	OV-V	VP-160, Rev. 0,
	6.	UFSAR Ta	able 4.1-3.					
	7.	ComEd Ri	sing Stem MO	V Data Sheet,	1-2301-8, 04	/06/95, 2	2:34.	
	8.	Thrust values are taken from static VOTES test 10 performed 02/23/91.						
	9	EMS Calculation CE-DR-030, "Pressure Locking Analysis of Dresden Motor Operated Valves", dated 6/13/95.						
	10. ComEd Calculation NED-M-MSD-182, "Verification of Operability for Dresden and Quad Cities Injection Valves Susceptible to Pressure Locking", dated June 22, 1995.							
	11.	Thrust and	Torque Calcul	ation OTC-33	6, Rev. 2.			
VI.	CALC	CULATIONS	5					
	The following is provided for MOV 1-2301-8.							
	MathCad calculation of:							
 the pressure locking unseating force, the available motor gearing capability to unseat while pressure locked 								
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CALCULATION NO. QDC	C-2300-M-0082	PROJECT NO. N/A	PAGE NO.10
VI QCNPS Valve 1-	2301-8		·. ·
Bonnet Pressure Upstream Pressure Downstream Pressure	P _{bonnet} := 1085 psi P _{up} := 0 psi P _{doum} := 0 psi	Reference 6 Assumption 3 Assumption 3	
Disk Thickness, Avg Seat Radius Hub Radius Hub Length Seat Angle Poisson's Ratio (disk)	t := 2.085 in a := 5.625 in b := 1.625 in L := 2.0 in theta := 5 deg v := 0.3	Reference 4 Reference 4 Reference 4 Reference 4 Reference 7 Ref. 1 & 4, Carb. Steel	
Mod. of Elast. (disk) Static Pullout Force (Test 10)	$E := 27.6 \cdot 10^{6} \cdot \text{psi}$ F po := 52681 · lbf	Ref. 1 & 4, Carb. Steel Reference 8	
Stem Diameter	D _{stem} = 2.375 in	Reference 7	
Valve Factor:	VF := 0.55	Reference 5	

MU CALCULATION

Coefficient of friction between disk and seat: (Reference 3)

 $mu := VF \cdot \frac{\cos(\text{theta})}{1 - VF \cdot \sin(\text{theta})}$

PRESSURE FORCE CALCULATIONS

Average DP across disks:

$$DPavg := P_{bonnet} - \frac{P_{up} + P_{down}}{2}$$

mu = 0.575

 $DPavg = 1.085 \cdot 10^3 \cdot psi$

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CALCULATION NO. QDC-2300-M-0082	PROJECT NO.	N/A	PAGE NO. 11
Disk Stiffness Constants (Reference 1 Table 24)			
$D := \frac{E \cdot (t)^{3}}{12 \cdot (1 - v^{2})}$	D =	2.291•10 ⁷	•lbf in
$G := \frac{E}{2 \cdot (1 + v)}$	G =	1.062•10 ⁷	•psi
Geometry Factors: (Reference 1, Table 24)			
$C_{2} := \frac{1}{4} \cdot \left[1 - \left(\frac{b}{a}\right)^{2} \cdot \left(1 + 2 \cdot \ln\left(\frac{a}{b}\right)\right) \right]$	C 2	= 0.177	
$C_{3} := \frac{b}{4 \cdot a} \left[\left[\left(\frac{b}{a} \right)^{2} + 1 \right] \cdot \ln \left(\frac{a}{b} \right) + \left(\frac{b}{a} \right)^{2} - 1 \right]$	С ₃ -	= 0.031	
$C_{8} = \frac{1}{2} \left[1 + v + (1 - v) \left(\frac{b}{a} \right)^{2} \right]$	C ₈	=0.679	
$C_{9} := \frac{b}{a} \cdot \left[\frac{1+v}{2} \cdot \ln\left(\frac{a}{b}\right) + \frac{1-v}{4} \cdot \left[1 - \left(\frac{b}{a}\right)^{2} \right] \right]$	- c و C	= 0.28	
$\mathbf{L}_{3} := \frac{\mathbf{a}}{4 \cdot \mathbf{a}} \left[\left[\left(\frac{\mathbf{a}}{\mathbf{a}} \right)^{2} + 1 \right] \cdot \ln \left(\frac{\mathbf{a}}{\mathbf{a}} \right) + \left(\frac{\mathbf{a}}{\mathbf{a}} \right)^{2} - 1 \right]$	L ₃ =	= 0	
$L_{9} := \frac{a}{a} \left[\frac{1+v}{2} \cdot \ln\left(\frac{a}{a}\right) + \frac{1-v}{4} \cdot \left[1 - \left(\frac{a}{a}\right)^{2} \right] \right]$	L 9 =	•0	
$L_{11} := \frac{1}{64} \cdot \left[1 + 4 \cdot \left(\frac{b}{a}\right)^2 - 5 \cdot \left(\frac{b}{a}\right)^4 - 4 \cdot \left(\frac{b}{a}\right)^2 \cdot \left[2 + \left(\frac{b}{a}\right)^2 \right] \cdot 1$	$n\left(\frac{a}{b}\right)$ L ₁₁	=0.007	
$L_{17} = \frac{1}{4} \left[1 - \frac{1 - v}{4} \left[1 - \left(\frac{b}{a} \right)^4 \right] - \left(\frac{b}{a} \right)^2 \left[1 + (1 + v) \ln u \right] \right]$	$\left(\frac{\mathbf{a}}{\mathbf{b}}\right)$] L_{17}	=0.152	
Moment (Reference 1, Table 24, Case 2L)			
$M_{rb} = \frac{-DPavg \cdot a^{2}}{C_{8}} \cdot \left[\frac{C_{9}}{2 \cdot a \cdot b} \cdot (a^{2} - b^{2}) - L_{17} \right]$	M _{rb}	=-1.473•1	0 ⁴ ·lbf
$Q_{b} := \frac{DPavg}{2 \cdot b} \cdot (a^{2} - b^{2})$	Q _b =	= 9.682•10 ³	<u>.lbf</u> in
REVISION NO. 0			

CALCULATION NO. QDC-2300-M-0082PROJECT NO.N/APAGE NO. 12Deflection due to pressure and bending:(Reference 1, Table 24, Case 2L)
$$y_{bq} := M_{rb} \frac{a^{3}}{D} C_{2} + Q_{b} \frac{a^{3}}{D} C_{3} - \frac{DPavg.a^{4}}{D} L_{11}$$
 $y_{bq} = -0.002$ inDeflection due to pressure and shear stress:(Reference 1, Table 25, Case 2L) $K_{ss} := -0.3 \left[2 \cdot \ln \left(\frac{a}{b} \right) - 1 + \left(\frac{b}{a} \right)^{2} \right]$ $K_{ss} := -0.47$ $y_{sq} := \frac{K_{ss} DPavg.a^{2}}{r, G}$ $y_{sq} = -7.291 \cdot 10^{-4}$ inDeflection due to bub stretch (from center of hub to disk):P force $= -3.1416 \cdot (a^{2} - b^{2}) \cdot DPavg$ P force $= -3.1416 \cdot (a^{2} - b^{2}) \cdot DPavg$ y stretch $= \frac{1}{3.1416 \cdot b^{2}} \cdot \frac{L}{(2 \cdot E)}$ y stretch $= \frac{1}{2.4 \left(\frac{a}{a} \right) \cdot \ln \left(\frac{b}{a} \right) \cdot \frac{a}{(2 \cdot E)}$ y stretch $= y_{stretch} = -7.2737 \cdot 10^{-4} \cdot in$ Deflection due to pressure forces:y $_{stretch} = \frac{1}{2.4 \left(\frac{a}{a} \right) \cdot \ln \left(\frac{b}{a} \right) \cdot \frac{a}{c}} \right]$ y $_{stretch} = \frac{1}{2.4 \left(\frac{a}{a} \right) \cdot \ln \left(\frac{b}{a} \right) \cdot \frac{a}{c}} \right]$ Use $= -\frac{1}{12} \left(\frac{a^{3}}{a} + \ln \left(\frac{b}{a} \right) + \frac{a}{c} \right) - L_{2} \right] - \left[\left(\frac{a}{b} \right) C_{3} \right] + L_{3} \right]$ y $_{stretch} = \frac{a}{3.1416 \cdot b^{2}} \left(\frac{C \cdot 2}{c} \right) - L_{3} \right] - \left[\left(\frac{a}{b} \right) C_{3} \right] + L_{3} \right]$ y $_{stretch} = \frac{a}{3.1416 \cdot b^{2}} \left(\frac{C \cdot 2}{c} \right) - L_{3} \right] - \left[\left(\frac{a}{b} \right) C_{3} \right] + L_{3} \right]$ y $_{stretch} = \frac{a}{3.1416 \cdot b^{2}} \left(\frac{C \cdot 2}{c} \right) - L_{3} \right] - \left[\left(\frac{a}{b} \right) C_{3} \right] + L_{3} \right]$ Deflection due to ub compre

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CALCULATION NO. QDC-2300-M-0082	PROJECT NO. N/A	PAGE NO. 13
Seat Contact Force for which deflection is equal previous from pressure forces:	ously calculated deflection	in .
$\mathbf{F}_{\mathbf{S}} := 2 \cdot \pi \cdot \mathbf{a} \cdot \frac{\mathbf{y}_{\mathbf{q}}}{\mathbf{y}_{\mathbf{W}}}$	$F_{s} = 5.869 \cdot 10^{\circ}$	⁴ •lbf
UNSEATING FORCES (Re	eference 2)	
F _{pecking} is included in measured static pullout Force		
$F_{piston} := \frac{\pi}{4} \cdot D_{stem}^2 \cdot P_{bonnet}$	$F_{piston} = 4.807 \cdot 10^3 \cdot lbf$	
$F_{vert} := \pi \cdot a^2 \cdot sin(theta) \cdot (2 \cdot P_{bonnet} - P_{up} - P_{down})$	$F_{vert} = 1.88 \cdot 10^4 \cdot lbf$	
$F_{\text{preslock}} = 2 \cdot F_{s} \cdot (\text{mu} \cos(\text{theta}) - \sin(\text{theta}))$	$F_{\text{preslock}} = 5.706 \cdot 10^4 \cdot 10^4$	bf
$F_{total} = F_{piston} + F_{vert} + F_{preslock} + F_{po}$	$F_{po} = 5.268 \cdot 10^4 \cdot lbf$	
$F_{total} = 1.237 \cdot 10^5 \cdot 10^5$		

MOTOR / GEARING CAPABILITY INPUTS:

Motor Torque:	MR := 447-ft-lbf	Reference 5
Temperature Factor:	Tf := 1.0	Reference 5
Degraded Voltage:	DV = 177 volt	Reference 7
Under Voltage Factor:	n := 1.0	Reference 5
Stem Factor:	SF := 0.0322 ft	Reference 7
Overall Ratio:	OAR = 92.12	Reference 7
Pullout Efficiency:	Eff po = 0.4	Reference 7

MOTOR / GEARING CAPABILITY CALCULATIONS:



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ALCULATION NO.	QDC-2300-M-0082		PROJECT NO.	N/A	PAGE NO. J
For purposes of thi thrust value based room temperature °F will be approxim	is calculation, the Ope on an assumed temp of 150 °F). Actual ope nated by the following:	n Structural Limit . of 250 °F (based prating temp. will r	will be set to the O on conduction fro equire verification.7	pen Weak m FW at 34 The thrust v	Link 10 °F and alue at 250
OWL 100deg := 158	862·lbf		Referen	ce 11	
OWL 350deg := 1360	620·lbf		Referen	ce 7	
$\Delta Temp := 250 deg$		•			
ow	L 100deg - OWL 350de	9			
OWL 100deg		$-150 \cdot \text{deg} = 1.455 \cdot$	10 ³ ·Ibf Assump	tion 4	
OPEN STRUCTUR	ALLIMIT: Structu	ralLimit := 145500-	lbf		
OPEN LIMIT:	Limit	= (min((StructuralL	imit MGC)))		
	.				
MARGIN:	Margin :=	tal	Margin = 0.176		
	^F total				
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VII. COMPARISON OF MODEL OF OTHER SOURCES OF INFORMATION

Finite Element Calculation to Determine Seat Contact Force due to Bonnet Pressure

Results from the Reference 9 calculation demonstrate that the contact force between the seat and disk which is calculated using the MathCad model above is conservative and reasonably accurate (within 5%) for the 16" valve size modeled in this calculation. Once this Finite Element Analysis calculation is finalized, this calculation will be updated to provide the actual values.

Comparison to Actual Test Data

The reference 10 calculation demonstrates that the MathCad model calculates the pressure locking unseating thrust for the 6" flex-wedge Velan gate valve with good accuracy. In addition, pressure locking testing performed on a 10" Crane 900# Class flex-wedge gate valve at Quad Cities on July 21, 1995 indicates that the model accurately and conservatively predicts that pressure locking unseating force. A graph showing the initial results of this testing is provided below. Further testing is scheduled for later this year. The results of the entire test sequence will be formally documented in a ComEd report after all testing is completed.



COMMONWEALTH EDISON COMPANY

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VIII. SUMMARY AND CONCLUSIONS

This calculation has determined that the force required to unseat MOV 1-2301-8 (F_{total}) is 123,700 lbf and the Motor/Gearing Capability (MGC) is 196,800 lbf. The Open Structural Limit for MOV 1-2301-8, calculated at an assumed temperature of 250 °F, is the weak link value of 145,500 lbf. The margin was determined by finding the difference between the limiting open force, in this case the weak link value, and the unseating force, then dividing the resultant value by the unseating force to produced a margin of 17.6%.

The calculated margin of 17.6% is greater than the 15% minimum required margin, therefore, a pressure locking event will not prevent the valve from performing its design function.

IX. ATTACHMENTS

Telecopy from Bruce Harry (Crane-Aloyco Valves) to Ken Higgins (Bechtel) dated 10/25/95, Page A-1.

TO: KEN HIGGINS / X 3236

Bruce,

10/25/95

Could you please provide the seat radius, hub diameter, hub length, disk material and plate thickness for the following valves:

	1-1301-49 4" 783-U	1-2301-8 14" 783-U	
-Seat Radius	3.56¢	11.250	CONTRACT SEAT DIA.
Hub Diam. (D)	1.25¢	3.25¢	
Hub Length	0.875	2.00	
Disk Material	AZIG GK. WCB	AZIT GR. WCG	
Plate Thicknes (t)	0.844	2.085	DHUB E

756

Thanks for your assistance Bruce,

620

Ken Higgins, Quad Cities, Ext 3236

When FAXing (309) 654-2241X3026 you need to insert 5-6 pauses before the ext. number, or FAX to Brad Gebhardt @ QC.

1001-34 A(B) 16" 331/2-4 SOLID OF (FLEX) WEDGE?

nemo 7671 / * of pages > /
From B. HARRY
CO CRANE
Phone \$ 815-740-7570
Par # 815-727-4246

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