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**A CONTRIBUTION
TO THE ANALYSIS OF ENERGY LOSSES IN
TRANSIENT PIPE FLOW**

ALAN T.K. FOK Ph. D

1987



Alan T.K. Fok, Ottawa, Canada, 1987.

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UNIVERSITÉ D'OTTAWA
UNIVERSITY OF OTTAWA

A New Approach to the Analysis of Energy Losses in
Transient Pipe Flow

BY

Alan T.K. FOK

A thesis presented to the School of Graduate Studies
and Research in partial fulfillment of the requirement

for

the degree of Doctor of Philosophy

in the

Department of Civil Engineering
University of Ottawa

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ABSTRACT

The analytical approaches adopted in this thesis are a combination of: i) Dimensional reasoning adopted as a tool to establish the dimensionless parameters representative of transient characteristics, and the functional relationships between them; ii) Numerical model simulations carried out to quantify the relevance of these governing parameters; and iii) Physical modelling data used for identifying causes of energy losses which are unaccounted for numerically.

A new method of interpreting transient phenomena, the Energy Balance Concept, is developed in this research. The instantaneous elastic energy of the entire pipe system is determined by integrating the past history of transients.

The most important transient properties in transient pipe flows are investigated. These are: i) where and when rigid water body movement can be assumed; ii) how the friction affects maximum surge energy; and iii) determination of the optimal type of flow closure.

Some concepts currently adopted in unsteady friction theories of transient flows need to be reconsidered. Most important among these are: i) the use of Newton's second law of motion to analyse flow closure; ii) the use of friction dissipation to describe the decay of transients; and iii) the use of an unsteady friction factor normally considered to be the sole cause of unaccounted energy losses.

Conclusions from this research are: i) elastic properties actually dominate the pipe hydraulics in transient flow conditions; ii) non-frictional and frictional energy transfer are the prime mechanisms responsible for the decay of transients; and iii) energy dissipation not related to the friction factor may actually account for anomalous losses.

Friction models are selected and appropriately defined for their applications to various hydraulic conditions. Models recommended for general practice are: i) steady friction; ii) quasi-steady friction; and iii) time-dependent friction.

The present analysis covers a comprehensive range of practical hydraulic conditions. A set of hypothetical flow characteristics is thus developed and adopted for this purpose. These are used to generate realistic unsteady flow conditions associated with common surge control devices.

The analysis of surge control devices is conducted in parallel to the above research, providing information valuable for further scientific research and for engineering applications.

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TABLE OF CONTENTS

<u>Title</u>	<u>Page</u>
ABSTRACT	i
ACKNOWLEDGEMENTS	ii
TABLE OF CONTENTS	iii
LIST OF TABLES	viii
LIST OF FIGURES	ix
NOTATION	xiii
1.0 INTRODUCTION	1
1.1 General Problems of Hydraulic Transients	1
1.2 Sources of Surge Initiation	2
1.3 Protective Devices for Water Hammer	4
2.0 LITERATURE REVIEW	7
2.1 Types of Unsteady Flow	7
2.1.1 Non-transient (inelastic) flow	7
2.1.2 Transient (elastic) flow	8
2.1.3 Discussion of unsteady flow types	8
2.2 Unsteady Non-transient Flow	9
2.2.1 Non-transient U-tube flow	9
2.2.2 Non-transient pulsating flow	12
2.2.3 Discussion of non-transient flow	14
2.3 Unsteady Transient Flow	15
2.3.1 Research and practice in transient flow	15
2.3.2 Transient uni-directional flow	16
2.3.3 Periodic transient flow	22
2.4 Conclusions	25
3.0 RESEARCH OBJECTIVES AND SCOPE	32
3.1 Statement of Objectives	32
3.2 Scope of Research	33
3.2.1 Hydraulic system	33
3.2.2 Surge source and control	34
3.2.3 Means of analysis	34
3.2.4 Basic assumptions and constraints	34

4.0	METHODOLOGY	36
4.1	Dimensional Analysis as part of Analytical Approach	36
4.1.1	Governing parameters in dimensional analysis	37
4.1.2	Transient properties in dimensional analysis	43
4.1.3	Surge control devices in dimensional analysis	45
4.1.4	Discussion of dimensional analysis	49
4.2	Model Simulation as part of Analytical Method	51
4.2.1	Selection of modelling techniques	51
4.2.2	Development of the numerical model	53
4.2.3	Friction losses in transient flow simulation	58
4.2.4	Establishment of physical model data	61
4.2.5	Choice of interpreting method	63
4.2.6	Discussion of model simulations	64
4.3	Development of Hypothetical Flow Closure	65
4.3.1	General characteristics of flow closure	65
4.3.2	Classification of flow variation at source	67
4.3.3	Range of flow closing parameters	69
4.3.4	Discussion of hypothetical flows	70
4.4	Conclusions and Proposed Research	70
4.4.1	Analytical approach	70
4.4.2	Present research	72
4.4.3	Future research	73
5.0	ANALYSIS OF TRANSIENTS IN A NON-FRICTIONAL SYSTEM	77
5.1	Scope of Analysis	77
5.1.1	Cases illustrated for the mass balance	77
5.1.2	Conditions presented by pressure response	78
5.2	Energy Relations in Transient Flow	78
5.2.1	Internal energy in a pipe	78
5.2.2	Steady-state conditions in a pipe	79
5.2.3	Instantaneous flow closure in a pipe	79
5.2.4	Normal flow closure in a pipe	86
5.2.5	Discussion of Energy Balance Concept	87
5.3	Periodic Fluctuation in a Normal Flow Closure	87
5.3.1	General trends of periodicity	87
5.3.2	Uniform deceleration in a flow closure	89
5.3.3	Decreasing deceleration in a flow closure	94
5.3.4	Increasing deceleration in a flow closure	96
5.3.5	Discussion of periodic fluctuation	102
5.4	Analysis of Maximum Transient Energy	103
5.4.1	Maximum surge energy	103
5.4.2	Maximum counter-surge energy	105
5.4.3	Discussion of maximum transient energy	107
5.5	Conclusions	107

6.0	ANALYSIS OF TRANSIENTS IN STEADY FRICTION SYSTEM	119
6.1	Energy Relations in Transient Flow	119
6.1.1	Scope of analysis	119
6.1.2	Steady state condition in a pipe	120
6.1.3	Friction dissipation in a transient flow	122
6.1.4	Concept of energy balance in a pipe flow	123
6.1.5	Discussion of energy relation	125
6.2	Friction Transfer in Transient Flow	125
6.2.1	Friction transfer during a surge period	125
6.2.2	Friction transfer during a decaying period	133
6.2.3	Discussion of friction transfer	137
6.3	Parameters Governing Friction Transfer	138
6.3.1	Relevant parameters to surge energy	139
6.3.2	Relevant parameters to decaying energy	142
6.3.3	Discussion of governing parameters	145
6.4	Maximum Transient Energy	145
6.4.1	Scope of analysis	146
6.4.2	Maximum surge energy	147
6.4.3	Maximum counter-surge energy	149
6.4.4	Peak decaying energy	151
6.4.5	Discussion of maximum transient energy	153
6.5	Conclusions	153
7.0	ANALYSIS OF TRANSIENTS IN QUASI FRICTION SYSTEM	172
7.1	Scope of Analysis	172
7.1.1	Cases of transient history used for analysis	172
7.1.2	Flow regimes adopted for analysis	172
7.2	Energy Relations in a Quasi-steady Friction System	174
7.2.1	Friction dissipation in transients	174
7.2.2	Concept of energy balance in transient flow	176
7.2.3	Discussion of energy relation	176
7.3	Quasi (Steady) Friction Transfer in Transients	177
7.3.1	General Characteristics	177
7.3.2	Quasi friction transfer in a surge period	178
7.3.3	Quasi friction transfer in a decaying period	183
7.3.4	Summary of quasi friction transfer	187
7.3.5	Discussion of quasi friction transfer	187
7.4	Parameters Governing Quasi Friction Transfer	188
7.4.1	Quasi friction transfer to surge energy	188
7.4.2	Quasi friction transfer to decaying energy	191
7.4.3	Discussion of governing parameters	192

7.5	Maximum Transient Energy	193
7.5.1	Evaluation criteria for maximum transients	193
7.5.2	Maximum surge energy	194
7.5.3	Peak decaying energy	197
7.5.4	Discussion of quasi friction energy decay	199
7.6	Surge Control Devices in Quasi Friction System	200
7.6.1	Flow regimes in surge control systems	200
7.6.2	Transient behavior in surge control system	201
7.6.3	Discussion of control devices in quasi system	202
7.7	Conclusions	202
8.0	ANALYSIS OF TRANSIENTS IN UNSTEADY FRICTION SYSTEM	220
8.1	Current Unsteady Friction Theories	220
8.1.1	Reference from mass oscillations	220
8.1.2	Scope of analysis	221
8.1.3	Unsteady friction in uni-directional flow	223
8.1.4	Unsteady friction in periodic transient flow	232
8.1.5	Theories of unsteady friction	241
8.1.6	Discussion of current unsteady friction	245
8.2	Energy Relation in an Unsteady Friction System	246
8.2.1	Selected unsteady friction theory	246
8.2.2	Energy in flow with periodic fluctuations	248
8.2.3	Transients in an unsteady friction system	253
8.3	Analysis of Time-dependent Friction Model	255
8.3.1	Re-simulation with time-dependent model	256
8.3.2	Parameters governing unsteady friction	259
8.3.3	Theory from oscillating flow	264
8.3.4	Unsteady friction in uni-directional flow	265
8.3.5	Discussion of time-dependent friction model	267
8.4	Maximum Transient Energy in Unsteady Friction Pipe	268
8.4.1	Maximum surge and counter-surge energy	268
8.4.2	Peak decaying energy	268
8.4.3	Discussion of maximum transient energy	271
8.5	Friction Model for Surge Control Devices	271
8.5.1	Current application of unsteady (f_u) model	271
8.5.2	Selected unsteady friction model in practices	274
8.6	Conclusions	276
9.0	PRINCIPAL CONCLUSIONS AND RECOMMENDATIONS	294
9.1	Principal Conclusions	294
9.1.1	General analytical approach	294
9.1.2	Transient phenomena	295
9.1.3	Development of unsteady friction theory	296

9.2	Principal Recommendations	299
9.2.1	Recommendations for present research	299
9.2.2	Recommendations for future research	300
A.0	ANALYSES RELATED TO THE PRESENT RESEARCH	304
A.1	Practical Range of Friction Head	304
A.2	Selection of Maximum Transient Energy	306
A.3	Boundary of Alternate Fluctuating Flow	309
A.4	Optimal Energy in Uni-directional Flow	311
A.5	Classification of Friction	315
A.6	Investigation of Oscillating Flows	321
B.0	ANALYSIS OF RIGIDITY IN PIPE FLOW	339
B.1	Introduction	339
B.2	Development of Rigid Water Column Theory	239
B.3	Constraints Associated with Classical RWCT	344
B.4	Analysis of Rigid Flow Motion in Typical Flows	346
B.5	RWCT Classification in Hypothetical Flows	351
B.6	Regions of New RWCT in Maximum Surge Energy	357
B.7	Practical Applications	362
B.8	Storm Sewer Flow Simulations	366
B.9	Conclusions	369
C.0	HIGHLIGHTS OF ANALYSIS IN SURGE CONTROL DEVICES	386
C.1	Transient Analysis of Valve Closure in Pipeline	387
C.2	Transient Analysis of Pump Trip-out in Pipeline	388
C.3	Transient Analysis of Air Chamber in Pipeline	389
C.4	Transient Analysis of Surge Tank in Pipeline	390
D.0	SOME CONTRIBUTIONS FROM THE THESIS	392
	REFERENCES	394

LIST OF TABLES

<u>NUMBER</u>	<u>TITLE</u>	<u>AFTER PAGE</u>
2.1	Summary of Literature Review	30
2.2	Conflicts of Friction theories	31
4.1	Modeling as a Tool for Transient Analysis	74
5.1	Summary of Non-frictional Analysis	111
6.1	Summary of Steady Friction Analysis	158
6.2	Summary of Friction Dissipation	159
6.3	Summary of Friction Transfer	159
7.1	Summary of Quasi-steady Friction Analysis	206
7.2	Summary of Quasi Friction Dissipation	207
7.3	Summary of Quasi Friction Transfer	207
8.1	Summary of Unsteady Friction Transfer	280
8.2	Unsteady Friction Model in Control Device	280

LIST OF FIGURES

<u>NUMBER</u>	<u>TITLE</u>	<u>PAGE</u>
Chapter 1		
1.1	Schematics of a Pipe System with Various Protective Devices	6
Chapter 4		
4.1	Characteristics of Flow Closure	
4.2	Analytical Approach and Present Research	76
Chapter 5		
5.1	Transient History of Non-frictional, Instant Flow Closure	112
5.2	Transient History of Non-frictional, Uniform Deceleration Flow	113
5.3	Transient History of Non-frictional, Decreasing Deceleration Flow	114
5.4	Transient History of Non-frictional, Increasing Deceleration Flow	115
5.5	Flow History of Periodic Surge Fluctuations	116
5.6	Maximum Surge Energy of Non-frictional Flow System	117
5.7	Maximum Counter-surge Energy of Non-frictional Flow System	118
Chapter 6		
6.1	Transient History of Frictional, Instant Flow Closure	160
6.2	Transient History of Frictional, Uniform Deceleration Flow	161
6.3	Transient History of Frictional, Decreasing Deceleration Flow	162
6.4	Transient History of Frictional, Increasing Deceleration Flow	163
6.5	Frictional Transfer During Flow Closing Period	164

6.6	Frictional Transfer During Decaying Period	165
6.7	Maximum Surge Energy induced by Typical Flow Closures	166
6.8	Conditions for same level of Maximum Surge Energy	167
6.9	Maximum Counter-surge Energy induced by Typical Flow Closures	168
6.10	Hydraulic Conditions required for same Energy Reduction during Counter-surge Period	169
6.11	Peak Decaying Energy induced by Typical Flow Closures	170
6.12	Hydraulic Conditions for same Energy Reduction during Decaying Period	171

Chapter 7

7.1	Transient History of Instant Flow Closure with Quasi-steady Friction Factor	208
7.2	Transient History of Uniform Deceleration Flow with Quasi-steady Friction Factor	209
7.3	Transient History of Decreasing Deceleration Flow with Quasi-steady Friction Factor	210
7.4	Transient History of Increasing Deceleration Flow with Quasi-steady Friction Factor	211
7.5	Quasi-steady Friction Transfer During Flow Closing Period	212
7.6	Quasi-steady Friction Transfer During Decaying period	213
7.7a	Maximum Surge Energy induced by Uniform, Decreasing and Increasing Deceleration Flow with Quasi-steady Factor	214-216
7.8	Conditions required for 10 % Maximum Surge Energy Reduction with Quasi-steady Factor	217
7.9	Peak Decaying Energy for Various Typical Flow Closures with Quasi-steady Factor	218
7.10	Hydraulics Required for 50 % Peak Decaying Energy Reduction with Quasi-steady Factor	219

Chapter 8

8.1	Linear Momentum Friction Theory (an example from Dally, 1956)	281
8.2	Non-linear Momentum Friction Theory (an example from Popov, 1974)	282
8.3	Application of Unsteady Friction Model in Surge Tank (an example from Sarpkaya, 1959)	283
8.4	Viscous Friction Theory (an example from Sauer, 1969)	284
8.5	Some Weighting Functions in Time-dependent Friction Factor Application	285
8.6	Effects of Time-dependent Friction Factor in Holmboe's Works (1964)	286
8.7	Evaluation of Alternative Weighting Function (an example from Hirose, 1971)	287
8.8	Effects of Time-dependent Friction Factor on Transients during the Time of Decay	288
8.9	Effects of Time-dependent Friction Factor on Transients in Uni-directional Flows	289
8.10	Peak Decaying Energy for Various Typical Flow Closures with Time-dependent Friction Factor	290
8.11	Hydraulics Required for 50 % Peak Decaying Energy Reduction with Time-dependent Friction Factor	291
8.12	Application of Unsteady Friction Factor in Valve Closure (an example from Apelt, 1983)	292
8.13	Application of Unsteady Friction Models in Pumping System (an example from Apelt, 1983)	293

Appendix A

A.1	Friction Head in a Smooth Pipe	331
A.2	Comparison between Time and Enveloped Maximum Surge Energy	332
A.3	Hydraulic Conditions required for Optimal Maximum Surge Energy	333
A.4	Conditions required for Optimal Valve Closure	334
A.5	Friction Classification on Maximum Surge Energy	335

A.6	Friction Classification on Maximum Surge Energy (Valve)	336
A.7	Friction Classification on Maximum Surge Energy (Pump, Chamber and Tank)	337
A.8	Oscillation in U-tube (an example from Streeter & Wylie, 1981)	338

Appendix B

B.1	Region of RWCT (based on $T_r' < T < T_a$)	372
B.2	Region of RWCT (based on $T_r'' = T = T_a$)	373
B.3	Evaluation of RWCT for a Uniform Deceleration Flow in Frictional Systems	374
B.4	Region of RWCT based on maximum pressure envelope ($T_r = T = T_a$)	375
B.5	Region of RWCT in A Frictionless Uniform Valve Closure	376
B.6	Region of RWCT for a Valve System	377
B.7	Region of RWCT for Pump and Surge Tank Systems	378
B.8	Region of RWCT for an Air Chamber System	379
(5.2)	Transient History of Non-frictional, Uniform Deceleration Flow	380
(5.3)	Transient History of Non-frictional, Decreasing Deceleration Flow	381
(5.4)	Transient History of Non-frictional, Increasing Deceleration Flow	382
(5.5)	Flow History of Periodic Surge Fluctuations	383
(5.6)	Maximum Surge Energy of Non-frictional Flow System	384
(6.7)	Maximum Surge Energy induced by Typical Flow Closures	385

Appendix C

(1.1)	Pipe System with Various Protective Devices	391
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NOTATION

A) DIMENSIONAL

<u>Symbol</u>	<u>Description</u>
a	Surge wave speed
d	Pipe diameter
e	Internal system energy
e_m	Elastic energy of pipe system
e_w	Flow energy outflux
g	Gravitational acceleration
h	Pressure head in line
h_c	Joukowsky Head, $h_c = av/g$
h_{av}	inertial head, $h_{av} = l/g * dv/dt$
h_s	Friction head at source
h_{mx1}	Maximum surge pressure at source
h_s	Static head at source
h_v	Kinetic head, $h_v = v^2/2g$
k	Pipe roughness
l	Pipe length
m	Transient property
q	Flow rate
q_0	Steady flow rate
t	Time following the surge initiation
t_c	Duration for a surge wave travelling from the source to the pipe-end, $t_c = l/a$
t_d	Duration of unsteady flow closure
v	Flow velocity
ρ	Density of fluid
γ	Specific weight of fluid
ν	Kinetic viscosity of fluid
τ	shear stress
w	Oscillating frequency of flow, $w = 4 l/v$

B) DIMENSIONLESS

<u>Symbol</u>	<u>Description</u>
E	Internal energy of pipe system
E_m	Transient (Elastic) energy of pipe system
E_s	(Steady) friction energy $E_s = 0.5 H_s$
E_{mx}	Maximum transient energy
E_{mx1}	Maximum surge energy
E_{mx2}	Maximum counter-surge energy
E_d	Flow closure energy, $E_d = E_{we}(T_{x1}) - E_{ws}(T_d)$
E_{re}	Flow energy reversal, $E_{re} = E_{we}(T_{x1}) - E_{we}(T_{mx2})$
E_T	Peak decaying energy at time T
E_{101}	Peak decaying energy at time $T = 101$
E_w	Flow energy netflux, $E_w = E_{we} - E_{ws}$
E_{we}	Flow energy outflux at the pipe end

E_{ws}	Flow energy influx at the source of pipe
ED_z	Friction dissipation, in a form of heat loss
$ED_{\neq}$	Non-friction dissipation, in a form of heat loss which has the same magnitude of ET_{uqd}
ER_{mx2}	Steady counter-surge reduction, $ER_{mx2} = 1 - E_{mx2}/E_{mx1}$
ER_{101}	Steady peak decaying reduction, $ER_{101} = 1 - E_{101}/E_{mx1}$
ER_{101}^*	Altern peak decaying reduction, $ER_{101}^* = 1 - E_{101}/E_{mx2}$
ER_{qzs}	Quasi surge reduction, $ER_{qzs} = ET_{qzs}/E_{mx1}(f)$
ER_{qzd}	Quasi decay reduction, $ER_{qzd} = (E_T(f) - E_T(f_q))/E_T(f)$
ER_{uqd}	Unsteady decay reduction, $ER_{uqd} = 1 - E_{101}(f_u)/E_{101}(f_q)$
ET_z	Friction transfer in a form of mechanical process
ET_{zs}	Friction surge transfer, $ET_{zs} = E_w(H_z) - E_w(0)$, $T < T_{mx1}$
ET_{zd}	Friction decay transfer, $ET_{zd} = E_q(T_{z1}) - E_{zx}/2$, $T > T_{mx1}$
ET_{qc}	Quasi friction transfer, $ET_{qc} = ET_z(f) - ET_z(f_q)$
ET_{qzs}	Quasi surge transfer, $ET_{qzs} = ET_{zs}(f_q) - ET_{zs}(f)$
ET_{qzd}	Quasi decay transfer, $ET_{qzd} = ET_{zd}(f_q) - ET_{zd}(f)$
ET_{uq}	Unsteady friction transfer, $E_{uq} = ET_z(f_q) - ET_z(f_u)$
ET_{uqs}	Unsteady surge transfer, $ET_{uqs} = ET_{zs}(f_u) - ET_{zs}(f)$
ET_{uqd}	Unsteady decay transfer, $ET_{uqd} = ET_{zd}(f_u) - ET_{zd}(f)$
$ET_{\neq}$	Non-friction transfer, $E_{wn}(T_{zs} < T < T_{z1}) - E_{ws}(T_{zs} < T < T_q)$
f	Darcy-Weisbach friction factor
f_q	Quasi-steady friction factor
f_u	Unsteady friction factor
H_m	Transient pressure, $H_m = h_m/h_c$
H_z	Friction head, $H_z = h_z/h_m$
H_o	Pressure head at source (e.g. valve, pump and tank)
H_o^*	Pressure head at air chamber (at absolute datum)
H_{or}	Head loss for steady flow passing through orifice
H_{mx}	Maximum transient pressure at the source
H_{mx1}	Maximum surge pressure at source
H_s	Static pressure head at source
H_v	Kinetic head, $H_v = h_v/h_m$
K_{or}	Ratio of differential orifice
L	Pipe length
N	Positive integer
M	Transient property
Q	Flow rate
Q_x	Flow rate at the pipe-end
Q_o	Steady flow rate
Q_s	Flow rate at the source
R	Flow Reynolds Number
S	Shape factor of flow closure
S^*	Optimal shape factor of maximum surge energy
S_x	Upper bound of additive periodic fluctuation, $S_x = 2^+$
S_y	Lower bound of additive fluctuation, $S_y = 1.7$
T	Time of transient, following surge initiation
T_1	Specific time instant during the transient period.
T_{qz}	Time of quasi flow interception, when $Q_x(f) = Q_x(f_q)$
T_c	Duration for a surge wave travelling along line
T_{z1}	Time of pipe-end flow closure, when $Q_x(T_{z1}) = 0$
T_{zs}	Time of flow interception, when $Q_x(T_{zs}) = Q_s(T_{zs})$
T_m	Mechanical travel time of devices
T_{mx}	Time at maximum transient energy to occur
T_{mx1}	Time at maximum surge energy to occur

T_{mx2}	Time at maximum counter-surge energy to occur
T_q	Time of flow closure, $T_q = t_q/t_c$
T_s	Period of surge fluctuation, $T_s = 4$
W	Weighting function in Zielke's equation
X	Location in pipe system
Z	Time variable in Zielke's equation

C) ABBREVIATION

<u>Symbol</u>	<u>Description</u>
A * B	A multiply B
O()	order of magnitude
F(X,Y,...)	Function of parameters X,Y,...
Ch. X	Chapter X
Sec. X.Y	Section X.Y
Sub. X.Y.Z	Subsection X.Y.Z
H.G.L.	Hydraulic Grade Line

D) TERMINOLOGY

<u>Term</u>	<u>Source</u>
Flow opening	Sec. 1.2
Flow closing	Sec. 1.2
Transient flow	Sec. 2.1
Non-transient flow	Sec. 2.1
Mass Oscillation	Sub. 2.1.1
Pipe flow System	Sub. 2.1.2
Uni-directional flow	Sub. 2.1.2
Bi-directional flow	Sub. 2.1.2
Instantaneous flow closure	Sub. 5.2.3
Normal flow closure	Sub. 5.2.4
Energy Balance Concept	Sub. 5.2.5
Periodic flow fluctuation	Sec. 5.3
Flow separation	Sub. 8.2.2
Boundary layer loss	Sub. 8.2.2



1.0 INTRODUCTION

1.1 General Problems of Hydraulic Transients

The treatment of unsteady flows in pipes, commonly termed waterhammer or hydraulic transients, is one of the most important and difficult subjects of pipeline analysis and design. In municipal systems, for example, uncontrolled pressure surges can cause air entrainment or back-siphoning of contaminated ground water into a water supply system. Other problems include vapor or air cavities in the hydraulic system which can be a nuisance or can cause significant damage, and pipe failure by collapse due to low or high internal pressure, or to periodic shocks induced by surge.

Historically, the study of hydraulic transients is generally considered to have begun with the works of Joukowsky (1898) and Allievi (1902). Since then, a great deal of effort has been devoted to this subject, and as a consequence there are a number of approaches available which allow engineers to design large and complex closed conduit systems with various configurations and types of protective devices. These approaches include the experience and expertise gained from past theoretical developments, experimental tests and design practice. The recent application of high-speed computer numerical simulation has certainly provided a powerful tool for the analysis of waterhammer. However, at the same time there are still a number of fundamental aspects of hydraulic transient analysis which are not yet well understood or established. Further work is needed in several areas, including:

- i) Energy losses related to friction and non-friction (e.g. change of flow momentum);
- ii) local losses associated with the pipe configuration (e.g. joints and bends);

- iii) the impacts of cavity existence and collapse;
- iv) the hydraulic responses in a branching or looping network system; and
- v) the effect from non-elastic properties (e.g. soil backfilling and pipe fixation).

The present analysis attempts primarily to clarify one aspect of these research areas which has been considered to be of basic importance in hydraulic transient analysis. This is:

"Energy losses in transient pipe flows, case (1)."

The nature of this transient phenomenon is outlined in detail in Chapter 3 following a review of the relevant basic principles and literature.

1.2 Sources of Surge Initiation

When a change is made to flow at some point in a pipe system, transients arise. As a result of this change, surge waves travel through the piping system away from the point of initiation. Following the cessation of the initiating change, fluctuating pressure surges continue until completely damped.

Such changes can appear in many ways, but for convenience can be grouped into the following four conditions:

- i) Upstream flow closing --- which initiates a decrease of flow and a negative pressure surge at the upstream end of the system;
- ii) Upstream flow opening --- which causes a positive pressure surge upon the increase of flow;

iii) Downstream flow closing --- which causes a positive pressure surge at the downstream end of the system; and

iv) Downstream flow opening --- which results in an initial negative pressure surge.

If, for example, an upstream pump is suddenly shut off, case (i) occurs. After a pump trip-out, the only energy which remains to promote flow into the discharge line is the kinetic energy of the rotating motor and pump, and of the water contained in the pump. This energy is usually small compared to that required to maintain flow in the discharge line, and a rapid reduction in pump speed follows. This in turn reduces flow in the discharge line, and a water hammer wave of low pressure is formed. The subnormal pressure wave moves rapidly away from the pump until it reaches a terminal point such as a reservoir or a manhole, at which time a wave reflection occurs. The wave reflection returns in the direction of the pump, rebounds again, and continues to oscillate until friction losses in the system nullify the energy potential of the pressure wave. As this is happening, flow continues to decrease at the pump location. If no backflow condition is available to check flows, there is a tendency for flows to pass backward through the pump, which may then act as a turbine as long as the terminal reservoir provides adequate water to the pump.

Hydro power turbines (located at the downstream end of a conduit, case (iii)), act in manner generally similar to a pump placed in the upstream end of the system except that the sign of the pressure surge wave is opposite.

This inherent similarity in different conditions of surge initiation is used in the present research to reduce the amount of analytical effort and minimize redundant analyses associated with the above four conditions of surge initiation. Although this thesis focuses on situations related to the upstream flow closing (case (i)) in a pipe system (illustrated in Fig. 1.1), results of the present

analysis can also be used for the other three cases of flow closing or opening by adjustment of the sign and/or the magnitude of the transient flow and pressure along the system.

1.3 Protective Devices for Water Hammer

Water hammer induced by increases of pipe flow can be eliminated or minimized by relatively simple surge protection methods (Fok, 1985). For instance, over-pressure or air venting problems can usually be removed by a slow and progressive valve opening, or by sequential pump start-up. Normal operating sequences of system performance are designed to reflect this surge control measure.

In contrast, uncontrolled emergency conditions (e.g. shaft failure in a pump or turbine) require a well-conducted waterhammer analysis, and a reliable protective system in place before the event.

The most commonly employed measures to control water hammer can be grouped into the following four categories, sketched in Figure 1.1:

- i) Valve operation --- this is implemented by the installation of various types of surge relief valves. Generally this measure controls the surge conditions by pre-set valve closing characteristics. Subsequently, the rise of counter-surge pressure is also restricted within a tolerable limit.

- ii) Kinetic energy in the pumping system --- this may be achieved by employing oversized pump motors and flywheels to increase kinetic energy stored in rotating parts, and thereby buffer a rapid flow closure.

- iii) Surge tank --- this measure controls transients by converting stored potential energy in an elevated water body into kinetic energy, which forces supplementary

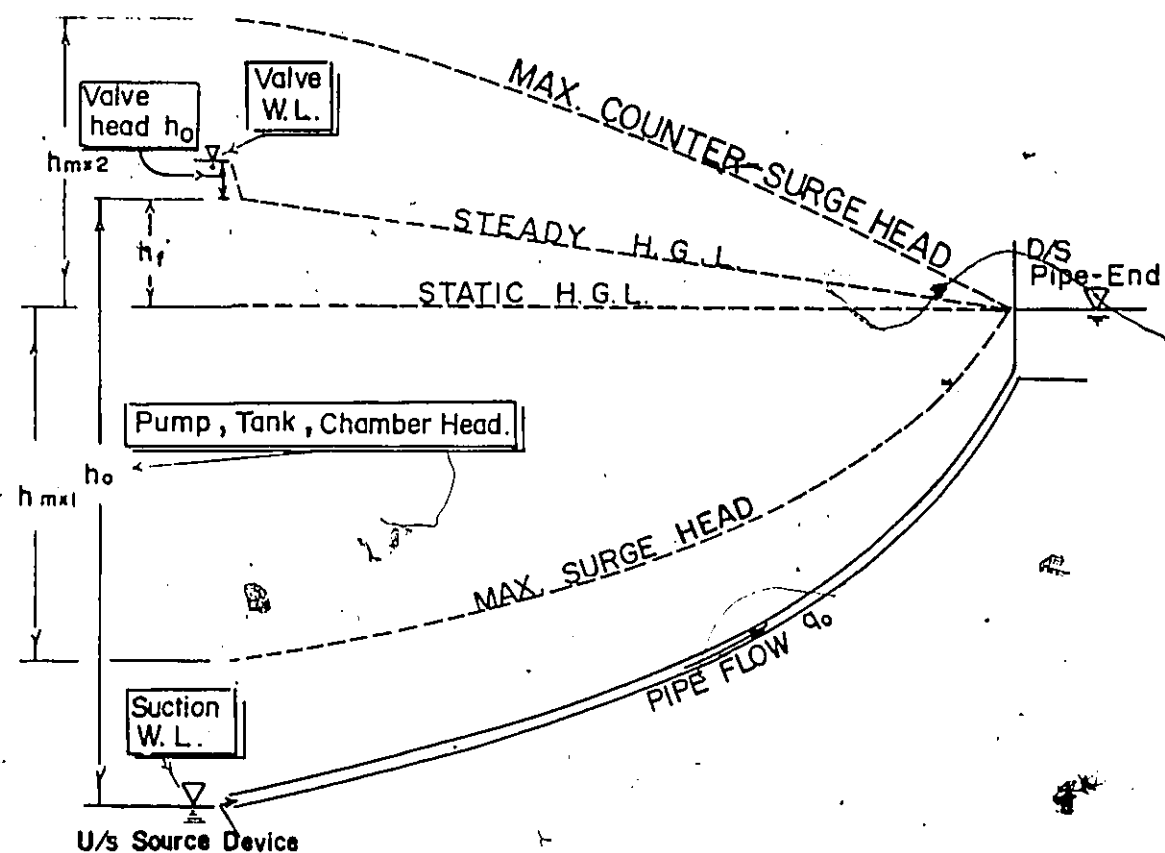
flows into the piping system during periods of rapid flow variation.

iv) Air chamber --- this control device functions in a manner similar to that of the surge tank, except that the potential energy is stored as compressed air.

These four protective devices differ significantly in terms of physical appearance, characteristics, reliability, complexity, flexibility, cost of construction, and operation and maintenance. However, these devices have a common basis in that all four attempt to achieve waterhammer protection by reducing the rate of flow change, hence minimizing the impact of transients. This common characteristic of unsteady flow movement is used in this thesis to minimize the complexity of the transient analysis. The hypothetical categorization of arbitrary flow closure is made in this research (Sec. 4.3) to provide a forum for a systematic and comprehensive transient analysis representing various surge initiations and protective devices.

A literature review of energy losses in unsteady pipe flows is conducted in the next chapter, covering various types of surge initiation and control devices.

a) Hydraulic Conditions



b) Alternative Devices at Source

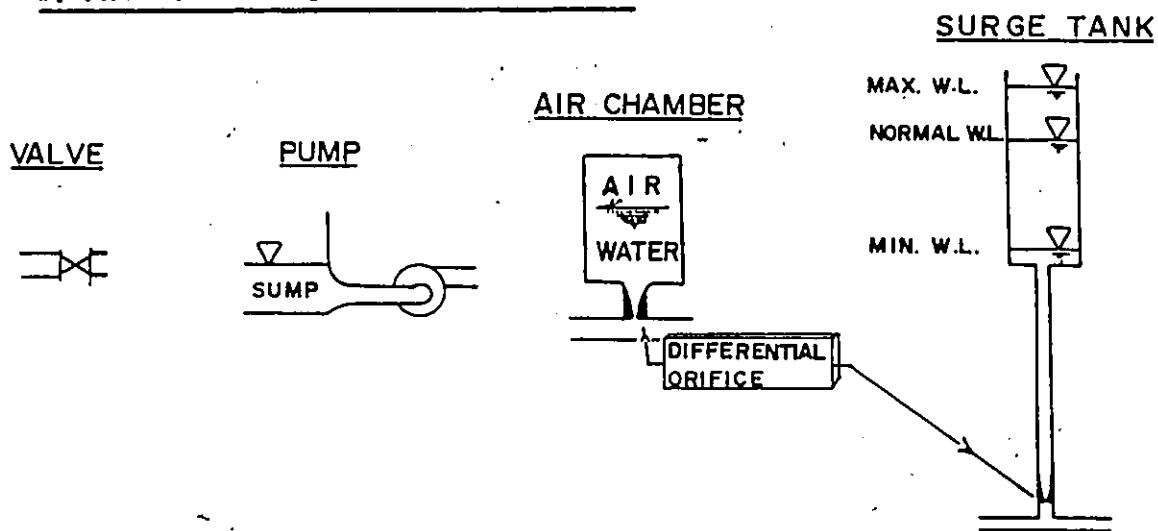


Fig. 1.1 — Schematics of a Pipe System with Various Protective Devices.

2.0 LITERATURE REVIEW

It is well known that flow through pipe systems is subject to energy loss due to the existence of fluid friction. In fact, friction is generally considered to be the primary reason for the decay of unsteady flow. Conversely, the quantity of friction loss is strongly affected by the unsteadiness in the flow.

The phenomenon of friction loss in unsteady pipe flows has been studied very extensively during this century. Ample analytical and experimental work has been conducted and documented. Some of this work, reflecting various aspects of unsteady friction losses in closed conduit systems, is presented below.

2.1 Types of Unsteady Flow

Analyses of unsteady pipe flows are classified in this thesis into two:

- i) Non-transient flow - in which the water column moves as a rigid body; and
- ii) transient flow - in which the elastic properties of pipe and water dominate the flow motion.

Each type of unsteady flow can then be further subdivided as discussed below.

2.1.1 Non-transient (inelastic) flow

Research dealing with this type of flow is commonly considered as a subject of physics as opposed to applied engineering, but it provides some basic concepts and theoretical background for later work in transient conditions. Two sub-types exist:

1) U-tube oscillating flow - where the total flow column in a conduit oscillates freely due to alternate conversion of energy between potential and kinetic form;

ii) Pulsating flow - where oscillatory motion in a conduit is sustained by an external periodic source of energy;

2.1.2 Transient (elastic) flow

This type of unsteady flow is more closely related to general engineering practice, and are central to the present research. General transient flows are normally conveyed by a pipe system, where the water levels at both the source and the pipe-end are constant during the transient condition. Two sub-types of transient flow further exist:

iii) Uni-directional flow - where the transient condition at the source location is not oscillatory, and results from a 'once only' initiating event or source; and

iv) Bi-directional flow - where an initiating disturbance results in subsequent periodic transient flow motion which travels back and forth (in two opposite directions) along the length of the conduit.

These two transient flows can exist in the same pipe system. For example, the case of uni-directional flow exists during the period of flow closure, while the bi-directional flow initiates upon the cessation of unsteady flow at the source.

2.1.3 Discussion of unsteady flow types

Generally, non-transient and transient flows can be distinguished by the relative extent of effects from the pipe and from the fluid elasticity. In the former flow type (U-tube and pulsating flows), the whole water column moves and can be approximated as being rigid. In the latter flow

type (uni-directional and bi-directional flows), the transient flow moves back and forth through the pipe system in a form of wave.

The above method of unsteady flow classification is unique among current works. Alternatives could be proposed; however, these categories of unsteady flow have been defined to guide the review of literature and the subsequent analysis of water hammer in this thesis, and proved to work well. It is noted that further subdivisions between laminar and turbulent conditions could be imposed in any of the above flows, depending on its flow regime.

Available literature, principal properties, and practical applications representing each individual type of unsteady flow will be discussed in this chapter. Table 2.1 groups the references cited and serves to guide the following discussion.

2.2 Unsteady Non-transient (inelastic) Flow

Generally, unsteady friction theories (f_u) for U-tube and pulsating flows have been well developed and documented. These have been frequently reviewed by other researchers.

2.2.1 Non-transient U-tube flow

The transient condition at the initial stage of the U-tube flow is generally disregarded because of its short time span. Hence the unsteady flow motion can be represented as a rigid water column (Wylie and Streeter 1978, page 205). In such a case, Newton's second law of motion is adequate to describe the unsteady flow characteristics.

2.2.1.1 U-tube laminar flow

A number of workers have examined unsteady flows in the U-tube environment for laminar conditions. Among the first of these were Richardson and Tyler (1929), who investigated the transverse velocity profile, from experiments on

oscillating U-tube flows. They observed that the maximum flow velocity does not remain in the center of the pipe cross section but instead is close to the wall. This tendency increases significantly with the frequency (ω) of periodic flow motion.

Sexl (1930) derived a theoretically exact solution of the Navier Stokes equations which explain this distortion of velocity profile. This was accomplished by assuming a sine wave variation of the pressure gradient along the pipeline. Uchida (1956) later extended this theory by superpositioning periodic oscillations on an average pipe flow, which is represented by a sum of sine waves. Practical application of this concept was achieved by Lambossy (1952), who established an empirical relation between the wall shear stress and the sinusoidal varying mean velocity (v) and acceleration (dv/dt).

Safwat and Polder (1973) conducted physical experiments on U-tube flow oscillation for laminar flow and for a critical zone approaching turbulent flow. They concluded that the acceleration or deceleration (dv/dt) of flow movement governs the unsteady friction factor (f_u), even for small frequencies of oscillation.

The current application of 'parallel flow theory' was first proposed by Jayasingh and Leutheusser (1972) who suggested a closed-form solution for an arbitrary pressure disturbance in a U-tube.

2.2.1.2 U-tube turbulent flow

Turbulent flow in U-tubes has not received as much attention as laminar flow. Hino et al (1977) suggested a frictional resistance law for an oscillatory pipe flow with smooth boundaries. For a turbulent flow the unsteady friction factor (f_u) was found to be governed by the flow velocity (v) and the flow frequency (ω) in the system.

2.2.1.3 Unsteady friction concept

In all of the literature related to the U-tube type flow, it is found that the unsteady friction factor (f_u) is described as a function of the flow velocity (v) and the acceleration (dv/dt) or oscillation frequency (w):

$$f_u = F(v, dv/dt) \quad 2.1$$

or

$$f_u = F(v, w). \quad 2.2$$

It should be noted that for non-transient (inelastic) flow conditions both the frequency (w) and deceleration (dv/dt) of the unsteady flow reflects the same characteristics, i.e. the inertia of unsteady flow movement.

The concept of flow oscillation is equivalent to a free vibration where the energy is being dissipated in a form of friction loss during the whole period of unsteady flow condition. This also implies that the change of velocity profile across the pipe section can affect greatly the shear forces associated with the conduit wall, and consequently the entire unsteady hydraulic phenomenon.

2.2.1.4 U-tube flow application

Sarpkaya (1969) performed a numerical experiment and concluded that the adoption of an unsteady friction factor (f_u) in a simple surge tank operation can result in the prediction of a significant decrease in damping effects. The pipe hydraulics in this surge tank system could be treated partly as a case of the U-tube problem (i.e. one-armed U-tube oscillation).

His finding, however, can be questioned as it is contradicted by results obtained from many other sources (see for example Safwat and Polder, 1973).

2.2.2 Non-transient pulsating flow

The forced vibration in the pulsating case balances energy dissipated by friction losses with external energy which is continuously input into the system during the entire unsteady flow condition. In the U-tube case, this external energy supply is not present.

2.2.2.1 Pulsating laminar flow

Laminar flow in pulsating systems was considered by Phillips and Chiang (1973), who developed a practical mathematical model of laminar pulsatile flow on the basis of Newton's equation of motion.

The major contribution in this area, however, was made by Ohmi et al (1976-81) who presented a series of papers on pulsatile laminar flow. Based on results of extensive laboratory work, they classified this unsteady flow pattern into three types, as follows:

- i) quasi steady if $W^* < 1.75$;
- ii) intermediate if $1.75 < W^* < 785$; and
- iii) inertia dominant if $W^* > 785$.

where $W^* = 1.57 (d^2/\nu) * w$.

Here W^* is dimensionless frequency and w is dimensional frequency of pulsatile flow. The terms d and ν refer to pipe diameter and fluid kinetic viscosity, respectively.

2.2.2.2 Pulsating turbulent flow

Schultz-Grunow (1960) conducted physical experiments on pulsatile flows in uniform (constant diameter) and non-uniform (variable diameter) pipe. He observed a similarity between the velocity distribution of decelerated flow ($dv/dt < 0$) in a uniform pipe and steady flow ($dv/dt = 0$) in a

divergent pipe (increasing diameter). This similarity can also be found between accelerating flow ($dv/dt > 0$) in a uniform pipe and steady flow in a convergent pipe (decreasing diameter). Results of these laboratory tests are unequivocal evidence of the distortion of velocity profile (or wall shear) by the unsteadiness of flow.

Kita et al (1980) presented a Reynolds shear stress model which can be used to determine the cross-sectional velocity profile and unsteady friction factor for turbulent pulsatile flows. In this case, the periodically oscillating flows are superimposed on an average pipe flow. During the performance of laboratory work, they observed that the quasi-steady model (f_q) underestimates the friction losses during the accelerating flow, and that opposite results can be found for the decelerating flow.

Baird et al (1971), Gerrard (1971), Mizushima et al (1973, 1975), Kataoka et al (1975), Maruyama et al (1976), and Ramaprian and Tu (1980) investigated the effect of unsteady friction factor by extensive laboratory experiments. Results of these works support the conclusion that the adoption of a quasi-steady friction factor (f_q) is not adequate for the analysis of this unsteady flow phenomena; an unsteady friction model (f_u) is required.

Ohmi et al (1976 - 1981) also conducted comprehensive and systematic physical experimental work on turbulent (as well as laminar) pulsating flows. Their observations on the relationship between the quasi-steady (f_q) and actual unsteady friction (f_u) factors essentially agree with those of Kita et al (1980). Ohmi et al also defined three turbulent flow regions as given below:

- i) Quasi-steady if $R_w^* < 0.021$;
- ii) Intermediate if $0.021 < R_w^* < 2.25$; and
- iii) Inertia dominant if $R_w^* > 2.25$.

where $R_w^* = W^* / (f_a * R)$. Here the term ' R_w^* ' is Ohmi's frequency Reynolds Number and ' R ' is the average flow Reynolds Number.

The above literature supports the concept that pulsating pipe flow possesses all the principal properties discussed for U-tube oscillating flow, differing only in that the former is a consequence of forced rather than free oscillation.

2.2.2.3 Pulsating flow application

The literature reviewed suggests that the physical relevance of pulsating flow assumptions is restricted to certain mechanical (e.g. piston pumps) or biological (e.g. circulatory) systems. Applications of pulsatile flow methods in normal piping systems (e.g. water and oil) are neither suggested nor supported by the literature.

2.2.3 Discussion of non-transient (inelastic) flow

The above literature review suggests that the following concepts are valid in unsteady non-transient flow systems:

i) The water column moves as a rigid body in the closed conduit, hence, the adoption of Newton's second law of motion (to be presented in Eq. 8.2) is sufficient for the analysis of unsteady flow behavior.

ii) Energy in a mass oscillation is being dissipated during both the unsteady flow movement by a form of heat transfer. This energy loss is termed friction dissipation (ED_f).

iii) The distortion of velocity profile across the pipe has in fact been observed during the rapid unsteady flow motion, suggesting a potential increment of frictional losses. This is defined as the unsteady friction (f_u) loss.

iv) The effect of flow inertia dominates the distortion of the velocity profile, in which the unsteady friction (f_u) loss increases very rapidly with the frequency (ω) or the deceleration (dv/dt) of the flow movement.

v) The quasi-steady friction factor, (f_q), is not adequate to explain unsteady friction phenomena. In addition to the parameter of velocity (v), the effect of inertia (ω or dv/dt) should be considered, as illustrated in Eq. 2.1 and 2.2 respectively.

Despite the validity of these concepts for non-transient (inelastic) flows, it is questionable whether they will also be relevant to transient (elastic) flows, which differ from the U-tube or pulsating flow phenomena.

2.3 Unsteady Transient (elastic) Flow

In the following review, the principal characteristics and practical applications of transient flow systems are discussed in a general manner (summarized in Table 2.1). A critical review and detailed analysis will be made in Chapter 8 of each specific non-transient friction theory. The new concepts and the theoretical background of these concepts will be introduced as this thesis develops, in Chapters 4 to 7.

2.3.1 Research and practice in transient flow

Conflicting conclusions have been made about the effect of unsteady friction (f_u) in a transient pipe flow system and some of these conclusions are documented in Table 2.2. Three points should be noted:

i) The concepts listed are fundamental but relevant to the research of unsteady friction loss;

ii) each conclusion, represented as an authorian's opinion in the table, has actually been made by a group of authors and researchers; and

iii) there is no specific trend of research direction regarding to these conclusions in terms of time and location.

Among many conflicting findings, two are most important and are discussed below:

1) Non-transient vs. transient flow --- The theories developed for non-transient flows are frequently extended to transient pipe flow systems. This has led to the general belief that the existence of friction can reduce transient pressure in a transient pipe flow system (Stephenson, 1984, Wylie and Streeter, 1978). Conclusions opposite to this have been made by Ruus (1977, 1981 and 1985). The lack of understanding of the frictional phenomenon (with steady friction factor) in transient flows greatly increases the difficulty of further analysis in the unsteadiness of friction factor.

2) Research vs. application --- Current engineering practice evident in the literature has predominantly adopted the simplifying assumption of a constant friction factor (f) in transient simulations (see discussion in Shuy and Apelt, 1983). However, even the quasi-steady friction factor (f_q) is considered inadequate by many researchers (e.g. Dally et al 1956, Carstens and Roller 1959, Denisov 1972, Popov 1974 and Shuy, and Apelt 1983). These researchers have strongly suggested that unsteady friction models (f_u) are more reliable and accurate.

One of the main tasks in this research is to resolve the above conflicts.

2.3.2 Transient uni-directional flow

2.3.2.1 Uni-directional laminar flow

During a review of available literature, no publication could be found on the subject of unsteady friction losses

during the flow closing or opening period in uni-directional laminar flow. This transient flow condition can however be found in some small diameter oil pipelines or machinery fuel systems.

2.3.2.2 Uni-directional turbulent flow

Ample analysis of turbulent flows has been conducted in uni-directional systems. Some of these are discussed below.

1) First order derivative momentum model (dv/dt) ---

The momentum theory was considered by Dally et al (1956) to investigate the effect of unsteady friction losses. Based on comparisons between the data measured from laboratory tests and from computations based on Newton's basic equation of motion, they found unsteady friction to be slightly greater than that obtained from the corresponding quasi-steady friction factor during the accelerating phase of flow variation (i.e. $f_u > f_s$ for $dv/dt > 0$). Conversely, they found that if flow is decelerated, the friction loss in unsteady flow would be much less than in the quasi-steady condition (i.e. $f_u \ll f_s$ for $dv/dt < 0$). The authors of the above paper concluded that the wall shear stress can be expressed by the instantaneous flow conditions, as expressed by Eq. 2.1 (i.e. velocity v and acceleration dv/dt).

Carstens and Roller (1959) extended this momentum friction concept and used a power law velocity distribution to establish a practical equation for the determination of the unsteady friction factor.

Considerable doubt has been expressed about this finding (Zielke 1966, and, Shuy and Apelt 1983), because the equation was derived from laboratory data exhibiting a large amount of scatter. Also, conflicting results were obtained under the same experimental conditions (e.g. Zielke, 1966).

2) Second order derivative momentum model (d^2v/dt^2) ---

Denisov (1972) and Popov (1974) performed a series of physical experiments in a short pipe. They concluded that the unsteady friction factor can also be influenced by the rate of change of acceleration (e.g. d^2v/dt^2). They observed the ratio of unsteady to quasi-steady friction factor (f_u/f_q) to be as large as 100, far greater than that found by Dally et al (1956). In the development of this second derivative momentum theory, Newton's second law of motion was also adopted, since the water column was considered to be rigid.

The existence of such a large ratio (f_u/f_q) strongly suggests inadequacy in the existing theory. The current practice of adopting the constant friction model (f) and the concept from this second derivative momentum model (f_u) of Denisov and Popov cannot be reconciled.

2.3.2.3 Applications to uni-directional flows

1) Surge tank simulation ---

Despite the lack of periodicity in the pipe flow, it is possible to relate the uni-directional flow to the first half of the period of mass oscillation in a surge tank system.

Sarpkaya (1959) adopted the expression from Carstens and Roller (1959) in the numerical simulation of a simple surge tank installation. He claimed that the maximum rise or drop of water level in the tank can be underestimated by as much as 20 % if the effect of unsteadiness on friction loss is disregarded. He also found that the period of flow oscillation in the pipe system differed significantly between a constant (f) and an unsteady friction (f_u) system. The conclusion made by Sarpkaya (1959) has frequently been used by some researchers as a bench mark for the evaluation of unsteady friction losses in transient flows (Ruus 1977, 1981 and 1984).

Compared with the experimental data obtained from the U-tube mass oscillation, it has been found that the unsteady friction factor affects both magnitude and phase angle of the oscillation during the first-half oscillation period much less than 20 % even for a laminar flow (Safwat and Polder 1973). Therefore, it is not likely that a deviation of 20% could appear in highly turbulent surge tank installations, which act as one-armed U-tube oscillators. The case of surge tank simulation will be further discussed in Chapter 8.

2) Valve and pump ---

Shuy and Apelt (1983) conducted a series of numerical model tests on valve and pump installations, and provided an excellent review of available literature. Five friction models were incorporated into the method of characteristics model to simulate the transient phenomenon. These five models included:

- i) Steady friction factor;
- ii) quasi-steady friction factor;
- iii) unsteady friction model in non-transient U-tube flow (Hino et al, 1976-1977);
- iv) first derivative momentum model in uni-directional flow (Daily et al 1956); and
- v) time-dependent friction model in bi-directional flow (Zielke 1966).

The last three are the current unsteady friction models under different types of unsteady flows.

No significant deviation of transients was observed from comparing these five friction model simulations. As a result of this, Shuy and Apelt (1983) concluded that there

is an urgent need for a new unsteady friction model to allow for general analysis of transients in turbulent pipe flow.

However, before a new model could be put forward, the following concerns should be addressed in their research:

i) Concepts inherited from non-transient flows --- Hino's (1976-1977) work was carried out with U-tubes. Here the frequency parameter (w) reflects the flow inertia characteristics in a U-tube flow oscillation. This is:

$$w = 0.25 v/l$$

where the parameters ' v ' and ' l ' are the flow velocity and fluid length in the tube.

In a valve or pump system, the frequency (w') of surge wave reflects the elastic properties in a transient pipe flow as:

$$w' = 0.25 a/l$$

where the parameter, ' a ', is the surge wave velocity.

In their analysis, Shuy and Apelt (1983) simply assumed the equality of these two frequencies. As a result of this assumption, an error should be expected in their analysis of transient phenomena during the flow closing period.

ii) Inelastic behavior assumed in transient flow --- The momentum friction model was based on the assumption of rigid water column movement in the pipe. Hence, it raises a problem as to whether or not Carstens and Roller's equation is appropriate in uni-directional flow simulations, in which the water column in a piping system is not rigid.

iii) Concept inherited from bi-directional flow --- Zielke's time-dependent friction theory was developed from the bi-directional flow (during the test with an instantaneous flow closure). Its eligibility in the uni-directional flow has not been confirmed, by the theoretical or by the laboratory treatment.

2.3.2.4 Discussion of uni-directional flow

Literature reviewed in this subsection suggests the need to clarify the following transient behaviors in uni-directional flow:

i) Elasticity --- One of the most important transient characteristics evident in the literature of uni-directional flows is the existence of elasticity, which is time-variable and spatial in the piping system, during the flow closing or opening period. The evident need exists to define under what conditions the rigid water column movement can be taken as an adequate approximation of unsteady flow condition.

ii) Friction loss --- Another aspect of uni-directional flows which arises as an important issue is that of the effectiveness of the friction loss itself (with a constant factor) during the flow closing period. In a transient condition where a steady friction (factor) loss is found to be non-effective, further analysis of unsteady friction (factor) is redundant.

iii) Flow inertia --- Yet another issue is related to the effect of the flow inertia (w or dv/dt) on unsteady friction factor (f_u). The above mentioned authors (Daily et al 1956, Carstens and Roller 1959, Denisov 1972, and Popov 1974) devoted much effort to retain or even to extend the relationship established from the U-tube or pulsating type flows, in flow closing or opening conditions.

Current research adopts the rigid flow concept and attributes all the deviation obtained from laboratory works (in the analysis of uni-directional flows) as a result of inaccuracy in the modeling of unsteady friction (factor).

Analysis in this research is designed to evaluate the relevance of these traditional concepts used in transient pipe flows.

2.3.3 Bi-directional flow

Bi-directional flow is restricted to the transient flow movement along the pipe, after the closure of source flow.

2.3.3.1 Bi-directional laminar flow

Laminar flow conditions in a bi-directional pipe flow system have been investigated by a number of authors. Brown (1962), D'Souza and Oldenburger (1964), Brown and Nelson (1965), Rouleau and Young (1965) and Holmboe and Rouleau (1967) adopted the 'linear transfer' technique to analyze transient flow and pressure at two specific locations in a pipe. They concluded that the wall shear stress is a function of flow velocity (v) and frequency (w). This transfer technique is usually tedious and not practical for the analysis of pipe systems, should any of the common surge protection devices be installed. Also, this technique was not designed for a detailed analysis, where the transient flow, pressure and friction factor are time-dependent and spatially varied.

Zielke (1966) utilized the frequency-dependent property of the above linear transfer concept, and developed a time-dependent unsteady friction model which can be conveniently solved by the method of characteristics. The model relates the wall shear stress to the instantaneous mean velocity and to previous changes in velocity. Laboratory tests with instantaneous flow closure were undertaken. All measured energy losses which were more than those predicted by the

quasi-steady friction model were attributed to the effect of an unsteady friction factor (f_u).

In Zielke's work, the predicted transient pressure was found to be closely correlated with that observed from the experiment, during the whole duration of the test. However, no attempt was undertaken by Zielke to interpret the real physical meaning of this time-dependent friction factor (f_u) during the transient condition. A question now arises whether the decay of transients is governed by the unsteadiness of friction loss, or in fact it is induced by other energy losses associated with transient flow but explained by unsteady friction. This is a key issue to be resolved in this research (Ch. 8).

2.3.3.2 Bi-directional turbulent flow

Turbulent flow in periodic systems was considered by Wood and Funk (1970), who derived a 'viscous model' for transient flows. They classify the pipe flow into laminar boundary layer and in plug flows. All the viscous losses were assumed to occur in the laminar layer, and its velocity distribution is expressed as a function of variable pressure gradient and core velocity. Subsequently, Funk and Wood (1974) extended their earlier work to determine the attenuation of pressure fluctuations. Some good agreement was observed between experimental data and the predictions of viscous theory. This was also found by Sauer (1969). As usual, however, the existence of friction was assumed to be the sole cause of any energy loss observed. The quasi-steady friction factor had to be increased up to four fold, depending on the initial Reynolds Number. The increment of the friction factor is homogeneously applied in the entire system, during the whole transient decaying period. Consequently, the major drawback of this theory lies with non-local and non-instantaneous correction of friction factor during the transient period. As such, it has not been possible to simulate the real friction variation which

is necessary to have a better understanding of unsteady flow phenomena.

Zielke (1966) suggested that the time-dependent component of his unsteady friction model be extended to use in turbulent flow, though the model was originally developed for laminar flow conditions. Trikha (1975) modified Zielke's equation for a more convenient and practical application with the method of characteristics. However, Trikha's simplified version fails in the application for many pipe systems which have relatively high turbulent flow regimes. Hirose (1971) suggested another simplification of Zielke's model for the use in turbulent flow conditions. In that study, some effort was given to interpret the phenomena behind Zielke's unsteady friction model development, and its use in turbulent flow conditions. However, no physical model test was undertaken by Hirose to verify his modified version of the numerical model.

2.3.3.3 Application of bi-directional flow

Shuy and Apelt (1983) extended their transient simulation to the cases of bi-directional flows. No significant difference of transient decay was found by the comparison of results from the five friction models described above. They blamed the failure of the tests on the lack of good unsteady friction models.

It is questionable, however, whether the failure of the transient response on various unsteady friction models was due to lack of an accurate and practical model (as claimed by these two authors), or whether there would in fact be no impact on transients from the unsteadiness of friction losses in those pipe hydraulic conditions, or insufficient time of transient simulation. This question will be answered towards the end of this thesis (Ch. 8).

2.3.3.4 Discussion of bi-directional flow

A review of available unsteady friction theories on bi-directional flows suggests that:

i) After the end of valve closure, the elasticity of pipe wall and water becomes the most important parameter in analysis of bi-directional flows. This implies that the friction factor, if varying, should be a function of time and space.

ii) The quasi-steady friction model is generally considered not adequate for the analysis of bi-directional flows.

iii) An unsteady friction model should be used to account for the anomalous energy losses.

iv) The time-dependent model (Zielke, 1966) appears to be superior to other current unsteady friction models (e.g. the linear transfer and viscous friction models), by considering the range of hydraulic conditions, the discretization of pipe system and the theory of the model. The time-dependent model possesses non-local and unsteady characteristics, and can be directly applied in both laminar and turbulent flow regimes.

Discussions of unsteady friction loss in bi-directional flows also suggest future research to assess the effect from other potential sources of energy losses; apart from friction.

2.4 Conclusions

General conclusions of the above literature review in unsteady friction losses are summarized in this section.

1) Theories in non-transient (inelastic) flows --- It appears that sufficient works have been conducted to establish the behavior of U-tube and pulsating flows, in

particular in laminar flow conditions. The well known friction theories established from these non-transient flows include:

i) It is concluded that the universal Newton's second law of motion can adequately be applied in the U-tube oscillating and pulsatile flow.

ii) It is also concluded that friction dissipation (ED_f) is the dominant source of energy losses.

iii) It has been confirmed that the well-known fundamental assumption, namely that the unsteadiness of friction factor (f_u) increases rapidly with the rate (w or dv/dt) of flow variation is correct.

iv) Finally, all past works unmistakably agree that the quasi-steady friction factor (f_u) is not adequate to explain the unsteady friction phenomena. The unsteady friction model (f_u) is recommended in literature and should be made a function of velocity and deceleration (or frequency).

One of the needs of present research is to assess whether these well established non-transient frictional concepts can or should be used for the cases of transient flows (e.g. water or oil pipe systems).

2) Problems in transient (elastic) flow analysis --- Although it is recognized that friction effects in general are of fundamental importance in hydraulic transient problems, there is no consensus as to how friction effects should be assessed. Inconsistent conclusions have been made, and confusion remains. For example,

i) The effect of energy loss in a steady friction (factor) system has not been well investigated for transients, particularly in uni-directional flows. A school of works concludes that the existence of

friction will reduce the transient pressure while another suggests the opposite effect.

ii) A similar conflict also arises as to whether the effect of unsteady friction factor (f_u) on the decay of transients is positive or negative. Two opposite conclusions have frequently been found from the available literature.

iii) Finally the conflict continues with respect to the selection of a friction model which should be applied to the analysis of general transients. There is literature supportive to the position that unsteady friction models need not be used. At the same time, there are those who maintain that gross errors will result if appropriate unsteady friction effects are not accounted for. At present, common practice uses the most elementary assumption, namely a constant friction factor.

The solution of these problems becomes very important, prior to the development of an unsteady friction model for transient flow conditions.

3) Utilization of current laboratory works --- The bulk of the available literature relies on laboratory work as a basis for its conclusions. Two conclusions follow from this:

i) Physical data for validation of new theory is already in existence; and

ii) Data are there but their physical phenomena cannot be properly explained.

The primary need at this point is a consolidation and re-interpretation of the current theoretical background of hydraulic transients.

4) Potential of numerical investigations --- One approach to the analysis adopted in this study is to carry out a systematic and comprehensive assessment based on numerical modeling techniques. Despite a preponderance of physically based work, there is ample justification in the literature for a numerical approach. Numerous other workers have relied on numerical results in the past, and such an approach is considered to be acceptable.

It should be noted that a uniform and systematic approach to representation of unsteady flow and pipe characteristics has not yet been devised. If a comprehensive analysis of the transient phenomenon is to be made, such a representation is required. Lack of such an analysis is held to be one factor contributing to conflicts in interpretation and analysis in the past.

5) Works of friction model development --- Among other things, this review has led to the following recommendations for the future research in friction model development:

i) First, there is a need to establish under what conditions, if any, constant, quasi-steady, or other unsteady friction effects can be used;

ii) Secondly, there is a need to provide an unsteady friction model, if required, to account for the energy losses in the transient conditions; and

iii) Finally, the current unsteady friction theories, if possible, should then be selected and recommended for the analysis in transient flow conditions.

6) Works in transient phenomena --- Prior to any further development of unsteady friction models, an evident and urgent need is to have a better understanding of transient phenomena. Apart from many other things, the analysis of the following two conditions should come first:

i) The existence of the elastic property in pipe systems and its mechanical transfer behavior affecting transient simulations; and

ii) the existence of various sources of energy losses, including those in frictional or in non-frictional forms.

Table 2.1 Summary of Literature Review

Flow System	Literature		Characteristics (Key words)	Practical Application
	Laminar Flow	Turbulent		
U-tube mass oscillation	Richardson & Tyler, 29 Sexl, 30 Uchida, 56 Lambossy, 52 Jayasinghe & Leutheusser, 72 Safwat & Polder, 73	Safwat & Polder, 73 Hino et al, 76-77	Free vibration Rigid water column Friction dissipation Shear stress profile Accumulated effect f_a is not adequate $f_a = F(v, dv/dt)$	Partial damping effect on: i) Two-way surge tank ii) Two way air chamber
Pulsatile flow	Phillips & Chiang, 73 Chambre, 78 Ohmi et al, 76-81	Schultz & Grunow, 60 Gerrard, 71 Balrd et al, 71 Mizushima et al, 73, 75 Kataoka et al, 75 Maruyama et al, 76 Kita et al, 80 Ramaprian & Tu, 80 Ohmi et al, 76-81	Same as above except Forced vibration	Machinery fuel system Biologic circulatory
Unl-direct- ional Flow	??????	Daily et al, 56 Carstens & Roller, 59 Denilsov, 72 Popov, 74 Vardy, 80 Van de Sande et al, 80	Rigid water column Friction dissipation Shear stress profile One way flow motion f_a is not adequate $f_a = F(v, dv/dt)$	All common surge control devices
Periodic Transient Flow	Zielke, 66 Brown, 62 & 69 D'Souza & Oldenburger, 64 Brown & Nelson, 65 Rouleau & Young, 65 Homboe & Rouleau, 67	Brown et al, 69 Sauer, 69 Wood & Funk, 70 Funk & Wood, 74 Hirose, 71 Triakha, 75	Friction dissipation Shear stress profile Time-dependent Linear transfer Viscous layer f_a is not adequate $f_a = F(v, dv/dt)$	Damping effect on: i) Valve closure ii) Pump trip-out with check valve iii) One-way tank iv) One-way air chamber
Surge control devices	??????	Sarpkaya 59, for two- way surge tank; Shuy & Apelt 83, for pump and valve system	Use f_a derived from u-tube or pulsatile; Conflict results and model selection; Random and short- time analysis	All common surge control devices

Table 2.2 Conflicts of Frictional Theories1) Present Practice and Research

Steady "f" used in current engineering practice By many engineers	or	Unsteady " f_u " is popular in current research By many researchers
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2) Responses from Steady Friction loss

Friction reduces transient By Stephenson, S.A. 1966	or	Friction induces transient By Ruus, Canada 1984
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3) Relevance of Unsteady Friction " f_u "

" f_u " is not important By Papatakis, USA 1984	or	" f_u " is very important By Denisov, USSR 1972
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4) Responses from Unsteady Friction " f_u "

" f_u " increases transient By Daily, USA 1956	or	" f_u " decreases transient By Brown, USA 1969
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5) Method of Analysis

Adopted elastic flow motion By Wylie, USA 1978	or	Adopted rigid flow motion By Carstens, USA 1959
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6) Basic Assumption

Forward velocity profile By Popov, USSR 1974	or	Backward velocity profile By Wood, USA 1970
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7) Status of Research

" f_u " theory has been solved By Zielke, Germany 1966	or	New " f_u " theory is needed By Apelt, Australia 1983
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3.0 RESEARCH OBJECTIVES AND SCOPE

3.1 Statement of Objectives

The foregoing literature review has identified the area of energy loss in current transient simulation and analysis as a potential research area. The principal objective of this research is defined as follows:

- 1) To resolve what conditions define the need for constant, quasi-steady, and unsteady friction (factor) loss assumptions in hydraulic transient analysis.

In short, to rationalize and consolidate some aspects of energy losses in waterhammer analysis.

In order to accomplish this principal objective, the following secondary objectives also need to be defined:

- 2) To appropriately interpret what are the physical phenomena governing transient flows. This may require a new analytical approach.
- 3) To find out what parameters induce energy losses in unsteady flows, so as to permit characterization of typical physical problems in a systematic manner.
- 4) To define where and how the friction loss (with constant factor) affects the transient behavior.
- 5) To determine (or develop, if necessary) an unsteady friction model suitable for the analysis of general transient conditions.
- 6) To systematically characterize the behavior of specific representative water hammer initiation and protective devices using the parameters defined in objectives 2 - 5.

This last objective evidently requires determination of what water hammer protective devices exist, and what physical ranges of conditions adequately define hydraulic systems encountered in engineering practice.

3.2 Scope of Research

The following physical conditions and boundaries in the closed conduit have been incorporated into the present thesis:

3.2.1 Hydraulic system

1) Fluid and flow Properties --- Fluid density and viscosity can be taken to be those of water at standard temperature and pressure. However to extend the possible range of use of the analysis, these quantities are varied to include fluid conditions representative of viscous fluids such as light oil. The upper bound of flow velocity in the pipe has been taken to be the order of 10 m/s.

2) Pipe material --- Representative pipe materials are taken to be cast iron, steel, concrete or plastic. These are the primary materials in engineering use today. Aside from dimensions, the only difference between these materials lies in the elastic modulus and hydraulic roughness.

3) Pipe geometry --- Pipe diameter and lengths cover a range which would be associated with projects ranging from local municipal (diameter of about 0.1 m) to major transmission lines (diameter of about 5 m) conduits, of varying length. Wall thicknesses have been chosen consistent with common practice for the materials, overall dimensions and pressure conditions considered.

4) Surge wave velocity --- A derived quantity, namely the surge wave velocity, is implicit in the above ranges of parameters. Under two extreme conditions in general engineering practice, the surge wave speed has a range encompassed by 150 to 1500 m/s.

5) Frictional condition --- Another derived quantity, the steady state friction loss has been taken over a conservative range of 0 to 500 m, which also is consistent with the values implied by the above parameter ranges.

6) System configuration --- The system is taken to be bounded by constant surface elevation reservoir at both ends. Single and uniform pipes are assumed.

3.2.2 Surge source and control

7) Surge initiation --- In all cases, the surge is taken to be initiated by flow decrease or shut-down at the upstream end of the system.

8) Surge control --- Four common surge control devices, namely, valve, pump, surge tank and air chamber have been considered.

3.2.3. Means of analysis

9) Friction models --- Constant, quasi-steady and various unsteady friction models reviewed in the literature have been adopted.

10) Flow regimes --- Laminar and various turbulent flow conditions are considered in the investigation of the effect of above friction model applications.

A wide and comprehensive analysis can be considered in the present research, by the adoption of above conditions.

3.2.4 Basic assumptions and constraints

The following assumptions have been considered in the present research and represent some specific constraints in the scope of this thesis.

11) Elasticity of pipe system --- The surge wave velocity 'a' is constant throughout the whole duration of transients. This assumption is reasonable since the plastic deformation of a pipeline (e.g. due to soil backfilling, pipe material and anchorage) is usually too small to affect the elastic behavior during the transient period.

12) Impacts of vapor pressure --- The existence of vapor pressure is suppressed. This restriction is imposed because of its complicating implication. In vapor forming conditions, multi-phase flow exists, and fluid boundaries are variable.

13) Local losses (junctions and bends etc.) --- Local losses are restricted to a condition where at most one local loss point, acting like an orifice, exists only at the surge initiation point. The local losses along the pipe system are usually relatively small and do not adversely affect the transient simulation.

14) Pipe network system --- Hydraulic impacts induced by the branching or looping system of a pipe network have not been considered. Normally, the hydraulics of the main line dominates both steady and unsteady conditions of a network system.

Although the present work is intended to be as general as is reasonably possible, it is felt that analysis incorporating these last three conditions (condition 12, 13 and 14) would be too wide in scope.

4.0 METHODOLOGY

The analytical approach adopted in this research consists of a number of techniques. The three most important are discussed in some length in this chapter. These are:

i) Dimensional reasoning, applied to the development of a systematic basis for the description of transient cause and effect,

ii) Selection of a model, used to widen the scope of the transient simulation involved in this research, and

iii) Definition of variables, utilized to represent arbitrary transient flow conditions.

In the course of this thesis methods for the determination of practical friction ranges and techniques for the quantification of important transient properties are also discussed, and these are documented in Appendix A.

4.1 Dimensional Analysis as part of Analytical Approach

Dimensional analysis has been employed in this work as a basic technique to resolve and integrate the results obtained during the course of this research. The technique involved is well documented (e.g. Birkhoff 1955, Sedov 1959, Langhaar 1962 and Yalin 1971). It has been applied in many fields of scientific research and some functional relationships established in this work might have been previously derived by other means. However, the application of dimensional reasoning in a systematic analysis of transient behavior has only recently been introduced (Fok 1978 and 1980).

4.1.1 Governing parameters in dimensional analysis

4.1.1.1 Steady state flow (governing parameters)

i) Non-frictional condition of steady flow --- Any dimensional property, m , of steady flow in a non-frictional pipe system can be expressed by the following functional relation:

$$m = F(l, d, \rho, v, g) \quad 4.1$$

where l = length;
 d = diameter;
 ρ = density;
 v = flow velocity; and
 g = gravitational acceleration.

By choosing ρ , g and d as repeated variables and using the Buckingham Pi theorem the five dimensional independent parameters in Eq. 4.1 are reduced to two equivalent dimensionless groups:

$$M = F(l/d, h_v/d) \quad 4.1a$$

where l/d reflects length of the pipe system (l); and $h_v/d = (v^2/2g)/d$ is the kinetic head per unit pipe diameter, which incorporates flow velocity (v).

In a pipe system, the pipe length characteristic (d/l) in Eq. 4.1a is not relevant, and the second variable (h_v/d) does not represent a meaningful physical property. Hence, these two parameters should be combined, because the property 'M' will then only be governed by kinetic head in a steady frictionless piping system, as expressed below:

$$M = F(h_v/l) \quad 4.1b$$

where h_v/l is the kinetic energy head per unit length of pipe.

11) Frictional condition of steady flow --- Two parameters related to frictional properties, i.e. effective pipe roughness (k) and the kinetic viscosity of the fluid (ν) are introduced here. The functional relation (Eq. 4.1a) is then extended to:

$$M = F(1/d, h_v/d, k/d, R) \quad 4.2a$$

where k/d is the relative pipe roughness, reflecting the parameter (k); and

$R = vd/\nu$ is the Reynolds Number, incorporating the viscosity (ν).

The two dimensionless frictional parameters (R and k/d) are normally replaced by a single frictional parameter (f), which can be determined from the Moody Diagram (Moody, 1944). This friction factor is:

$$f = F(k/d, R) \quad 4.2b$$

where the relevance of the pipe relative roughness (k/d) disappears for a smooth pipe.

With the aid of Eq. 4.2b, the four parameters in a frictional system (Eq. 4.2a) can be combined into a single parameter, as expressed below:

$$\begin{aligned} M &= F(1/d, h_v/d, F(k/d, R)) \\ &= F(1/d, h_v/d, f) \\ &= F(h_f/l) \end{aligned} \quad 4.2c$$

where $h_f/l = \{ [f * (1/d) * (v^2/2g)] / 1 \}$, representing the friction head per unit pipe length.

In a pipe system, the friction head is normally the predominant factor governing the steady hydraulics,

because the effect of the kinetic energy becomes insignificant, (i.e. $h_v \ll h_s$).

4.1.1.2 Unsteady state flow (for governing parameters)

1) Non-frictional system --- In addition to the five parameters in Eq. 4.1, the property 'm' of unsteady flows is governed by five more parameters:

$$m = F(l, d, \rho, v, g, d^S v/dt^S, t_a, a, x, t) \quad 4.3$$

where $d^S v/dt^S$ = rate of flow velocity change at source
and 'S' is the order of derivative;

t_a = time of flow closure;

a = surge wave speed;

x = location in the system; and

t = time of transient.

Certain combinations of parameters may be derived which are suitable for experimental planning. Among these,

$h_c = av/g$, is the Joukowsky head, which is the absolute maximum of transient pressure;

$t_c = l/a$, is the surge wave travel time.

Choosing the same basic quantities as in the steady flows (ρ , g and d), a dimensionless functional relation can be expressed as:

$$M = F(d/l, H_v, S, T_a, v/a, X, T) \quad 4.3a$$

where S = the shape of flow closure, reflecting rate of flow change at source ($d^S v/dt^S$);

$T_a = t_a/t_c$, is termed the time of flow closure, reflecting time span of flow change (t_a);

d/l = reflects the pipe length (l);

$H_v = h_v/h_c$, reflects the kinetic energy (h_v);

v/a = reflects the surge wave velocity (a);

$X = x/l$, reflects the location (x); and

$T = t/t_c$, reflects the time of transient (t).

In a transient pipe flow system, the roles of the three parameters, d/l , H_v and v/a in Eq. 4.3a are irrelevant and they can be usually ignored. Hence, the above equation becomes:

$$M = F(S, T_a, X, T) \quad 4.4$$

ii) Steady friction (factor) system --- As discussed previously for the steady flow condition (Eq. 4.2c), the kinetic head (H_v) should be replaced by the friction head (H_f) in a frictional system. Hence, the property, M , in a steady frictional pipe system is governed by:

$$M = F(H_f, S, T_a, X, T) \quad 4.5$$

where H_f is determined from a steady friction factor, which remains constant during the transient period.

iii) Quasi-steady friction (factor) system --- As is well known, the friction factor actually varies with the flow regime condition. In this case the quasi-friction factor (f_q) should be included in the investigation. This is expressed as:

$$M = F(H_f, S, T_a, X, T, f_q)$$

The pipe roughness (k/d) of Eq. 4.2b remains constant during the transient period and hence can be neglected. On the other hand, the unsteady Reynolds number, $R(T>0)$, follows the unique 'hydraulically smooth' line and hence its characteristics can also be determined at the initial steady condition, i.e. $R(T=0)$. Therefore, any transient property of a quasi-friction system can be determined as:

$$M = F(H_f, S, T_a, X, T, R) \quad 4.6$$

All the dimensionless parameters inside Eqs. 4.4 to 4.6 can be pre-determined from initial steady state conditions, prior to the initiation of transients.

iv) Unsteady friction (factor) system --- In the literature review (Ch. 2) it was suggested that the transient property 'M' may also be governed by the existence of unsteadiness in the friction factor (f_u). The functional relationship of an unsteady friction system can be expressed as:

$$M = F(H_z, S, T_a, X, T, R, f_u) \quad 4.7$$

The unsteady friction factor (f_u) of Eq. 4.7 becomes part of transient properties and hence cannot be pre-determined from the initial steady state conditions. Investigation of the application of this unsteady friction factor (f_u) in transient flow conditions is one of the primary objectives in this research.

The following paragraphs discuss some potential problems affecting the analysis of the unsteady friction factor from the view point of dimensional reasoning.

4.1.1.3 Relevance of unsteady friction factor (f_u)

Traditional analyses attribute all anomalous losses to the distortion of the shear stress distribution. Current unsteady friction theories have been developed on this basis. Two major concerns are addressed here regarding the adoption of this traditional analytical method.

1) Elasticity --- Review of available literature leads to a common belief that the unsteady friction factor (f_u) is governed by the flow velocity (v) and the flow deceleration rate (d^2v/dt^2), in which the friction factor is equivalent to Eq. 2.1. This can be expressed as:

$$f_u = F(v, d^2v/dt^2) \quad 4.8$$

Rewritten in a dimensionless term, this becomes:

$$f_u = F(R, S) \quad 4.8a$$

where R =Reynolds Number, reflecting flow velocity (v)
 S =Shape factor, reflecting deceleration (d^2v/dt^2)

Inserting Eq. 4.8a into Eq. 4.6,

$$\begin{aligned} M &= F(H_z, S, T_a, X, T, R, F(R, S)) \\ &= F(H_z, S, T_a, X, T, R) \end{aligned} \quad 4.8a$$

Here, Eq. 4.8a and 4.6 are identical. This means that the transient property 'M' can be determined by the rate of flow deceleration (d^2v/dt^2), regardless of which friction model is applied.

One of the major concerns addressed in this research is how to determine whether a quasi-steady (f_q), or an unsteady (f_u) frictional relationship prevails in transient conditions.

ii) Non-frictional losses --- Another major concern addressed in this thesis is the allocation of other potential losses unrelated to the unsteadiness of the friction factor (f_q). This concern is not addressed in the available literature. Should the non-frictional losses indeed exist but are overlooked, the effect of another uncertain parameter in Eq. 4.7, i.e. the unsteady friction factor (f_u), may be unrealistically increased. This will be discussed in detail later in Chapter 8.

For simplicity and consistency, the unsteady friction factor (f_u) is retained in this thesis as a governing parameter, and the term 'unsteady friction system' is adopted to keep it distinct from other friction terms, even though the unsteadiness of the friction factor (f_u) may or may not be relevant.

4.1.1.4 Discussion of functional relations

The relationships expressed in Eqs. 4.4 - 4.7 attempt to provide a clear approach to the analysis of transients, covering the non-friction condition ($H_f=0$), the constant friction factor (f), the quasi-steady friction factor (f_q) and the unsteady friction factor (f_u).

4.1.2 Transient properties (M) in dimensional analysis

4.1.2.1 Pressure as a transient property

The most common practice in physical or numerical experiments for transient evaluation is based on the measurement of pressure head variation $H_m(X,T)$, in which one (e.g. $X = 0$) or several points (e.g. $X = 0, 0.25, 0.5$ and 0.75) along the pipe are analyzed.

4.1.2.2 Energy as a transient property

A new method is introduced in this research for quantifying transient pressure in the system. The transient energy, $E_m(T)$, at a specific time T , is defined herein as the integration of the transient pressure head $H_m(X,T)$ along the entire length (i.e. $L = 1$) of the system as shown below:

$$E_m(T) = \int_0^1 H_m(X,T) dX \quad 4.9$$

The adoption of this new term has three advantages:

- i) It reflects the transient properties (i.e. elastic energy) of the whole pipe system, instead of the traditional limited point-specific pressure condition;
- ii) it allows a new concept of analysis to incorporate all forms of energy exchange mechanisms, e.g. mechanical transfer and energy dissipation. This new

concept of energy balance will be introduced in Ch. 5 and adopted throughout this thesis; and

iii) it simplifies the functional relationship in equations 4.4 to 4.7 by eliminating the distance parameter, X .

Consequently, the above four functional equations (Eqs. 4.4 - 4.7) adopted for analysis of transient phenomena can now be simplified as shown below:

$$E_m(T) = F(S, T_a, T) \quad 4.4a$$

$$E_m(T) = F(H_s, S, T_a, T) \quad 4.5a$$

$$E_m(T) = F(H_s, S, T_a, T, R) \quad 4.6a$$

$$E_m(T) = F(H_s, S, T_a, T, R, f_u) \quad 4.7a$$

where $E_m(T)$ refers to the elastic energy of the entire pipe system at time T .

4.1.2.3 Maximum energy as a transient property

The maximum value of a transient is normally the most important observation for general engineering practice. The maximum transient energy (E_{mx}) is defined herein as:

$$E_{mx} = E_m(T_{mx}) = \int_0^1 H_m(X, T_{mx}) dX \quad 4.10$$

where T_{mx} is the time of occurrence of maximum transient energy (E_{mx}).

Maximum transient energy (E_{mx}) in this thesis consists of:

- 1) maximum surge energy (E_{mx1}) - maximum transient energy occurred during the flow closing period;

ii) maximum counter-surge energy (E_{mx2}) - maximum transient energy occurred during the first transient flow reversal; and

iii) peak decaying energy (E_T) - the peak value of transient energy in the vicinity of time, T , which occurs after the counter-surge period (i.e. $T > T_{mx2}$).

The parameters T_{mx} , T_{mx1} and T_{mx2} are times of occurrence corresponding to maximum transient (E_{mx}), maximum surge (E_{mx1}) and maximum counter-surge (E_{mx2}) energy respectively.

The functional relationships concerning maximum transient energy can now be further simplified for the non-frictional, steady, quasi-steady and unsteady friction (factor) system respectively as below:

$$E_{mx} = F(S, T_Q) \quad 4.4b$$

$$E_{mx} = F(H_z, S, T_Q) \quad 4.5b$$

$$E_{mx} = F(H_z, S, T_Q, R) \quad 4.6b$$

$$E_{mx} = F(H_z, S, T_Q, R, f_u) \quad 4.7b$$

where the E_{mx} refers to events of maximum surge (E_{mx1}), maximum counter-surge (E_{mx2}) and peak decaying (E_T) energy.

4.1.3 Surge control devices in dimensional analysis

In conducting this work, it was found that the characteristics of the four common control devices could also be conveniently represented by similar relations as expressed in the above functional relationship equations. For simplicity, only the cases involving the use of the steady friction factor (f) are provided in the following discussions. However, where a quasi-steady or an unsteady friction model is required, the Reynolds Number (R) or the unsteady friction factor (f_u) may also need to be included.

4.1.3.1 Valve as a surge control device

In a valve operation a new parameter, valve head (H_v), reflecting the local orifice loss through the valve, must be added. The terms S and T_v of flow closure now become the shape and time of valve closure. The functional relationship of maximum transient conditions can be expressed for valve closure in a frictional system as:

$$M = F(H_v, S, T_v, H_0) \quad 4.11$$

4.1.3.2 Common characteristics of Pump, tank and chamber

The functional relations of the other three devices are complicated by the fact that the shape factor (S) and time (T_v) of flow closure cannot be pre-determined from the steady state conditions. These two parameters themselves become part of transient properties (M), in which they have to be determined from other dependent variables. However, the new term device head (H_v) of each specific device is usually known for the steady state condition.

Another new term T_m , mechanical travel time is also required which will be defined for each individual device below. In general, the mechanical travel time (T_m) is very complex in nature as well as strictly device-specific. Its magnitude is governed by a number of pipe hydraulic (e.g. steady flow Q_0 and head H_0) and various device (e.g. types and sizes) characteristics, in which it can also be determined prior to the initiation of transients.

In a case that the device is equipped with a check valve to prevent flow reversal, its damping behavior is similar to that of a valve operation. Otherwise, the whole body of water can travel back and forth along the pipeline as in the case where the check valve is non-existence.

4.1.3.2 Pump as a surge control device

The pump head (H_p) is the height between the suction level and the hydraulic grade at pump. The mechanical travel time of a pump system can be estimated as:

$$T_m = 0.514 [(WR^2 * NR)/(g * TR)] / t_c \quad 4.12$$

where WR^2 = inertia of pump and motor;

NR = pump speed;

TR = pump torque; and

$t_c = l/a$

Accordingly, the functional relationship of any transient property 'M' is expressed as:

$$M = F(H_s, H_p, T_m, T) \quad 4.12a$$

In the case that only the critical transient events are required, the transient property 'M' consists of the magnitudes (E_{mx}) as well as the time occurrences (T_{mx}) of the maximum transient energy.

4.1.3.3 Air chamber as a surge control device

The chamber head (H_{ch}), commonly adopted in the current practice, is the sum of normal device head (H_p) and atmospheric pressure. The mechanical travel time (T_m) of an air chamber system can be considered as an indicator of the time required to discharge the total air out of the chamber at a rate of steady flow. The value of this ' T_m ' can also be pre-determined as expressed below:

$$T_m = (c_o/q_o) / t_c \quad 4.13$$

where c_o = initial air volume inside the chamber;

q_o = steady flow rate of the pipe system; and

$t_c = l/a$

In normal practice for a two-way air chamber, the transient characteristics of an air chamber system may be further complicated by the installation of an orifice. An additional parameter ($H_{ox} = h_{ox}/h_c$) is needed to account for the head loss (h_{ox}) when the steady flow (q_o) passes through the orifice. The differential orifice can also be represented by an additional term (K_{ox}), reflecting the ratio of head losses between the incoming and outgoing flows with the same rate.

In the case of a two-way air chamber system, the functional relationship of an air chamber can be expressed as:

$$M = F(H_z, H_o^*, T_m, H_{ox}, K_{ox}, T) \quad 4.13a$$

In the case of a one-way air chamber system, the above equation can be simplified as:

$$M = F(H_z, H_o^*, T_m, T) \quad 4.13b$$

4.1.3.4 Surge tank as a surge control device

Current engineering practices adopt the rigid water column concept to investigate the unsteady flow behavior of a surge tank system (Parmakian 1963, page 128 - 129 and Fok 1980). In reality, the elastic properties of the system may play a significant role in surge tank simulation, particularly in the case of an orifice installation. A new mechanical travel time (T_m) is introduced in this research, as expressed below:

$$T_m = 0.5 \left[\left(\frac{l}{g} \right)^{0.5} * \left(\frac{d^*}{d} \right) \right] / t_c \quad 4.14$$

l = pipe length;
 d^* = diameter of surge tank;
 d = diameter of pipe; and
 $t_c = 1/a$

Accordingly, the functional relationship for two-way surge tank transient characteristics can be expressed as:

$$M = F(H_z, T_m, H_{oz}, K_{oz}, T) \quad 4.14a$$

A one way tank is common in some low head water and sanitary forcemains (Stephenson, 1972 and Fok 1980). The difference of water level at the tank and the H.G.L. at the tank is commonly defined as the tank head (H_o). The relation is then expressed as:

$$M = F(H_z, H_o, T_m, T) \quad 4.14b$$

It should be noted that the surge tank functions in a very similar manner to that of an air chamber. The difference between these two devices is that the energy is stored in the elevated water body, instead of the compressed air.

4.1.4 Discussion of dimensional analysis

Dimensional analysis has been used in this section to develop sets of functional relations between transient properties and their governing parameters.

1) Governing parameters --- The development from the above dimensional analysis provides a list of potential relevant parameters governing the transient characteristics in a closed conduit. These include:

- i) Flow closing characteristics, such as S and T_c ;
- ii) Frictional conditions, such as H_z , R , and f_w ;
- iv) Time and location of transient, as represented by T and X respectively.

2) Transient properties --- Two sets of functional relationships have been established for:

- i) Time history --- Eqs. 4.4 - 4.7 (part a) can be used for the analysis of transient energy history, i.e. $E_m(T)$ vs. T ; and

ii) Maximum events --- Eqs. 4.4 - 4.7 (part b) can be used for the investigation of maximum transient energy achieved, i.e. E_{mx1} , E_{mx2} and E_T .

3) Frictional system --- Four functional relationships have been established for directing the present research in different frictional systems. These include:

i) Non-frictional system (Eq. 4.4);

ii) Steady friction (factor) system (Eq. 4.5);

iii) Quasi-steady friction (factor) system (Eq. 4.6);

iv) Unsteady friction (factor) system (Eq. 4.7).

4) Energy losses --- The effect of unsteadiness in the friction factor (f_u) may be greatly exaggerated by the omission of the following two energy parameters:

i) Elastic energy as a result of the flow velocity change; and

ii) energy losses associated with transient flows, but unrelated to the unsteadiness of friction factor.

5) Practical applications --- The functional relations have been established in Eqs. 4.11 - 4.14, for valve, pump, air chamber and surge tank respectively.

Detailed analytical work to investigate the functional relationships on these four common surge control measures are not discussed in this thesis, but will be presented in a series of related technical reports to be published separately. Highlights of these works are appended in Appendix C.

The complexity and variability of these control device characteristics indicate a need to define unsteady flow

variables (i.e. S and T_a) to describe the transient phenomena in a consistent and uniform manner. The characteristics of these two variables are identified and are discussed later in this chapter.

4.2 Model Simulation as part of the Analytical Approach

4.2.1 Selection of modeling technique

In formulating the above non-dimensional relationships by the use of the Buckingham Pi theorem, there is no requirement to validate the relevance of the governing parameters inside those functional relationship equations. It is tactically assumed that any irrelevant dimensionless parameter can be discarded, while the relevant ones will be quantified by further analysis. In this respect, two general categories of transient simulation are possible to investigate the functional relationships developed from the dimensional analysis. One alternative is to carry out physical model tests in the laboratory or field; the other, is to undertake simulations using numerical models.

4.2.1.1 Relevance of modeling

To provide insight into the most appropriate approach to investigate the transient behavior in the present research, the following paragraphs discuss some of the advantages and disadvantages of the two alternative approaches:

- 1) The physical model approach, often termed 'black-box', can be used to determine the relevance of each governing parameter, while the physics of the system and fluid-flow behavior need not be well addressed. For example, the current laboratory works in unidirectional transient flows were based on the rigid water column movement without the knowledge of its validity. The results thus obtained may be mistakenly attributed to the hydraulic responses from the existence of friction, instead of the real governing factor, i.e. the existence of elasticity in the system.

11) The numerical model approach, often applying the method of characteristics technique, takes the existence of the surge wave into consideration, thus ensuring incorporation of the pipe and fluid elasticity in the system. On the other hand, it is very rare and sometimes impractical to develop a numerical model which can examine all the dynamic and thermodynamic equations governing transient phenomena encountered in pipe flow systems. For example, the energy losses associated with the periodic transient flow movement (traditionally claimed as a result of the unsteadiness of the friction factor, f_u) have not been accounted for in the current numerical model. A physical model is often required to determine these losses.

In order to utilize the advantages of these two models, the most proper analytical approach to scientific applications of general transient flows should incorporate both numerical and physical modeling analysis.

4.2.1.2 Choice of modeling priority

To undertake a thorough numerical model analysis of energy losses in transient conditions is time consuming and laborious. To conduct a physical model analysis with the same wide scope is even worse. To carry out both modeling analyses was considered prohibitive.

On this basis, it was decided that an numerical modeling simulation as supported by physical modeling data borrowed (with courtesy) from available sources provided an acceptable analytical procedure for this study.

The wide scope of work proposed in this thesis (Sec. 3.2) does specify further the need for such a choice of modeling techniques as discussed below:

- 1) The primary advantage of numerical model simulations lies in the ability to assess a wider range of

parameters under a wider range of hydraulic conditions with less effort than physical approaches require. Obviously, to physically assess the range of conditions considered in this work would require many man-years of effort. Relying on physical experiments would therefore require a significant compromise on the range of conditions tested, or an inordinate level of effort;

ii) Except for some simple cases such as an instantaneous flow closure, the application of physical model tests to general transient flows is exceedingly difficult. Further, scale effects, the speed of transient response and limitations in the accuracy of measurements make the interpretation of model results uncertain;

iii) Subsequent testing of numerical results against field observations is still possible. The ultimate validity of theories based on numerical techniques can still be subsequently verified; and

iv) Use of a numerical model based on accepted theory makes it possible to draw conclusions regarding the consequence of existing fundamental theory, which is inherently impossible with data obtained from a single modeling approach.

Since the thrust of this research is towards an effective interpretation and consolidation, and possibly an extension of existing fundamental theory, a numerical model becomes the clear method of choice for this research. It therefore remains to obtain a particular numerical model for use in this work. The establishment of such a numerical model is discussed in the next subsection.

4.2.2 Development of the numerical model

Part (a) of Table 4.1 outlines the key features and principal characteristics of the present numerical model

(Otthammer, 1987), and this table serves as a guide to the discussions of Subs. 4.2.2 and 4.2.3.

4.2.2.1 Method of characteristics for model development

As indicated by the authors (Wylie and Streeter, 1978) and many pioneers (Chaudhry 1979, Waters 1979), the method of characteristics has a number of advantages in water hammer analysis. For this work, the primary advantage lies in the essentially analytic nature of the method of characteristics; with careful choice of boundary and other hydraulic conditions, numerical approximations are essentially eliminated, and the model is an accurate reflection of the fundamental differential equation describing water hammer.

The framework of the present numerical model has been written specially for the scope of work intended for this thesis. Thus, the basis of the model used in this work is that of a strong and well validated algorithm (established by Wylie and Streeter 1978), while the overall model is a new and effective development particular to this research but applicable to a variety of physical cases (Otthammer, 1987).

The theoretical background of the model structure is described below. Some specific modeling treatments related to unsteady friction losses are also presented herein.

4.2.2.2 Equations governing hydraulic transients

Transient phenomena in a pipe system is commonly considered as a one-dimensional flow condition. The dependent variables consist of the pressure head (h) and the mean sectional velocity (v), and the independent variables include the time (t) and location (x) along the pipe.

Three fundamental equations are commonly adopted to describe the transient flow behavior in a pipeline. These are:

i) The equation of motion,

$$g \delta h / \delta x + \left(v \delta v / \delta x + \delta v / \delta t + f \cdot v \cdot |v| / (2d) \right) = 0 \quad 4.15$$

where, g = gravitational acceleration;
 d = pipe diameter; and
 v = flow velocity.

ii) the equation of continuity,

$$\delta \rho / \delta t + \rho \delta v / \delta x + v \delta \rho / \delta x = 0 \quad 4.16$$

where, ρ = the fluid density, and

iii) The equation of state

$$\delta \rho / \rho = \delta p / K \quad 4.17$$

where, K = the bulk modulus of the fluid.

The elasticity of a pipe system is related to both the fluid and pipe wall. Hence, the surge wave speed 'a', which is currently used to represent this transient property, can be estimated as:

$$a = \left\{ (K/\rho) / [1 + (K/E_1) \cdot (d/e_1) \cdot c_1] \right\}^{0.5} \quad 4.18$$

where, E_1 = modulus of pipe elasticity

e_1 = thickness of pipe wall

c_1 = pipe anchorage constraint ($0.8 < c_1 < 1$).

4.2.2.3 Equations used for method of characteristics

The characteristics method adopted in the present numerical model solves the above equations, by converting them into differential equations as follows:

$$\begin{cases} (g/a) \cdot (dh/dt) + dv/dt + (f/2d) \cdot v \cdot |v| + (g/a) \cdot v = 0 \\ \{ \\ \text{along } dx/dt = v + a \end{cases} \quad 4.19a$$

$$\begin{cases} -(g/a)*(dh/dt) + dv/dt + (f/2d)*v|v| + (g/a)*v = 0 \\ \{ \text{ along } dx/dt = v - a \end{cases} \quad 4.19b$$

The above characteristic relations can be approximated by finite difference expressions and then integrated numerically with use of the digital computer.

4.2.2.4 Numerical treatment of friction

The friction term, $(f/2d)*v|v|$, which plays a very important role in the present research, has been directly incorporated into the equations of characteristics method. Some numerical treatments related to this friction term are discussed below:

1) Stability --- It has been reported that some inaccuracy in transient simulations can be induced due to the non-stable integration of the friction term (Wylie and Streeter 1978, and Wylie 1984). A stability criterion has been developed for the discretization of high frictional systems, in which the first-order model is acceptable if the 'Friction Stability Number' ($FS_{\pm}$), is less than 1 (Wylie and Streeter 1978, page 52). This means that:

$$FS_{\pm} = (f*dt*v)/(4d) < 1 \quad 4.20$$

where, 'dt' is the duration of each time step.

This criterion has been taken into consideration during the model set-up in the present research. The maximum value of Friction Stability Number is less than unity, even for the upper bound of friction head used in this work ($FS_{\pm} = 0.25 < 1$, at $H_{\pm} = 10$). The piping system is discretized into 20 segments for the analysis throughout this research.

ii) Order --- The second-order iteration of the friction term, which was developed by Wylie and Streeter (1978, page 53 and 65), has also been used in this thesis for the sensitivity test on the order of models. It has been found that less than 5 % error may be induced by the use of a first-order model in the above mentioned upper bound of friction head and system discretization conditions. The constant friction model (f) was used in this test.

The second-order model with the use of other friction factors (f_a or f_u) has not been well established, and consequently its application for other quasi and unsteady friction models is not possible at this time.

Based on the above rationale, the first order friction iteration is adopted in the present analysis.

iii) Linearity --- It has been recommended by some researchers (Shuy and Apelt, 1983) that the non-linear iterative solution is a necessary procedure to improve the accuracy of the present first-order model. This non-linear solution treats the friction term as an average of both the beginning and the end conditions of each time step, while the normal linear solution only considers the beginning condition. In contrast to Shuy and Apelt's prediction, no significant difference was found in this research from sensitivity testing on the linearity within the system friction and discretization conditions as described above (less than 1 % error).

In view of the insensitivity of the order and linearity of the model application within the broad range of frictional conditions, and the consistency required for the use of various friction models, the linear first-order friction iteration, which is the most fundamental modeling technique of friction treatment, has been used throughout the present research.

4.2.3 Friction losses in transient flow simulation

Three methods of treating the friction (factor) have been incorporated into the Method of Characteristics model used in this study. This allows for the computation of transients in the categorized flows under a wide range of hydraulic conditions. These are:

- i) constant (f),
- ii) quasi-steady (f_q), and
- iii) unsteady (f_u).

4.2.3.1 Constant friction factor in friction prediction

Current engineering practice commonly uses a constant friction factor in the analysis of hydraulic transients. This factor, denoted ' f ', (commonly known as Darcy-Weisbach friction factor) can be obtained from the initial steady flow condition. It actually represents a special case of the quasi-steady factor, and does not need special discussion.

4.2.3.2 Quasi-steady friction factor in friction prediction

This friction model uses variable friction factors chosen at each point along the system such that friction for an instantaneous value of flow velocity matches friction for fully developed steady flows with the same cross-sectional average velocity. As such, it is virtually an unsteady method, although based on steady state friction conditions.

The quasi-steady friction factor, f_q , can be obtained directly from the Moody diagram. For an efficient computer application, explicit forms have to be developed. These explicit relations are presented below.

i) Laminar flow --- For Reynolds Number (R) below 2000, the friction factor can be determined from the following theoretical equation:

$$f_a = 64 / R \quad 4.21$$

ii) Turbulent flow -- For Reynolds Number $R > 3000$, Wood's empirical equation (1966) is selected in the present study as shown below:

$$f_a = b_1 + b_2 R^{-0.22} \quad 4.22$$

where, $b_1 = 0.094 (k/d)^{0.22} + 0.53 (k/d)$;

$b_2 = 88 (k/d)^{0.44}$; and

$b_3 = 1.62 (k/d)^{0.134}$.

In a hydraulically smooth condition, the relative pipe roughness $k/d = 10^{-5}$.

iii) Critical Zone --- For a flow with $2000 < R < 3000$, characteristics related to the friction losses have not been well defined. For simplicity in numerical model application, the quasi-steady friction factor f_a has been assumed to decrease linearly with Reynolds Number (R).

4.2.3.3 Unsteady friction factor in friction prediction

There have been numerous unsteady friction models (f_u) available in literature. Those capable of incorporation into the present numerical model are:

1) Uni-direction flow ---

i) First derivative momentum model --- Daily (1956) first proposed this momentum friction model and later extended by Carstens and Roller's (1959). Its functional relationship can be expressed as:

$$f_u = F(v, dv/dt) \quad 4.23$$

ii) Second derivative momentum model --- Denisov (1972) and Popov (1974) proposed this momentum friction model which is in a form:

$$f_u = F(v, dv/dt, d^2v/dt^2) \quad 4.24$$

2) Bi-directional flow ---

iii) Time-dependent model --- Zielke (1966) introduced this unsteady friction model for the decay of transient energy. The relation between the unsteady friction factor and its relevant parameters can be expressed as:

$$f_u = F\left(v, \int_{t_1}^1 dv(t)/dt\right) \quad 4.25$$

where the second component is the summation of past historical changes of transient flow.

iv) Alternative time-dependent models --- Hirose (1971) and Trikha (1975) modified Zielke's time-dependent friction model, to obtain a simplified version.

All these unsteady friction models have been tested and most rejected as not useful in this work. The third one (Zielke's time-dependent model) is found to be useful, but for a different theoretical basis. More detailed discussions of these current unsteady friction models will be conducted as they arise (Ch. 8).

4.2.3.4 Surge control devices simulation

Common surge control devices have been incorporated into the present numerical model. These include valve, pump, air chamber and surge tank devices. In the present analysis, only one single device located at the upstream end of the pipe system is considered.

Though not considered in the present research, the model consists of the following functions related to the devices:

- i) Multiple devices can be combined for transient protection;
- ii) Devices can be located anywhere along the line; and
- iii) Vapor pressure, network hydraulics and local losses can also be simulated in this model.

4.2.3.5 Special features and formats

The present model contains the normal input and output forms as found in general practice. Furthermore, it possesses some features and formats specially established for this research. These include:

- i) non-dimensional input and output formats,
- ii) transient energy by integrating the pressure along the line, and
- iii) the balance characteristics of inflow and outflow to the pipe system, reflecting the real physical phenomenon of the pipe elasticity.

Following the development of the numerical model some physical model tests were carried out for its validation. This work is included in the next subsection.

4.2.4 Establishment of physical model data

As indicated in part (b) of Table 4.1, a large quantity of physical data and findings obtained from available published (e.g. papers and thesis) and unpublished (notebooks and reports) sources have been used in this research for the validation and interpretation of present numerical modeling approach. These are described below.

4.2.4.1 Physical tests used for numerical model validation

The theoretical background and application of the numerical model has been verified by numerous authors and engineers for various surge control measures under many different hydraulic conditions. These include field works conducted by Wylie and Streeter (1978), Parmakian (1953), Chaudhry (1979) and Stephenson (1981 and 1984).

In addition, a considerable amount of field testing has been performed by the writer, prior to this research. These include the on-site testing of pump trip-out and valve closure (Proctor and Redfern Ltd, 1979), surge tank installation (Maclaren Ltd, 1976), and air chamber operational control (Eok et al, 1978). The experience and expertise gained from these physical verifications are made use of in the present research, for the effective application of the numerical model.

4.2.4.2 Physical tests used for analyzing friction losses

Previous physical modeling work related to the present analysis of unsteady friction (f_u) includes:

- i) Uniform closure in turbulent uni-directional flow (Daily et al 1956 and Carstens and Roller 1959);
- ii) Non-uniform opening in turbulent uni-directional flow (Denisov 1972 and Popov 1974);
- iii) Damping in laminar bi-directional flow (Zielke, 1966); and
- iv) Damping in turbulent bi-directional flow (Sauer, 1969).

Physical model tests related to other frictional systems are not required, since the well validated numerical model can accurately predict the transient behavior in the case where the steady or the quasi-steady friction factor prevail.

With the numerical model developed and verified by the physical tests, the issue is how to best interpret the transient characteristics, which are simulated from the numerical model, and supplemented by the available physical model tests. The next subsection is intended to shed some insights on alternative interpretation methods.

4.2.5 Choice of interpreting method

During the course of this research, it has been found that some conflicts existed and some problems remain unsolved. Two of the most important among these are:

i) Frictional effect --- As concluded from the literature review, conflicting and opposite results about the existence, of friction (H_f) and its unsteadiness (f_u) on transient simulations have been frequently reported. Consequently, the fundamental question regarding the necessity of an unsteady friction model remains debatable.

The prolonged existence of these differences caused the writer to pause, and then review the overall analytical approach of which the interpretation method is part prior to further advancement of his research in this area.

ii) Unsteadiness of friction --- The dual modeling approach can identify anomalous losses unaccounted in the numerical model, a question still remaining is whether these unaccounted losses result from the effect of the unsteady friction factor (f_u), or from other sources unrelated to it. This problem has been previously addressed in the section of dimensional reasoning.

As is well known, apart from friction energy losses also exist wherever and whenever the flow velocity changes with location and time. The method of

characteristics (Wylie and Streeter 1978) has not included this energy loss phenomenon. In order to assess the potential impact due to the neglect of this energy loss, all the flow movement with respect to time and location has to be investigated which is a prohibitive task. An alternative interpretation method is required.

These two unsolved questions generate some serious concerns as to whether the traditional line energy (pressure) response method of interpretation is adequate. An alternate method is sought in this research, as discussed below.

Fluid mass balance --- In a pipe system, elastic behavior originates from the compression or expansion of the pipe system. This also implies that the elastic energy existing in the system can suitably be measured by the balanced characteristics of fluid mass influx and outflux within the pipe system.

This mass balance characteristic is adopted in the present research as an alternative interpretation method, in an attempt to more effectively analyze the physical phenomenon in a pipe system.

4.2.6 Discussion of model simulation

A numerical model, Otthammer, (Fok, 1987) has been developed in this thesis. The model adopts the well established method of characteristics as its theoretical background. In addition, it has been specially designed for the purpose of both scientific research and practical applications.

The works from previous physical model tests can be used for the validation of the present numerical model, as well as for the subsequent transient analysis related to unsteady friction losses.

Its application will be carried out later in this thesis, for a wide scope of transient flow simulations including

non-frictional, frictional and unsteady friction (factor) systems.

The concept of mass balance in the system is suggested for transient interpretation as an alternative to the current pressure response method.

4.3 Development of Hypothetical Flow Closure

4.3.1 General characteristics of flow closure

During the flow closing at the upstream end of the system, the common unsteady flow characteristics of surge control measures can be reflected from:

i) Flow direction --- This can be one-way or two-way flow movement at the source of the system. The research in this thesis focuses on the analysis of one-way flow motion for reasons of simplicity and consistency.

However, impacts of the two-way flow system on the transient damping have been analyzed in some detail and results are reported in the corresponding design nomographs, as highlighted in Appendix C.

ii) Time of flow closure (T_d) --- This parameter, T_d , represents the slow or fast flow closure for various devices.

It should be noted that the time of flow closure of common surge control devices cannot be pre-determined during steady state conditions, except for the case of valve operation.

iii) Shape of flow closure (S) --- The shape of flow closure at the upstream end of the system can rarely be pre-determined. It can be a continuous or a step-wise flow variation with uniform, increasing or decreasing deceleration. In this research, only the shape of

continuous flow response is considered as it exists in the majority of transient conditions.

The step-wise flow closure can be found in some rapid pump trip-out conditions which would generally require other surge protections in addition to its own kinetic energy. Furthermore, the relatively large quantity of elastic energy induced by this step-wise flow closure reduces the significance of the frictional effect in the pipe system.

Depending on the steady state hydraulic conditions, and on the type and size of surge protective devices installed, some forms of common characteristics of flow variations at the upstream end of the system can be established related to the time and the shape of flow closure. It should be noted that the same flow variation characteristics at the source initiates identical transient effects over the whole piping system.

The common nature of flow variations and inherent similarities of transient responses for these devices are made use of in the present work. Prior to the analysis of each individual control measure, the impacts of transients should be first evaluated from the categorized unsteady flow simulations, which represent the general characteristics of these four common devices. The transient analysis of flow closure simulations within the scope of this study is therefore in principle applicable in a wider range of hydraulic conditions for various surge protection devices than would otherwise be the case.

The present systematic discretization of unsteady flows at source to represent various surge control measures in general transient conditions is a unique method in current research practice.

4.3.2 Classification of flow variation at source

Governing conditions for flow closure can be divided into four groups representing commonly observed physical behavior, based on progressive changes in flow velocity (as shown in Fig. 4.1):

- i) Increasing deceleration;
- ii) decreasing deceleration;
- iii) uniform deceleration; or
- iv) instantaneous closure.

4.3.2.1 Increasing deceleration of flow closure

The flow pattern with increasing deceleration can be expressed in a dimensionless form, such as:

$$Q = 1 - (T/T_a)^S \quad 4.26$$

where, $Q = q/q_0$, the portion of the steady flow rate;

$T = t/t_c$, in terms of wave travel time (t_c);

$T_a = t_a/t_c$, time required to stop flow movement; and

S = shape of flow closure, which is greater than 1.

The absolute value of deceleration ($|dQ/dT| = S * T^{S-1}$) is increasing with time (T) as the exponent of time ($S-1$), is positive for $S > 1$.

4.3.2.2 Decreasing deceleration of flow closure

Flow with decreasing deceleration is most conveniently restructured as:

$$Q = (1 - T/T_a)^{-S} \quad 4.27$$

The shape factor (S) should be less than -1 when applying this equation. The use of these two different forms in flow

closures (Eq. 4.26 and Eq. 4.27) has the following advantages:

i) The flow closure characteristic, namely $Q(T)$ of the decreasing deceleration case in Eq. 4.27 is symmetrical with respect to the increasing deceleration in Eq. 4.26, about the diagonal line (uniform closure in Fig. 4.1). For example, at the time $T = 0.75$, the flows at source are $Q(S=2) = 0.4375$ and $Q(S=-2) = 0.0625$ which are symmetrical about $Q(S=1) = 0.25$.

ii) As a consequence of the symmetry properties of the flow closing functions, the transient behavior induced from both types of deceleration (possessing the same absolute value of shape factor $+S$ vs $-S$) can directly be used for the comparison of their elastic and frictional effects on the piping system. This will be demonstrated in later chapters of this thesis.

iii) The functions related to the percentage of area closing of all common valve operations (a total of seven) can be represented by either one of the flow deceleration equations. For example, Eq. 4.26 with $S = 2$ corresponds with the needle valve closing, while Eq. 4.27 with $S = -1.85$ represents butterfly valve operation. Actually, when the valve head (H_0) is sufficiently high, the flow passing through the valve follows exactly the same pattern of a specified shape factor (S) during the whole flow closing period.

It should be noted that different shapes other than the expressions of Eqs. 4.26 and 4.27 are possible in general practice and these can be conveniently incorporated into the present numerical model (Otthammer, 1987). The relations expressed in Eqs. 4.26 and 4.27 are considered adequate to represent the general flow closing characteristics, and hence adopted in this research.

4.3.2.3 Uniform deceleration of a flow closure

The case of uniform closure can be obtained when the shape factor is unity (i.e. $S = +1.0$ or -1.0 from Eq. 4.26 and 4.27 respectively).

4.3.2.4 Instantaneous flow closure

When the shape (factor) of flow closure approaches either plus or minus infinity, it becomes an instantaneous flow closure, independent of the time of flow closure (T_a).

4.3.3 Range of flow closing parameters

The ranges of the two flow closing parameters (S and T_a) commonly adopted in general practice were also adopted in this research.

4.3.3.1 Range of Shape factor (S)

Under normal flow closing conditions, the value of the shape factor seldom extends beyond the range of $-10 < S < 10$. In the case of the shape factor exceeding this range, the general characteristics of the transient can be interpolated from the analysis of results between the instantaneous flow closure ($S \Rightarrow +\infty$ or $-\infty$) and the boundary condition ($S = +10$ or -10). This range of the flow shape factor ($-10 < S < 10$) will be adopted for the evaluation of maximum transient events in this research.

4.3.3.2 Time of flow closure (T_a)

Time of flow closure (T_a) is normally used to reflect a fast or a slow unsteady flow movement during the flow closing period. The time of flow closure adopted in the present research varies from an instantaneous closure ($T_a = 0$) to a relatively slow flow movement ($T_a = 100$). The transient energy induced by flow closures with $T_a > 100$ is usually so small that they are beyond the means of practical applications.

4.3.4 Discussion of hypothetical flows

Categorized unsteady flows at the source of the system can be used to represent various surge control device simulations in general transient flows. The progressive flow closure at the source can be conveniently represented as increasing and decreasing deceleration as shown in Eqs. 4.26 and 4.27.

The investigation of these systematical flow variations in this thesis provides a better understanding of transient flow phenomena, meanwhile it also leads to a more efficient and accurate solution of transient analysis for various protective measures.

4.4 Conclusions

4.4.1 Analytical approach

The analytical approach adopted in this thesis combines the following methodologies, as shown in Part (a) of Fig. 4.2:

1) Dimensional reasoning --- Dimensional analysis has been employed to identify the relevant parameters governing transient phenomena under various hydraulic conditions. The functional relations (Eqs. 4.4 - 4.7) thus established are used as a guide for research direction.

2) Numerical modeling --- A numerical model (Otthammer, 1987) has been set up to carry out a comprehensive analysis of transients. Based on method of characteristics, this model minimizes numerical error. The development of this numerical model in this thesis fulfills two purposes:

i) to be specially designed for the present research on energy losses in transient flow conditions, and

ii) to be applicable to general engineering practice for those common surge control measures under a range of hydraulic conditions.

The present analytical approach adopts the numerical model simulation for transient analysis, and is supplemented by the physical modeling test where it is required.

3) Physical modeling --- Findings and data obtained from available laboratory works have been used to verify the numerical model. These are also used presently to identify the existence of other potential energy losses, which may not be properly accounted by a numerical model simulation.

This latter procedure, however, is only carried out to supplement the numerical modeling analysis in the unsteady friction system, after the effect of quasi-steady friction factor (f_q) has been investigated.

4) Measurement of pressure --- The current technique of data gathering (i.e. the measurements of instantaneous pressures at the source of surge initiation) is not an efficient means of evaluating hydraulic pressure responses in the pipe system. Instead, the integration of the instantaneous pressure envelope along the whole pipeline is more efficient to represent the pressure response, and hence it is adopted in this research.

5) Energy balance --- The concept of mass balance is adopted in this research to interpret the transient phenomena, as an alternative method of the traditional pressure response. The effectiveness of this new interpretation approach will be demonstrated in this thesis.

6) Unsteady flow characteristics --- Prior to the analysis of each individual surge control device, the unsteady flow characteristics at the source of surge initiation can be categorized into the shape (S) and the time (T_q) of flow variation. Together the transient analyses of these

categorized flows represent the impact of common surge protective device simulations in the piping system.

4.4.2 Present research

The present research incorporates the above analytical methodologies into the following four hydraulic systems as shown in part (b) of Fig. 4.2:

- 1) Non-frictional properties (Ch. 5) --- The present analysis begins with the most elementary hydraulic system, which is a non-frictional pipeline under categorized flow patterns. This initial stage of analysis provides a comprehensive review and evaluation of the elastic properties of the hydraulic system, in which the surge wave is the dominant parameter, in the absence of friction.
- 2) Steady friction (factor) properties (Ch. 6) --- The research then proceeds to explore the effect of the constant friction (factor) loss on surge analysis for uni-directional and bi-directional flow systems. In this second stage, the transient properties in constant friction (factor) systems have been carefully examined in conjunction with the elasticity in the pipe system for various specified flow closure patterns.
- 3) Quasi-steady friction (factor) properties (Ch. 7) --- The effect of a quasi-steady friction factor (f_u) application can be evaluated for different unsteady flow motions, flow regimes and different critical transient conditions. The results of this analysis provide some insights into the effect of an elementary system related to the unsteady friction factor.
- 4) Unsteady friction (factor) properties (Ch. 8) --- The effect of the unsteady friction factor (f_u) cannot be realistically analyzed without the data and findings provided from available physical model tests, to supplement the numerical modeling analysis.

5) Surge control devices --- Once the physical phenomena governing the unsteady flows have been well defined and analyzed, transient behavior in pipelines with various protective devices are simulated corresponding to various frictional conditions. In this thesis only those analytical results of common surge control device, relevant to the clarity or to the practical illustrations, are included.

Highlights of parallel research documenting design tools for engineering practice have been appended in Appendix C.

4.4.3 Future research

6) Future works relevant to both scientific research and practical application need to be undertaken in the near future. These include: i) the effect of a vapor pocket and its collapsing, ii) local energy losses affecting transient simulation, iii) hydraulic responses in a network system, and iv) external effect from soil backfilling and pipe fixation.

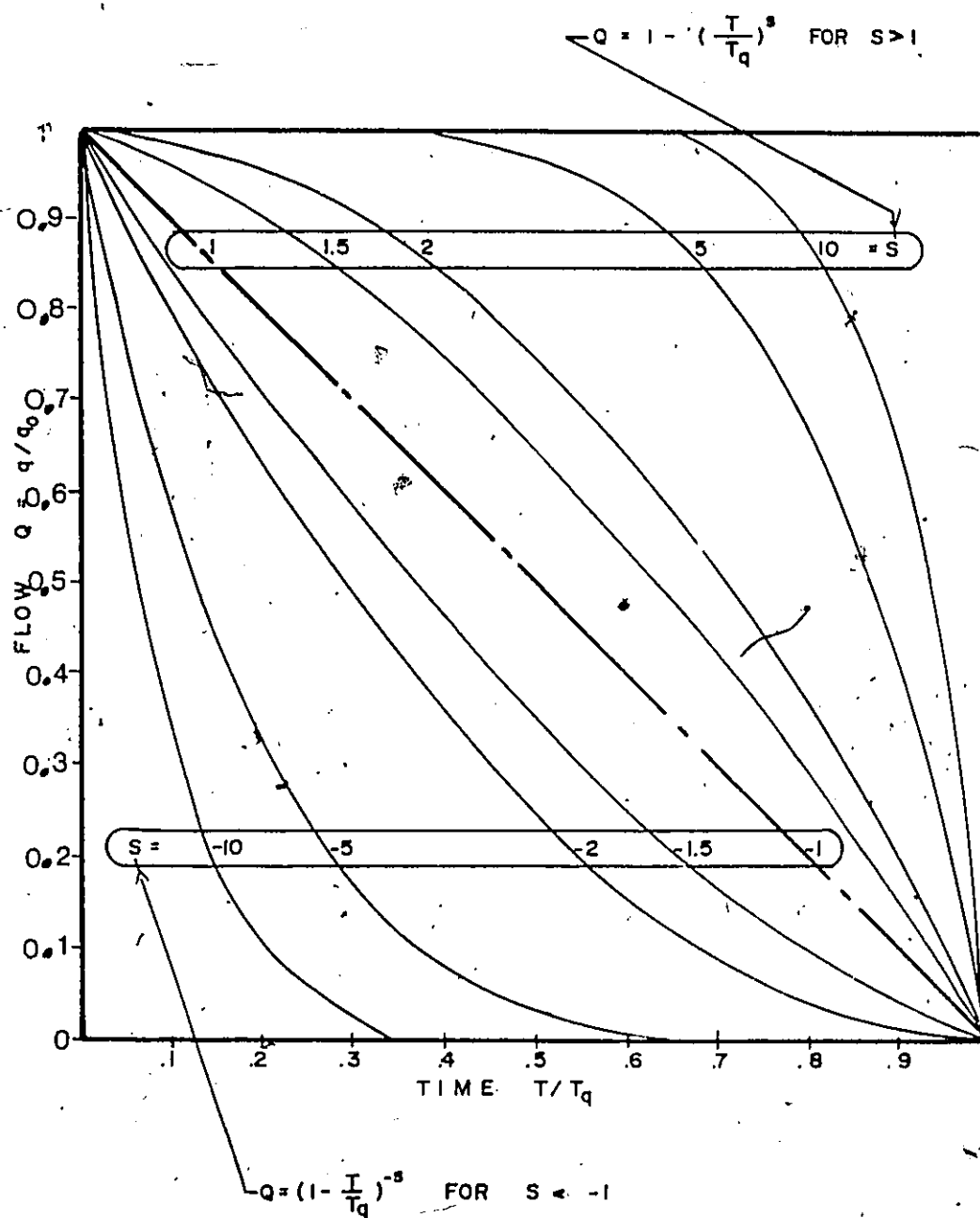
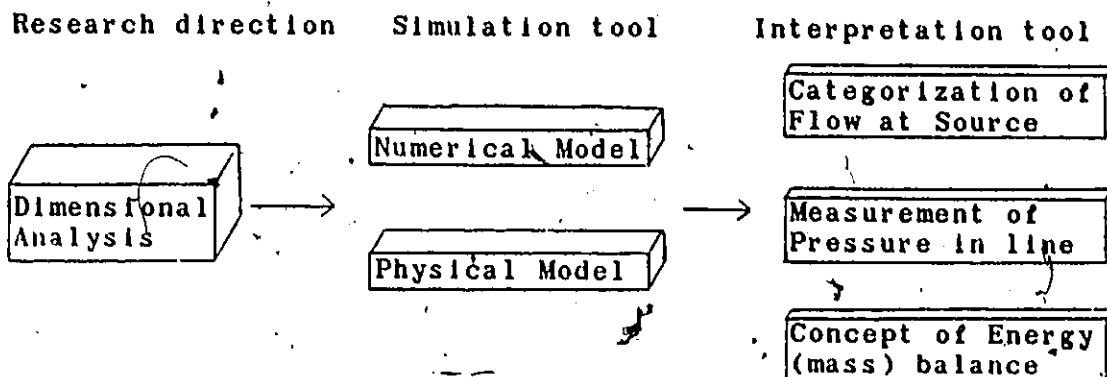


Fig. 4.1 — Characteristics of Flow Closure.

A) ANALYTICAL APPROACH



B) PRESENT RESEARCH

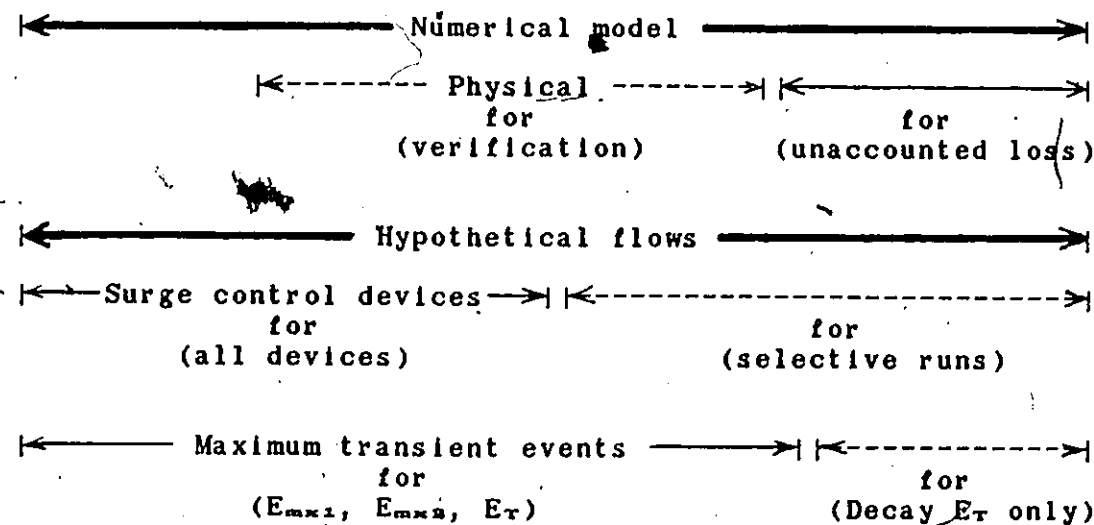
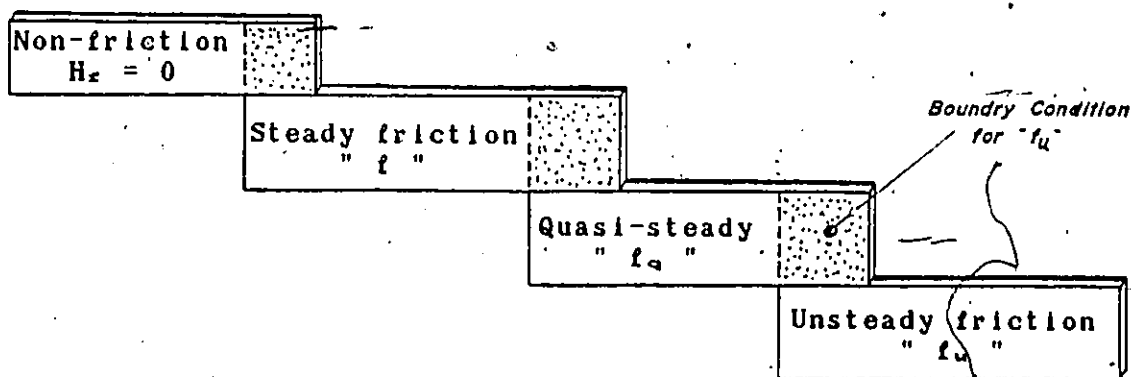


Fig. 4.2 ANALYTICAL APPROACH AND PRESENT RESEARCH

5.0 ANALYSIS OF TRANSIENTS IN A NON-FRICTIONAL SYSTEM

In the previous dimensional analysis a set of functional equations (Eqs. 4.4 & 4.7) were derived for various hydraulic conditions. A numerical model approach was established to verify and subsequently quantify the relevance of each non-dimensional parameter.

One concept introduced in this research is related to the balance of flow energy during the transient period. This chapter defines this energy balance concept and shows how it can be incorporated with the traditional integrated pressure response method for an effective transient analysis in non-frictional systems.

5.1 Scope of Analysis

The transient analysis in this chapter focuses on the categorized unsteady flow closing characteristics which are governed by the shape (S) and time (T_c) of flow closures.

5.1.1 Illustrations of the mass balance concept

Transient histories of typical flow closure are selected for discussion. These include:

Shape factor ($\bar{S}$)	Types of flow <u>represented</u>	<u>Shown in Figure</u>
∞	Instantaneous closure	5.1
1	Uniform deceleration	5.2
-2	Decreasing deceleration	5.3
2	Increasing deceleration	5.4

The magnitude and time of maximum transient energy in these four figures have been summarized in Table 5.1. In addition

to these the flow movement at the source and at the pipe-end with $S = 1.5$ and 5 are plotted for analyzing the effects of surge fluctuations (Fig. 5.5).

Three events of flow closure, $T_a = 2, 4$ and 6 , are investigated for each shape of decelerating flows (Figs. 5.2 - 5.5).

5.1.2 Types of transient response

A comprehensive analysis is also conducted by integrating maximum pressure envelopes along the pipe system to determine the maximum surge (E_{mx1}) and counter-surge (E_{mx2}) energies. This covers a wide range of flow shape factor, $-10 < S < +10$, and time of flow closure, $0 < T_a < 100$.

The proposed investigation of the detailed historical events ($E_m(T)$ or $Q(T)$ in Sub. 5.1.1) and the wide-ranged maximum energy (E_{mx} in Sub. 5.1.2) should adequately explain relevant transient phenomena in a non-frictional system.

5.2 Energy Relations in Transient Flow

This section introduces and defines the new concept of energy balance in transient flow conditions.

5.2.1 Internal energy in a pipe system

For an energy budget consideration, the control volume defined for a closed conduit system covers the whole pipeline, excluding only the points of source and pipe-end (as shown in Fig. 5.1a). In the absence of friction, the internal energy (E) remains constant in this non-frictional system during the steady or transient condition. This means it is possible to define the internal energy relative to some suitable datum as:

$$E(T) = 0$$

5.1

where T is the time during the transient.

The datum defined in this research is the water level at pipe-end plus the initial velocity head (H_v) . Since this head is usually very small compared with the transient pressure (H_m) , it is ignored in the numerical or graphical presentations throughout this thesis.

It is possible to redefine the control volume and the datum of the pipe system in many other ways. However, it is felt that the present definition is most convenient for this analysis.

5.2.2 Steady-state conditions in the pipe

Under steady-state conditions, the rate of flow passing through the pipe-end (q_x) and the rate of flow being supplied by the upstream source (q_s) are the same. Hence, mass balance in the pipe system is retained as:

$$\int_0^t q_s(t) dT - \int_0^t q_x(t) dT = 0 \quad 5.2$$

where $q_x(t) = q_s(t) = q_0$.

The balanced characteristics of flow mass in Eq. 5.2 is consistent with the inelastic behavior of steady flow motion.

5.2.3 Instantaneous flow closure in the pipe

5.2.3.1 Flow energy for instantaneous closure

1) Definition ---

With an instantaneous and complete cut-off of the source flow (i.e. $q_s(t > 0) = 0$), the pipe-end flow $(q_x(t))$ is still the same as the steady flow (q_0) , which reflects the imbalance characteristic of the flow energy. In this instantaneous closure condition, the netflow $q(t) = q_x(t)$.

At time $t_1 < t_c = 1/a$, the surge wave has not yet reached the pipe-end so that the mass imbalance can be expressed as:

$$\rho \int_0^{t_1} q(t) dt = \rho * q_0 * t_1 \quad 5.3$$

where $q(t) = q_x(t) = q_0$ and $t_1 < t_c$.

As a result of pipe wall and water expansion, an energy netflux (e_w) stored in the pipe system at $t = t_1$ is:

$$e_w(0 < t < t_1) = K \int_0^{t_1} q(t) dt = K * (q_0 * t_1) \quad 5.4$$

The imbalance flow energy characteristics indicated by this equation shows the existence of elasticity in the pipe system.

The energy netflux can be expressed in a dimensionless form as:

$$\begin{aligned} \{ E_w(T_1) &= E_{wx}(T_1) - E_{ws}(T_1) \\ \{ & \\ \{ & \\ \{ &= \int_0^{T_1} Q_x(T) dT - \int_0^{T_1} Q_s(T) dT \\ \{ & \\ \{ & \\ \{ & \\ \{ &= \int_0^{T_1} Q(T) dT \\ \{ & \end{aligned} \quad 5.5$$

where:

$e_w(t_c) = K * q_0 * t_c$, the unit energy quantity;

$T_1 = t_1/t_c$, the time; and

$Q(T) = q/q_0$, the transient flow.

$E_{wx}(T) = \text{outflux, reflecting the outflow } Q_x(T)$

$E_{ws}(T) = \text{influx, reflecting the inflow } Q_s(T).$

For simplicity, only one boundary condition (e.g. T_1) is written inside the bracket of the energy term, e.g. $E_w(T_1)$, indicating that the time domain begins from the time of surge initiation, i.e. $E_w(0 < T < T_1)$.

2) Illustration ---

An example of non-frictional system is selected below to evaluate the physical behavior of transient flow, prior to the introduction of energy balance concept.

- 1) Given: $l = 1000$ m for pipe length;
 $A = 1$ m² for cross-sectional area;
 $\rho = 1000$ kg/m³ for water density;
 $\gamma = 10^4$ Newton/m³ for water specific weight;
 $q_o = 1$ m³/s for flow rate;
 $v_o = 1$ m/s for flow velocity;
 $a = 1000$ m/s for surge wave velocity;
 $K = \rho * a^2 = 10^9$ kg/m-s² for the combined water and pipe wall bulk modulus;
 $g = 10$ m/s for gravitational acceleration;
 $t_c = 1$ sec. for a wave travel time.

11) Mass and energy imbalance --- At time the surge wave arrives the pipe-end (i.e. $t = t_c$), the mass imbalance reaches its maximum. Based on Eq. 5.3, the water mass in the pipe is 1000 kg less than the initial steady state at $t = 1$ sec. This is estimated as:

$$o * q_o * t_c = 1000 \text{ kg/m}^3 * 1 \text{ m}^3/\text{s} * 1 \text{ s} = 1000 \text{ kg}$$

where $q_o * t_c = 1$ m³ is the equivalent expanded volume.

Then the netflux at $t = t_c = 1$ second is:

$$\begin{aligned} e_w(t_c) &= 1 \text{ m}^3 * 10^9 \text{ kg/m-s}^2 \\ &= 10^9 \text{ kg-m/s}^2\text{-m} \\ &= 10^9 \text{ Joules} \end{aligned}$$

5.2.3.2 Elastic energy for instantaneous closure

1) Definition ---

Following an instantaneous flow closure, a pressure head equivalent to Joukowski head ($h_c = av/g$) is first induced at the upstream source.

The high pressure front then progresses toward the downstream end of the pipe system. At time $t = t_1 < t_c$, the pipe length sustaining the pressure of $h_m(x, t) = -h_c$ is that $0 < x < x_1$. This relation can be expressed as below:

$$\begin{cases} h_m(x < x_1) = -h_c, \text{ and} \\ \text{---} \\ h_m(x > x_1) = 0 \end{cases} \quad 5.6$$

where $x_1 < 1$.

Consequently, the transient (elastic) energy of the system is expressed as:

$$\begin{cases} e_m(t_1) = r * A * \int_0^1 h_m(x, t_1) dx \\ \text{---} \\ = r * A * \int_0^{x_1} h_m(x, t_1) dx + r * A * \int_{x_1}^1 h_m(x, t_1) dx \end{cases} \quad 5.7$$

The pressure $h_m(x, t_1)$ is governed by Eq. 5.6 and the constraint that $t_1 < t_c$ for the application of the above equation.

The transient (elastic) energy can be rewritten in a non-dimensional form as:

$$E_m(T_1) = \int_0^1 H_m(X, T_1) dX \quad 5.8$$

where the unit energy quantity, $e_m(t_c) = \gamma \cdot h_c \cdot l$ is represented by the rectangular pressure enclosed in the entire system (Fig. 5.1).

2) Illustration ---

i) Given: The same conditions of the previous pipe system are held in this example.

ii) Energy : At $t = t_c = 1$ sec, the whole pipe line is under a negative pressure of:

$$\begin{aligned} h_m(t_c) &= -av/g = - (1000 \text{ m/s} \cdot 1 \text{ m/s}) / 10 \text{ m/s}^2 \\ &= - 100 \text{ m} \end{aligned}$$

Accordingly, the transient (elastic) energy in the system can be estimated as:

$$\begin{aligned} e_m(t_c) &= r \cdot h_c \cdot A \cdot l \\ &= - 10^6 \text{ N/m}^2 \cdot 100 \text{ m} \cdot 1 \text{ m}^2 \cdot 1000 \text{ m} \\ &= - 10^9 \text{ Joules} \end{aligned}$$

where $r \cdot h_c = 10^6 \text{ N/m}^2$ is the pressure in line; and $A \cdot l = 1000 \text{ m}^3$ is the pipe capacity.

5.2.3.3 Energy balance during the surge period

Following the traditional sign convention, the surge energy is considered to be negative, while the magnitude of netflux is termed positive during the (first) surge period, which is defined for $0 < T < T_{mx1}$. For a non-frictional system with an instantaneous flow closure, $T_{mx1} = 1$.

Based on the Law of Energy Conservation, the internal energy inside the control volume should remain constant at all times if no energy dissipation occurs. The energy relation in a non-frictional system can then be combined from Eqs. 5.6 and 5.7 as follows:

$$\begin{aligned}
 \{ E(T_1) &= E_w(T_1) + E_m(T_1) \\
 \{ & \\
 \{ & \\
 \{ & = \int_0^{T_1} Q(T) dT + \int_0^1 H_m(X, T_1) dX \\
 \{ & \\
 \{ & = 0
 \end{aligned}
 \tag{5.9}$$

where $Q(T) = Q_0 = 1$ and $T_1 < 1$.

The above relationship has also been confirmed from the previous two illustrations. In a non-frictional system, Eq. 5.9 also implies that:

$$\begin{aligned}
 \{ E_m(T_1) &= -E_w(T_1), \text{ or} \\
 \{ & \\
 \{ \int_0^1 & H_m(X, T_1) dX = - \int_0^{T_1} Q(T) dT \\
 \{ &
 \end{aligned}
 \tag{5.9a}$$

The transient energy (E_m) of Eq. 5.9a is related to the instantaneous characteristics (at $T=T_1$) of pressure response along the whole line ($0 < X < 1$), and can be directly determined from the pressure envelope.

On the other hand, the flow energy (E_w) reflects the elastic properties of the pipe system by the imbalance characteristics of the water mass. This can be estimated by tracing the flow movement backward in time (i.e. $0 < T < T_1$).

The relationship (Eq. 5.9a) between these two energy terms forms the background of the concept of energy balance for an instantaneous flow closure during the surge period.

The maximum surge energy and the maximum energy outflux can then be recognized as:

$$\begin{aligned}
 \{ \quad E_{mx1} = E_m(T_{mx1}) &= \int_0^1 H_m(X, T_{mx1}) dX = -1; \text{ and} \\
 \{ \\
 \{ \\
 \{ \\
 \{ \quad E_{mx1} = E_w(T_{mx1}) &= \int_0^{T_{mx1}} Q(T) dT = 1 \\
 \{
 \end{aligned}
 \tag{5.10}$$

where $T_{mx1} = 1$ and $Q(T) = Q_0 = 1$ for $H_x = 0$ and $T_Q = 0$.

5.2.3.4 Energy balance during counter-surge period

Transient pressure and flow variations during the counter-surge period behave in a similar manner as those during the surge period. These transient properties are not discussed in detail here but can be referred to in many text books (Wylie and Streeter 1978, Parmakian 1963, Stephenson 1983). However, using the same logic the relations developed during the first surge period (Eqs. 5.5 - 5.10) should also be applied to the subsequent counter-surge period, which is defined as $T_{mx1} < T < T_{mx2}$. The signs of the terms in Eqs. 5.9 and 5.10 should then be reversed for the counter-surge condition.

Accordingly, for an instantaneous flow closure the maximum counter-surge energy and the maximum magnitude of (negative) energy outflux are:

$$\begin{aligned}
 \{ \quad E_{mx2} = E_m(T_{mx2}) &= \int_0^1 H_m(X, T_{mx2}) dX = 1; \text{ and} \\
 \{ \\
 \{ \\
 \{ \\
 \{ \quad E_{mx2} = E_w(T_{mx2}) &= \int_0^{T_{mx2}} Q(T) dT = -1 \\
 \{
 \end{aligned}
 \tag{5.11}$$

where $T_{mx2} = 3$, $Q(1) = 1$ and $Q(1 < T < 3) = -1$, for $H_x = 0$ and $T_Q = 0$.

For simplicity, the comparison of transient properties at different times (e.g. E_{m1} vs. E_{m2}) are referenced to their absolute magnitudes in this development.

5.2.4. Normal flow closure in the pipe

With respect to a normal flow closure the energy budget inside the control volume of the system at time $T_1 > 0$ can be expressed as:

$$\begin{aligned} E(T_1) &= E_m(T_1) + E_w(T_1) \\ &= E_m(T_1) + [E_{wE}(T_1) - E_{wS}(T_1)] \\ &= 0 \end{aligned} \quad 5.12$$

where the netflux $E_w(T) = \text{outflux } E_{wE}(T) - \text{influx } E_{wS}(T)$.

For a normal flow closure the elastic energy (E_m) at T_1 is:

$$E_m(T_1) = \int_0^1 H_m(X, T_1) dX \quad 5.13$$

where the transient pressure $H_m(X, T_1)$ varies continuously with the distance (X) along the pipeline. This is different from the step-wise pressure front moving towards the pipe-end since this also exists in an instantaneous flow closure (Eq. 5.6).

The energy influx (E_{wS}) and outflux (E_{wE}) by Eq. 5.12 can then be expressed as:

$$\begin{aligned} \{ & E_{wS}(T_1) = \int_0^{T_1} Q_S(T) dT; \text{ and} \\ \{ & E_{wE}(T_1) = \int_0^{T_1} Q_E(T) dT \end{aligned} \quad 5.14$$

where the source inflow $Q_s(T)$ and the pipe-end outflow $Q_e(T)$ are not the same, and the netflow ($Q_e - Q_s$) varies with time (T) in a progressive manner during the transient period. This is different from an abrupt flow reduction in an instantaneous flow closure.

Rearranging Eq. 5.12, the instantaneous elastic energy (E_m) is equal to the energy netflux across the control surface boundaries. This implies that:

$$\begin{aligned} \{ E_m(T_1) &= - \{ E_{we}(T_1) - E_{ws}(T_1) \} \\ \{ & \\ \{ &= \int_0^{T_1} Q_s(T) dT - \int_0^{T_1} Q_e(T) dT \end{aligned} \quad 5.15$$

where T_1 is measured from the time of surge initiation.

5.2.5 Discussion of the concept of energy balance

The relationship expressed in Eq. 5.15 can thus be used to integrate the past transient events into the present transient condition. This relationship in the equation forms the basis of the present *Energy Balance Concept* for a normal flow closing condition.

This alternative tool for interpretation has been used extensively to tackle the transient problems throughout this research. To begin with, the concept is adopted in this chapter to treat one of the fundamental mechanisms of transient pipe flow, namely the mechanical energy transfer in the absence of frictional energy losses.

5.3 Periodic Fluctuation in a Normal Flow Closure

5.3.1 General trends of periodicity

A continuous uni-directional source flow (Q_s) has been adopted in this research, in which the 'periodicity' condition does not exist at the source during the normal

flow closing. However, the non-existence of periodicity does not exist in the rest of the pipeline. Instead, two trends of periodic fluctuation exist:

i) Location --- The magnitude of the fluctuations increase toward the downstream end of the pipe. In the present application of the Energy Balance Concept, only the source (Q_s) and the pipe-end (Q_e) flow are presented, reflecting the two most extreme conditions for periodic fluctuations.

ii) Time --- The accumulated effect of past wave reflection may be either constructive (e.g. $S = -2$, as shown in Fig. 5.3a) or destructive (e.g. $S = 2$, as shown in Fig. 5.4a). This will increase the magnitude of the periodic fluctuations, during the source flow closing period (i.e. $T < T_a$). These two distinct transient properties require separate analyses for the different types of flow closure.

On the other hand, periodic surge fluctuations dominate the transient characteristics after the closure of source flow for any type of flow closure (i.e. $T > T_a$).

In any event, the flows in the pipe system are unsteady and spatially varied. Discussions in this section first focus on the periodicity for each type of typical uni-directional flow (i.e. $T < T_a$). Three types of uni-directional flow have been selected for the analysis of periodic fluctuations.

The effect on the bi-directional transient flows (i.e. $T > T_a$) will then be investigated subsequently.

5.3.2 Uniform deceleration in the flow closure

5.3.2.1 Maximum surge energy for flow with $S = 1$

As shown in Fig. 5.2c, maximum surge energy (E_{mx1}) always occurs at $T_{mx1} = 2$, disregarding the time of flow closure. For example, for the present case of $S = 1$ and $H_s = 0$,

$$i) \quad E_{mx1}(T_q=2) = E_m(T=2) = -0.5;$$

$$ii) \quad E_{mx1}(T_q=4) = E_m(T=2) = -0.25; \text{ and}$$

$$iii) \quad E_{mx1}(T_q=6) = E_m(T=2) = -0.166.$$

5.3.2.2 Periodicity for flow with $S = 1$

As shown in Fig. 5.2a, the pipe-end flow (Q_p) always intercepts the source flow (Q_s) at $T_{ps} = 2$. The term T_{ps} is defined in this research as the time of flow interception.

For $T_q > 2$, the two flows re-intercept with exactly the same hydraulic conditions at an interval equal to the period of surge fluctuation (T_s). This is here defined as four times the surge wave travel along the pipeline. Therefore:

$$T_s = 4 * T_c = 4 \quad 5.16$$

where the wave travel time $T_c = 1$, as previously defined.

For example, for the case of $T_q = 6$, $S = 1$ and $H_s = 0$, the periodic fluctuations are characterized by an alternate fluctuation with a constant absolute value:

$$E_w(2) = -E_w(2 < T < 4) = E_w(4 < T < 6) = 0.166$$

Within the period of fluctuation (T_s) the effect of the periodic fluctuations is negligible, as shown below:

$$E_w(2 < T < 6) = E_w(2 < T < 4) + E_w(4 < T < 6) = 0$$

The maximum surge energy therefore remains unaffected:

$$E_{mx1} = E_m(2) = E_m(6) = -0.166$$

This reoccurrence of same periodic surge fluctuations can be expressed in the form of functional relation as:

$$\text{for } T_{mx1} < T_Q < N \cdot T_S + T_{mx1},$$

$$E_{mx1} = E_m(T_{mx1}) = E_m(T_{mx1} + N \cdot T_S) \quad 5.17$$

where N is a positive integer.

The periodic fluctuations with the same intensity suggests the dominance of periodicity in a non-frictional uniform flow closure.

5.3.2.3 Non-friction transfer at $T < T_{mx1}$

The term non-friction transfer (ET_x) is defined in this thesis as the flow energy summed up after the time of first flow interception (i.e. $T > T_{xs}$). It is used to evaluate the effect of periodic fluctuations during the flow closing period. The non-friction transfer can be expressed as below:

$$\begin{aligned} \{ ET_x &= E_{ws}(T_{xs} < T < T_{x1}) - E_{ws}(T_{xs} < T < T_Q) \\ \{ & \\ \{ &= \int_{T_{xs}}^{T_{x1}} Q_x(T) dT - \int_{T_{xs}}^{T_Q} Q_s(T) dT \\ \{ & \end{aligned} \quad 5.18$$

where the time of pipe-end flow closure (T_{x1}) may or may not equal to the time of flow closure (T_Q). This is shown in the next illustrative example.

For the case of $S = 1$ and $H_x = 0$, the characteristics related to non-friction transfer are summarized below and are also shown in Fig. 5.2a:

- i) for $T_Q = 2$, $ET_m = 0$ and $T_{m1} = 2 = T_Q$;
- ii) for $T_Q = 4$, $ET_m = -0.25$ and $T_{m1} = 3 < T_Q$; and
- iii) for $T_Q = 6$, $ET_m = 0$ and $T_{m1} = 6 = T_Q$.

It should be noted that the zero or negative non-friction transfer (ET_m) does not increase the maximum surge energy as:

$$E_{m1} = E_m(T_{m1}) = E_m(T_{ms}).$$

The term E_Q , defined in this thesis as the flow closure energy, is the energy remaining in the pipe system in a uni-directional flow which affects the flow reversal mechanism in a bi-directional transient flow. This can be numerically estimated as:

$$E_Q = E_{w1}(T_{m1}) - E_{w2}(T_Q) \quad 5.19$$

where the magnitude of E_Q can be graphically presented as the net flow energy above the $Q = 0$ axis (Figs. 5.2 - 5.4).

In this uniformly decelerating flow the maximum surge energy occurs at the time of flow interception (T_{ms}). This is smaller than the time of pipe-end flow closure or the time of flow closure (i.e. $T_{m1} = T_{ms} < T_{m1} < T_Q$). Relation of Eq. 5.19 can then be rearranged as:

$$\begin{aligned} E_Q &= E_{w1}(T_{m1}) - E_{w2}(T_Q) \\ &= [E_{w1}(T_{ms}) + E_{w1}(T_{ms} < T < T_{m1})] - [E_{w2}(T_{ms}) + E_{w2}(T_{ms} < T < T_Q)] \\ &= [E_{w1}(T_{ms}) - E_{w2}(T_{ms})] + [E_{w1}(T_{ms} < T < T_{m1}) - E_{w2}(T_{ms} < T < T_Q)] \\ &= -E_{m1} + ET_m \end{aligned} \quad 5.20$$

where $-E_{m1} > E_Q$ as $ET_m < 0$.

The existence of periodic fluctuations does not therefore affect the maximum surge energy in this uniformly decelerating flow.

5.3.2.4 Non-friction transfer at $T_{mx1} < T < T_{mx2}$

For the present case of uniform flow closure the counter-surge period covers the whole duration of non-friction transfer (i.e. $T_{mx1} = T_{ns} < T_{x1} < T_d < T_{mx2}$). Hence, the existence of non-friction transfer (ET_n) should affect the transient behavior during the counter-surge period. A term energy reversal (E_{xx}) is defined as:

$$E_{xx} = E_{wx}(T_{x1}) - E_{wx}(T_{mx2}) \quad 5.21$$

It should be noted that the first half of the energy reversal (i.e. $0.5 E_{xx}$) must refill the system in order to recover the original non-elastic condition from the earlier flow closure energy (E_q) as:

$$E_q = - 0.5 E_{xx} \quad 5.22a$$

Only the second half reverse flow energy will contribute to the real counter-surge energy in the system. This means:

$$E_{mx2} = - 0.5 E_{xx} \quad 5.22b$$

The two groups can be subclassified to represent two extreme conditions during the counter-surge period in a uniform decelerating flow:

1) For $T_d = N * T_s$ --- As shown in Fig. 5.2a, for $T_d=4$, the flow closure energy (E_q) is zero. Consequently no counter-surge energy is expected to appear in the system. Using energy balance considerations, this means that:

For $H_x = 0$, $S = 1$ and $T_d = N * T_s$,

$$\begin{aligned} E_{mx2} &= - 0.5 E_{xx} \\ &= E_q \\ &= 0 \end{aligned} \quad 5.23$$

11) For $T_a = N \cdot T_s + 2$ --- As shown in Fig. 5.2a, for $T_a = 6$, the non-friction transfer $ET_s = 0$ and the flow closure energy $E_a = E_w(T_{mx1}) = 0.166$. Hence, the energy reversal

$$E_{xm} = -2 E_a = -2 * 0.166 = -0.33$$

The maximum counter-surge energy becomes

$$E_{mx2} = -0.5 E_{xm} = 0.166$$

This relationship can be expressed as:

For $H_z = 0$, $S = 1$ and $T_a = N \cdot T_s + 2$

$$\begin{aligned} E_{mx2} &= -0.5 E_{xm} & 5.24 \\ &= E_a \\ &= -E_{mx1} \end{aligned}$$

The existence of the non-friction transfer (ET_s) suggests that even in the absence of friction the mechanical energy transfer may still act as an efficient damper of the elastic energy in a transient pipe flow.

5.3.2.5 Effect of bi-directional transient flow (E_T)

After the end of counter-surge period ($T > T_{mx1}$), the non-friction transfer is non-existing (i.e. $ET_s(T > T_{mx2}) = 0$). Hence, the subsequent flow energy reversal is always twice that of the flow closure energy at the end of counter-surge period, [e.g. $E_{xm}(T_{mx2} < T < T_{mx2}+2) = 2 E_a(T_{mx2})$]. This implies the existence of a non-decaying transient property after the counter-surge period:

$$E_{mx2} = E_m(T_{mx2}) = -E_m(T_{mx2}+2) = E_m(T_{mx2}+4) = \dots = |E_T| \quad 5.25$$

5.3.3 Decreasing deceleration in a flow closure

5.3.3.1 Maximum surge energy for the flow closure

Maximum surge energy (E_{mx1}) always occurs before the round trip wave travel time (i.e. $T_{mx1} < 2$, as shown in Fig. 5.3c). In the present case of $S = -2$ and $H_z = 0$,

- i) for $T_q = 2$, $E_{mx1} = -0.65$ and $T_{mx1} = 1.5 < 2$;
- ii) for $T_q = 4$, $E_{mx1} = -0.38$ and $T_{mx1} = 1.7 < 2$; and
- iii) for $T_q = 6$, $E_{mx1} = -0.29$ and $T_{mx1} = 1.8 < 2$.

5.3.3.2 Periodicity for flow with $S < -1$

As shown in Fig. 5.3a, the source and pipe-end flows (Q_s and Q_e) intercept at $T_{zs} < 2$. This is the same as the time of maximum surge energy (i.e. $T_{zs} = T_{mx1}$). For $T_q > 2$, periodic surge fluctuations also appear after the flow interception (T_{zs}) with a period of T_s . For the case of $T_q = 6$, $S = -2$ and $H_z = 0$:

- i) $E_w(T_{zs}) = 0.29$;
- ii) $E_w(T_{zs} < T < T_{zs}+2) = -0.39 > E_w(T_{zs})$; and
- iii) $E_w(T_{zs}+2 < T < T_{zs}+4) = 0.26$

where $T_{zs} = 1.7$.

These alternating fluctuations are significant and may even be greater than the maximum flow energy, $E_w(T_{zs})$. In general, they dominate the entire period of flow closure for the case of decreasing decelerating flow.

5.3.3.3 Non-friction transfer at $T < T_{mx1}$

For the present case of $S = -2$ and $H_z = 0$ the non-friction transfer is:

- i) for $T_Q = 2$, $ET_m = -0.02$ and $T_{m1} = 1.6 < T_Q$;
- ii) for $T_Q = 4$, $ET_m = -0.13$ and $T_{m1} = 2.2 < T_Q$; and
- iii) for $T_Q = 6$, $ET_m = -0.13$, and $T_{m1} = T_Q = 6$.

The negative non-friction transfer is non-additive to the increase of maximum surge energy since $E_{mx1} = -E(T_{m1})$.

5.3.3.4 Non-friction transfer at $T > T_{mx1}$

Combining Eqs. 5.20 and 5.22, the maximum counter-surge energy can be determined from the flow closure energy as:

$$\begin{aligned}
 E_{mx2} &= E_Q \\
 &= E_w(T_{m2}) + ET_m \\
 &= -E_{mx1} + ET_m
 \end{aligned}
 \tag{5.26}$$

so $E_{mx2} < -E_{mx1}$, since $ET_m < 0$.

Applying Eq. 5.26 to the present case of $S = -2$ and $H_s = 0$, the maximum counter-surge energy is (Fig. 5.3a):

- i) for $T_Q = 2$, $E_{mx2} = 0.65 - 0.02 = 0.63$;
- ii) for $T_Q = 4$, $E_{mx2} = 0.38 - 0.13 = 0.25$; and
- iii) for $T_Q = 6$, $E_{mx2} = 0.29 - 0.13 = 0.16$.

The relationship of Eq. 5.26 indicates that the maximum surge energy is generally greater than that of the maximum counter-surge energy for flows with decreasing deceleration. The non-friction transfer (ET_m) has damped out a significant part of the transient energy during the counter-surge period (i.e. $T_{mx1} = T_{m2} < T < T_{mx2}$).

5.3.4 Increasing deceleration in the flow closure

5.3.4.1 Maximum surge energy for flow with $S > 1$

As shown in Fig. 5.4a and 5.4c the maximum surge energy (E_{mx1}) occurs at or after the time of flow closure (i.e. $T_{mx1} > T_a$). For the case of $S = 2$ and $H_f = 0$:

- i) for $T_a = 2$, $E_{mx1} = -0.63$ and $T_{mx1} = 2.4 > T_a$;
- ii) for $T_a = 4$, $E_{mx1} = -0.25$ and $T_{mx1} = 4 = T_a$; and
- iii) for $T_a = 6$, $E_{mx1} = -0.16$ and $T_{mx1} = 6.2 > T_a$.

It should be noted that for the present case, $S = 2$ represents only one of the typical flow closures with increasing deceleration ($S > 1$). Further sub-classification is carried out below in association with the analysis of flow periodicity.

5.3.4.2 Periodicity for flow with $S > 1$

Three groups of periodic surge fluctuations are subdivided. These are:

- i) Fluctuation-free;
- ii) Fluctuation with positive addition; and,
- iii) Fluctuation with alternating addition.

The key criterion for the above classifications is the effect of surge fluctuation on the potential increase of maximum surge energy during the flow closing period. The boundary (S_x) between the first two groups (i and ii) is only introduced for the sake of simplicity since both provide positive addition to the elastic energy during the flow closing period. The division (S_y) between the last two groups [(ii) and (iii)] can however be theoretically derived. This development will be presented in Appendix

A.3. For the discussion of the present thesis, $S_x = 2^+$ (e.g. for $S = 2$ and $T_d < 6$, $2 < S_x < 5$, in Figs. 5.5c and d) and $S_y = 1.7$ (as derived in A.3) are adopted.

1) Fluctuation-free ($S > S_x = 2^+$) ---

As shown in Fig. 5.5d, during the course of flow closing the rate of pipe-end flow outflux is always greater than the source flow influx (i.e. $Q_m > Q_s$).

If the shape factor is sufficiently high, the effect of the periodic surge fluctuation (e.g. $S = 5$) will entirely disappear. For this case the source and pipe-end flows do not intercept during the flow closing period. Here the surge energy increases rapidly with time and reaches its maximum after the end of flow closure (This implies that: $T_{mx1} = T_{ms} = T_{m1} > T_d$).

For a sufficiently long time of flow closure, flows at source (Q_s) and pipe-end (Q_m) will be parallel (but not collinear) in the sense that their rates of deceleration are equal (i.e. $dQ_s/dT = dQ_m/dT$).

2) Positive additive fluctuation ($S_y = 1.7 < S < S_x = 2^+$) ---

As shown in Fig. 5.5c or Fig. 5.4a (e.g. $S = 2$), the rate of flow outflux is greater than or equal to the influx at the source (i.e. $Q_m > \text{or} = Q_s$).

General characteristics of this additive fluctuation can be shown by an example of $T_d = 6$, $S = 2$ and $H_x = 0$ (Fig. 5.4a). Typical transient properties are identified as:

- i) The peak rate of flow difference ($Q_m(T) - Q_s(T)$) occurs at the mid-point between unit periods of surge fluctuation (i.e. $T = N \cdot T_d + 2$). For example:

$$Q_m(T) - Q_s(T) = 0.06 \text{ and } 0.04 \text{ for } T = 2 \text{ and } 6.$$

ii) Difference of two flow is negligible near the time instant at the unity of surge fluctuating period ($T = N \cdot T_s$) as:

$$Q_m(T) - Q_s(T) = 0 \text{ for } T = 4.$$

iii) The effect of periodic fluctuation is positive and decreasing during the flow closing period:

$$--- E_w(4) = 0.12; \text{ and}$$

$$--- E_w(4 < T < 6.2) = 0.04$$

where the two flows (Q_s and Q_m) do not intercept until the end of flow closure.

These properties of additive fluctuation imply that the surge energy will increase with time and reach its maximum at or after the time of complete flow closure.

iv) A periodic change of slope on the curve of transient energy $E_m(T)$ vs. time T may occur. For this example (Fig. 5.4c) the transient energy (E_m) increases rapidly with time until approaching $T=3$. The increment of transient energy at $3 < T < 5$ is then immaterial prior to the another rapid rise of E_m at $5 < T < 6.2$.

v) The dominance of netflux $E_w(4)$ over the subsequent periodic fluctuations, $E_w(4 < T < 6.2)$ suggests that the periodic fluctuation effect is diminishing during the flow closing period but with positive additive fluctuation.

3) Alternate additive fluctuations ($1 < S < S_y=1.7$) ---

The general characteristics of this alternate additive flow can be discussed with the example of $T_a = 6$, $S = 1.5$ and $H_z = 0$ (Fig. 5.5b). Three periodic properties are also identified:

i) The peak rate of flow difference ($Q_x(T) - Q_s(T)$) occurs within the interval $0.5 T_s$. For example:

$$Q_x(T) - Q_s(T) = -0.10 \text{ and } 0.07 \text{ for } T = 3.5 \text{ and } 5.5$$

ii) Two flows (Q_s and Q_x) intercept as an interval of $0.5 T_s$ as:

$$Q_x(T) - Q_s(T) = 0 \text{ for } T_{xs} = 2.5 \text{ and } 4.5.$$

iii) Half of these periodic fluctuating events are negatively additive, which results in an alternating increase in flow energy, as shown in Fig. 5.5b. Two further cases can be separated for discussion:

Case a) $T_q = N \cdot T_s + 2$ --- For example: $T_q = 6$,

$$E_w(T_{xs}) = E_w(2.5) = 0.14$$

$$E_w(T_{xs} < T < T_{xs} + 2) = E_w(2.5 < T < 4.5) = -0.06$$

$$E_w(T_{xs} + 2 < T < T_{x1}) = E_w(4.5 < T < 6.2) = 0.07$$

where $T_{xs} < T_q < T_{x1} = T_{mx1}$.

Case b) $T_q = N \cdot T_s$ --- For example: $T_q = 4$,

$$E_w(T_{xs}) = E_w(2.5) = 0.22$$

$$E_w(T_{xs} < T < T_{x1}) = E_w(2.5 < T < 3.8) = -0.06$$

where $T_{xs} = T_{mx1} < T_{x1} < T_q$.

The properties of these two conditions indicate the potential destructive effect of wave reflection on either the surge energy (Case a) or the counter-surge energy (Case b).

5.3.4.3 Non-friction transfer at $T < T_{mx1}$

Three flows with different types of periodic fluctuation are discussed herein:

1) Fluctuation-free ($S > S_x=2^+$) ---

As shown in Fig. 5.5d, the two flows (Q_s and Q_m) do not intercept during the flow closing period. For this case, $T_{mx1} = T_{ms} = T_{m1} > T_q$. The non-friction transfer (ET_m) is negligible.

2) Positive additive fluctuation ($S_y=1.7 < S < S_x=2^+$) ---

In the case of $S = 2$, $T_q = 6$ and $H_z = 0$,

$$ET_z = 0, \text{ and}$$

$$E_{mx1} = E_w(T_{m1}) = E_w(T_{ms}) = 0.16$$

where $T_{ms} = T_{m1} = T_{mx1} = 6.2 > T_q$.

Both flows with fluctuation free and positive additive fluctuation contain $ET_m = 0$, which ensures that the surge energy is increasing until the end of flow closure. However, the effect of periodic fluctuations exists in the latter flow (2), while it is negligible in the former flow (1).

3) Alternate additive fluctuation ($1 < S < S_y=1.7$) ---

Two cases are discussed in a separate manner (Fig. 5.5b):

Case 1) $T_q = N \cdot T_s + 2$ --- The non-friction transfer is positive though small, resulting a slow increment of surge energy toward the end of flow closing period. In the previous example of $T_q = 6$,

$$\begin{aligned}
 ET_{\Sigma} &= E_w(T_{\Sigma 2} < T < T_{\Sigma 1}) \\
 &= E_w(2.5 < T < 6.2) \\
 &= -0.06 + 0.07 \\
 &= 0.01 > 0
 \end{aligned}$$

$$\begin{aligned}
 -E_{mx1} &= E_w(T_{\Sigma 2}) + ET_{\Sigma} \\
 &= 0.14 + 0.01 \\
 &= 0.15
 \end{aligned}$$

where $T_{\Sigma 2} = 2.5 < T_Q < T_{\Sigma 1} = T_{mx1} = 6.2$.

Case ii) $T_Q = N \cdot T_{\Sigma}$ --- The non-friction transfer (ET_{Σ}) is negative, with result that the maximum surge energy occurs before the time of flow closure. For the case of $T_Q = 4$,

$$\begin{aligned}
 ET_{\Sigma} &= E_{wx}(T_{\Sigma 2} < T < T_{\Sigma 1}) - E_{ws}(T_{\Sigma 2} < T < T_Q) \\
 &= -0.06 < 0
 \end{aligned}$$

$$E_{mx1} = -E_w(T_{\Sigma 2}) = -0.22$$

where $T_{\Sigma 2} = T_{mx1} = 2.7 < T_{\Sigma 1} = 3.8 < T_Q$.

5.3.4.4 Non-friction transient transfer at $T > T_{mx1}$

Three flows with different types of increasing deceleration are discussed accordingly:

1) Fluctuation-free ($S > S_{x=2+}$) ---

Since there is no fluctuation during the flow closing period, $ET_{\Sigma} = 0$ or $E_Q = E_w(T_{\Sigma 2}) = -E_{mx1}$. This implies that Eq. 5.24 is valid for this condition.

For the case of $S = 5$ and $T_Q = 4$:

$$ET_{\Sigma} = 0, \text{ and}$$

$$E_{mx2} = -E_{mx1} = 0.63$$

2) Positive additive ($S_y = 1.7 < S < S_x = 2^+$) ---

For this case the periodic fluctuations occur prior to the counter-surge period because the non-friction transfer $ET_m = 0$. The relation given by Eq. 5.24 is also valid for this condition.

For the case of $S = 2$ and $T_q = 6$:

$$ET_m = 0.04 > 0, \text{ and}$$

$$E_{mx2} = -E_{mx1} = 0.16.$$

3) Alternate additive ($1 < S < S_y = 1.7$) --- Two cases are discussed below:

Case i) $T_q = N * T_s + 2$ --- The non-friction transfer $ET_m > 0$, which implies that Eq. 5.24 is valid for this situation. For the case of $S = 1.5$ and $T_q = 6$:

$$ET_m = 0.01 > 0, \text{ and}$$

$$E_{mx2} = -E_{mx1} = 0.15$$

Case ii) $T_q = N * T_s$ --- The non-friction transfer $ET_m < 0$. This confirms the validity of Eq. 5.26 applied to this condition. For the case of $S = 1.5$ and $T_q = 4$,

$$ET_m = -0.06 < 0, \text{ and}$$

$$\begin{aligned} E_{mx2} &= -E_{mx1} + ET_m \\ &= 0.22 - 0.06 \\ &= 0.16 \\ &< -E_{mx1} \end{aligned}$$

5.3.5 Discussion of periodic fluctuation

It has been demonstrated in this section that the physical phenomena of transient flows can be appropriately

interpreted by a method incorporating the Energy Balance Concept.

It can be concluded that the existence of periodic fluctuations governs transient behavior during and after the flow closure.

Subsequent to the detailed analysis of transient flow events to a limited number of typical cases the present findings will be incorporated into the interpretation of wide ranged flow closing conditions. This is undertaken in the next section.

5.4 Analysis of Maximum Transient Energy

The maximum transient energy (E_{mx}) was obtained from the numerical model, and results are summarized in Fig. 5.6 and 5.7 for the cases of maximum surge (E_{mx1}) and maximum counter-surge (E_{mx2}) respectively.

5.4.1 Maximum surge energy (E_{mx1} in Fig. 5.6)

5.4.1.1 Symmetrical property of E_{mx1}

The following symmetrical relation can be observed for a short time (T_d) of flow closure or high shape (S) of flow closure as:

$$E_{mx1}(+S) = E_{mx1}(-S) \quad 5.7$$

where $+S$ and $-S$ represent the shapes of flow closure with increasing deceleration and decreasing deceleration respectively, with the same absolute magnitude.

The transient energy is not affected by the periodic fluctuation during the surge period, or would be damped out during the subsequent counter-surge period.

The domain of this condition (Fluctuation-free) is limited, and diminishing very rapidly with the increase of T_d . For example, it is valid for:

i) all S , at $T_d < 2$;

ii) $|S| > 5$, at $T_d = 4$; and

iii) $|S| > 10$, for $T_d > 6$.

5.4.1.2 Non-symmetrical property of E_{mx1}

Most of the results in the figure agree with the relation below:

$$|E_{mx1}(+S)| < |E_{mx1}(-S)| \quad 5.28$$

Since the maximum surge energy of decreasing deceleration flow ($S < 1$) occurs prior to the development of periodic fluctuations, $E_{mx1}(-S)$ is not affected by the existence of periodic fluctuations.

On the other hand, periodic fluctuations usually exist during the surge period in flow with increasing deceleration ($S > 1$). Consequently, part of the surge energy (by wave reflection) would be destroyed during the flow closing period.

Therefore, the non-symmetric maximum surge energy of flow between $+S$ and $-S$ suggests that the effect of periodic fluctuation on $E_{mx1}(+S)$ may be assessed by the comparison of their non-friction transfer ($ET_{\pm}$) characteristics.

For the case of $T_d = 6$:

i) $ET_{\pm}(S = +2) = 0$, indicating a continued increase in surge energy during the entire flow closing period;

11) $ET_{-}(S = -2) = -0.13$, showing that the existence of periodic fluctuations will act as a damper during the counter-surge period.

For increasing deceleration flow with positive additive fluctuations, e.g. $S = 2$, the periodic fluctuation has completely dissipated its destructive effect (equivalent to $ET_{-} = 0.13$) during the surge period, resulting in:

$$-E_{mx1}(+2) = 0.16 < -E_{mx1}(-2) = 0.29$$

No further decay of transient energy occurs during the subsequent counter-surge period since:

$$-E_{mx1}(+2) = E_{mx2}(+2) = 0.16$$

5.4.1.3 Optimal flow closure during surge period

i) $S^* = 1$ is the optimal shape factor for $T_q < 2$; however

ii) $S^* = 1.5$ is the optimal shape factor for most flow closing conditions, i.e. $T_q > 2$.

The alternating surge fluctuations exist at $1 < S < 1.7$, resulting a great potential for the maximum surge energy reduction.

5.4.2 Maximum counter-surge energy (E_{mx2} in Fig. 5.7)

Two distinct transient properties are identified from the figure and are stated below.

5.4.2.1 Symmetrical property of E_{mx2}

The maximum counter-surge energy (E_{mx2}) is symmetric at $S = 1$ for any time of flow closure (T_q). This means:

$$E_{mx2}(+S) = E_{mx2}(-S) \quad 5.29$$

During the counter-surge period substantial transient energy has been decayed by the existence of periodic fluctuations for flow with decreasing deceleration ($S < -1$).

For the example of $T_a = 4$ and $S = -2$, there is no destructive effect from periodic fluctuations during the surge period since:

$$-E_{mx1}(-2) = 0.29 > -E_{mx1}(+2) = 0.16$$

However, the periodic fluctuations has by now exhausted all its destructive potential during the subsequent counter-surge period since:

$$-E_{mx1}(-2) = 0.29 > E_{mx2}(-2) = E_{mx2}(+2) = 0.16$$

5.4.2.2 Peak decaying energy (E_T)

All the potentially destructive energy has been exhausted either during the surge period (for $S > 1$), or during the counter-surge period (for $S < -1$). There will be no further damping effect (by wave reflection) afterward and the relation given by Eq. 5.25 is also valid for all type decelerating flows.

5.4.2.3 Optimal shape factor S^* of E_{mx2}

Optimal shape factors are located at three locations for different times of flow closure. The existence of alternate periodic surge fluctuation is the reason for their occurrence:

i) $S^* = 1$, for $T_a = N * T_s$ --- The large quantity of negative non-friction transfer, i.e. $-ET_s = E_w(T_{mx1})$ has totally damped out the transient energy during the counter-surge period.

ii) $S^* = 1.5$, for $T_a = N * T_s + 2$ --- The periodic fluctuations still possess destructive potential, during surge period, for flows with $1 < S < 1.7$.

iii) $S^* = -1.5$; for $T_a = N * T_s + 2$ --- For flows with $-1 > S > -1.7$, the existence of a periodic surge fluctuation affects the decay of the transient energy during the counter-surge period.

5.4.3 Discussion of maximum transient energy

In this section, it has been demonstrated that the maximum transient energy can be analyzed with a knowledge of periodic fluctuation properties which have been previously established by the concept of an energy balance [for example, refer to part (a) of Figs. 5.2 - 5.4].

It should be noted that using the traditional pressure response method it is impractical (if not impossible) to interpret the physical phenomena of these periodic fluctuations [for example, this may refer to part (c) of Figs. 5.2 - 5.4].

5.5 Conclusions

The research conducted in this chapter focused on the transient characteristics in a non-frictional closed conduit system under various categorized flow closing patterns.

One objective of this chapter was the development and application of an entirely new approach to the analysis of transient phenomena.

1.) Energy Balance Concept ---

This new method for the interpretation of the physical phenomena of transient flows is termed the *Energy Balance Concept*.

1) Characteristics --- This energy concept integrates the past transient history of the instantaneous surge condition and takes the hydraulics of the whole system into consideration.

ii) Present application --- It has been demonstrated that the concept of an energy balance provides a powerful alternative to the traditional instantaneous and point-specific force/reaction approach to hydraulic transient analysis. The former provides the best tool for a clear interpretation, while the latter can be conveniently applied to a broad-ranged analysis after the transient phenomenon has been well defined.

iii) Future application --- The energy balance concept introduced in this chapter will also be applied to the analysis of various frictional systems. The application to the present non-frictional system has successfully provided a well-defined set of criteria for the classification of various degrees of impact from steady friction and unsteady friction (factor) conditions. This will be presented in a set of solution domain contained in later chapters of this study.

Surge fluctuations with a period $T_s = 4$ generally exist throughout the entire transient period in a non-frictional pipe system. Their general characteristics can be summarized:

2) Fluctuation-free (E_{mx1} and E_{mx2}) --- The effect of periodic fluctuations is non-existent in a transient flow with:

i) short time of flow closure (T_d); and

ii) high shape of flow closure (S).

3) Periodic fluctuations to surge energy (E_{mx1}) --- The existence of fluctuations ($ET_s > 0$) may reduce the rate of the surge energy increase during the flow closing period, but will fail to decay transients during the counter-surge period (i.e. Eq. 5.24). This covers flows with:

i) $S > 1.7$ (for all T_a); and

ii) $1 < S < 1.7$ ($T_a = N * T_s + 2$).

4) Periodic fluctuations to counter-surge energy (E_{mx2}) --- On the other hand, the existence of periodic fluctuations ($E_{T_s} < 0$) has damped out a significant part of transient energy during the counter-surge period. The relationship given by Eq. 5.26 indicates that the maximum surge energy is generally greater than that of the maximum counter-surge energy for flows with:

i) $S < 1.7$ ($T_a = N * T_s$); and

ii) $S < -1$ (for all T_a).

5) Periodic fluctuations to peak decaying energy (E_T) --- After the counter-surge period ($T > T_{mx2}$), the effect of non-friction transfer ceases to exist throughout the decaying period (i.e. $E_{T_s} = 0$). This results:

$$E_{mx2} = E_m(T_{mx2}) = -E_m(T_{mx2}+2) = E_m(T_{mx2}+4) = \dots = |E_T|$$

Two transient properties relevant to general practices have also been investigated, and conclusions are made below:

6) Optimal shape of flow closure (S^*) ---

The least maximum transient energy for the same time of flow closure is located at the points of:

i) maximum surge energy --- $S^* = 1.5$ for $T_a > 4$;

ii) maximum counter-surge energy ---

a) $S^* = 1.5$ and $S^* = -1.5$ for $T_a = N * T_s + 2$; and

b) $S^* = 1$ for $T_a = N * T_s$.

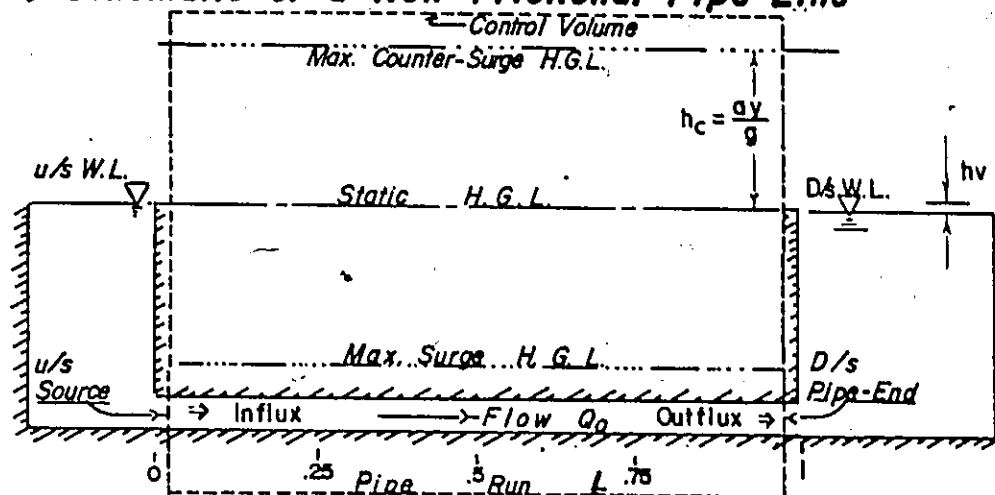
The traditional theory that a uniform flow closure produces the least transient energy is not valid for non-frictional conditions.

In this chapter, transient analysis has been completed in the non-frictional system. The present research proceeds to investigate how the mechanical transfer phenomena affect the transient behavior with the existence of friction. This will be carried out in next chapter.

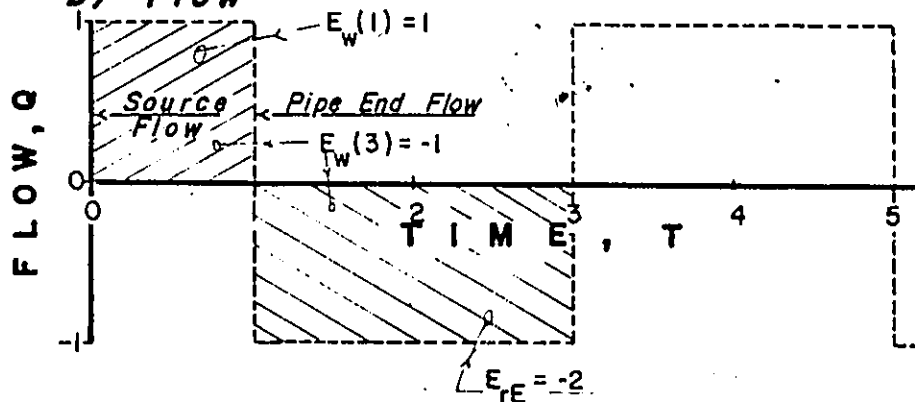
Table 5.1 Summary of Non-frictional Analysis

Shape S	Time T_a	Maximum transient energy			
		Surge		Counter-surge	
		E_{mx1}	T_{mx1}	E_{mx2}	T_{mx2}
1	2	-.50	2	.50	4
	4	-.25	2	0	--
	6	-.16	2	.16	8
-2	2	-.65	1.5	.63	3.5
	4	-.38	1.7	.25	4
	6	-.29	1.8	.16	8
2	2	-.63	2.4	.63	4.4
	4	-.25	4	.25	6
	6	-.16	6.2	.16	8
-1.5	2	-.60	1.7	.60	3.7
	4	-.35	1.8	.16	6
	6	-.23	1.9	.15	8
1.5	2	-.60	2.3	.60	4.3
	4	-.22	3.8	.16	6
	6	-.15	6.2	.15	8

a) Schematic of a Non-Frictional Pipe-Line



b) Flow



c) Energy

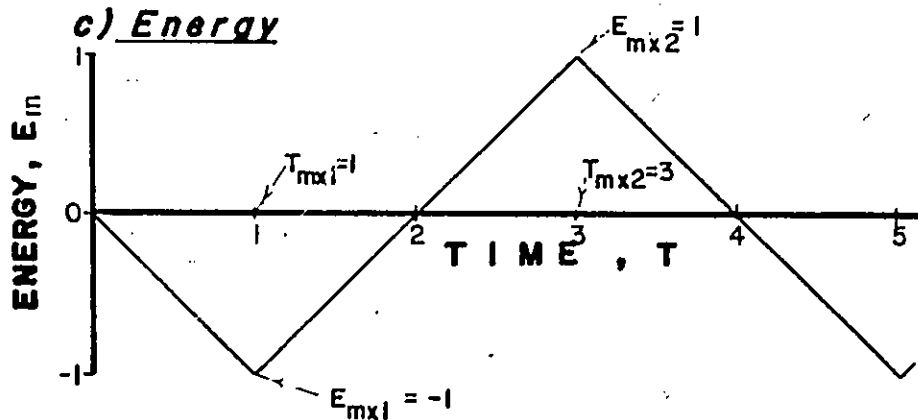


Fig. 5.1—Transient History of Non-Frictional, Instant Flow Closure.

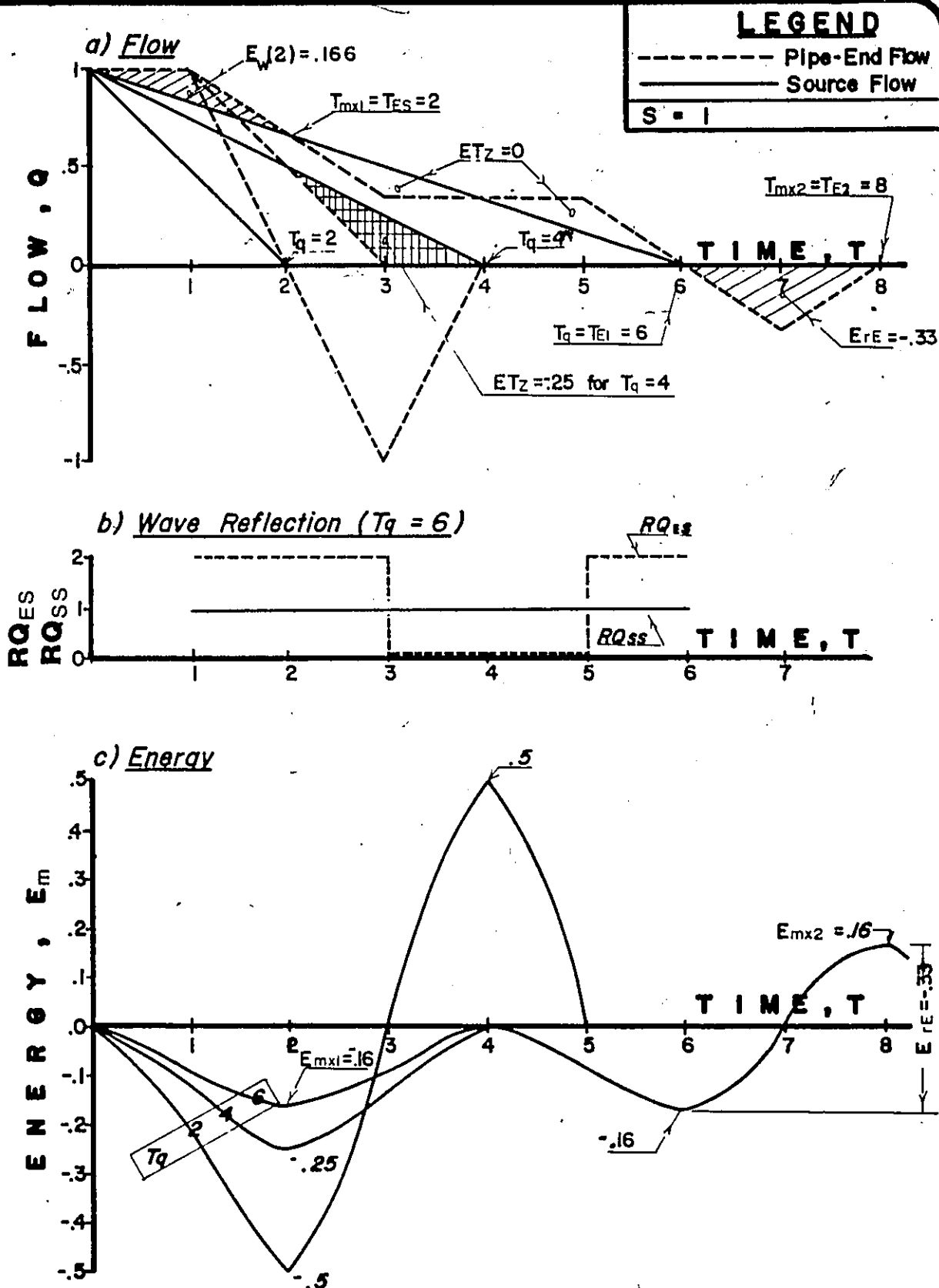


Fig. 5.2—Transient History of Non-frictional, Uniform Deceleration Flow.

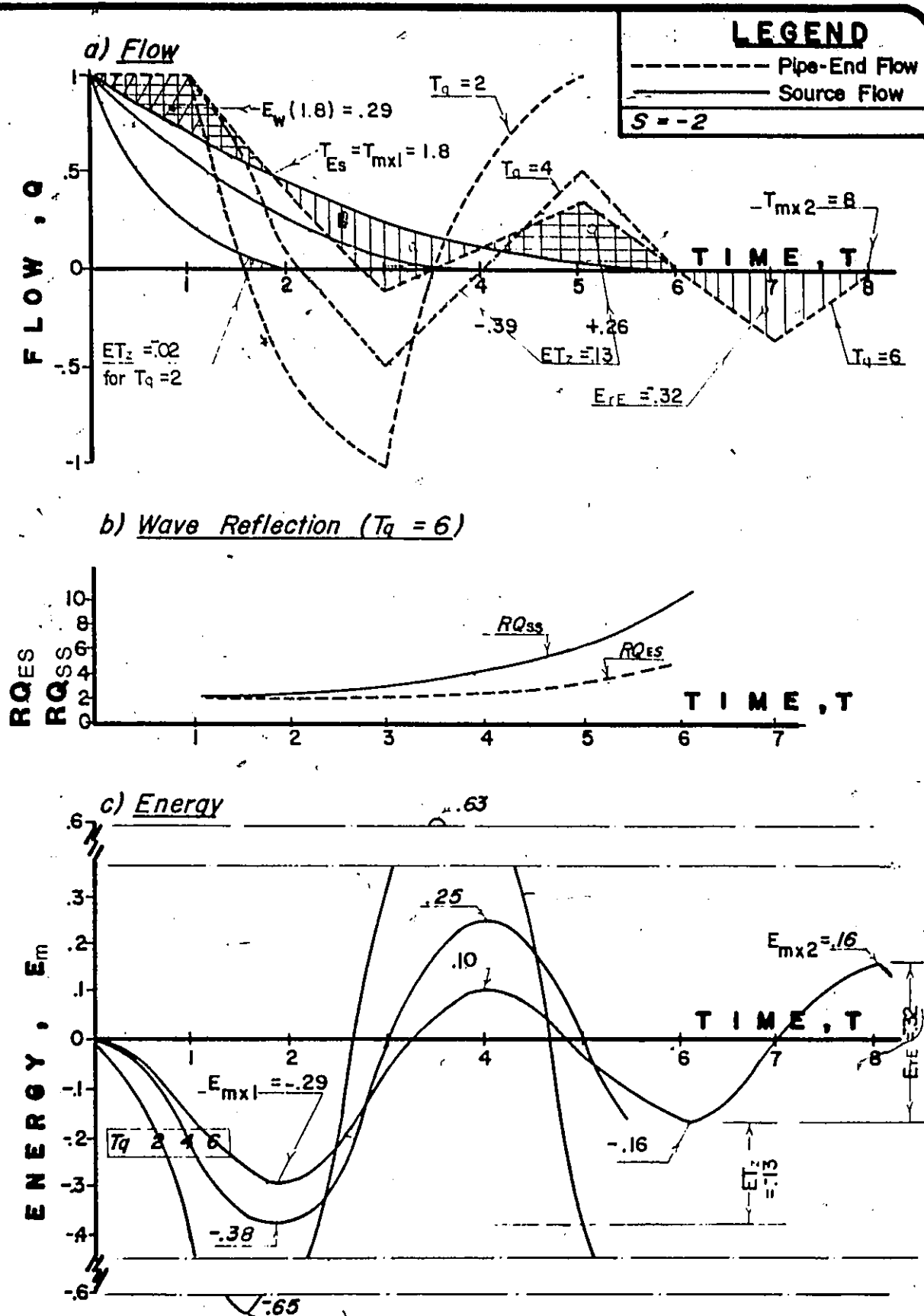


Fig. 5.3—Transient History of Non-frictional, Decreasing Deceleration Flow.

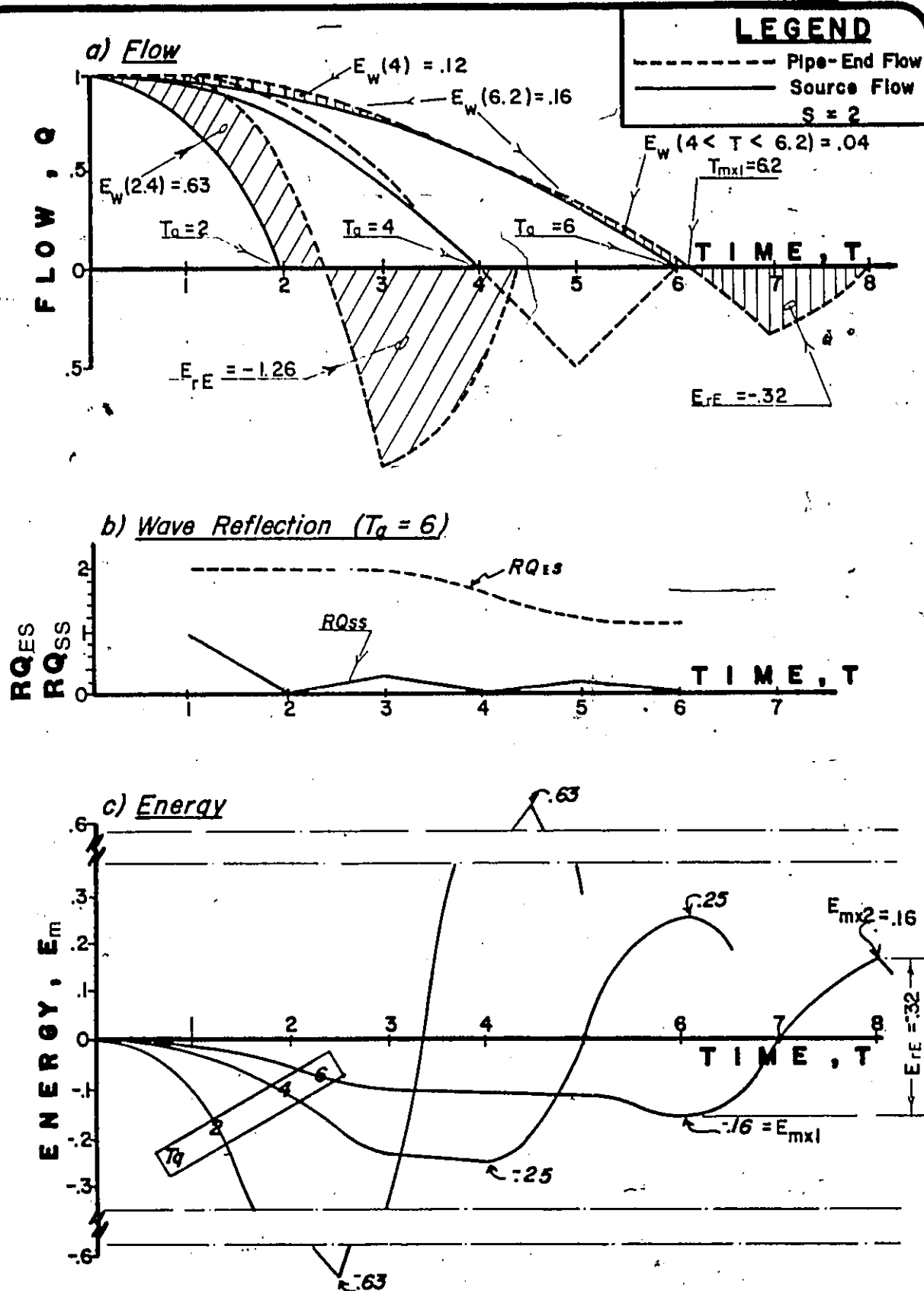


Fig. 5.4—Transient History of Non-Frictional, Increasing Deceleration Flow.

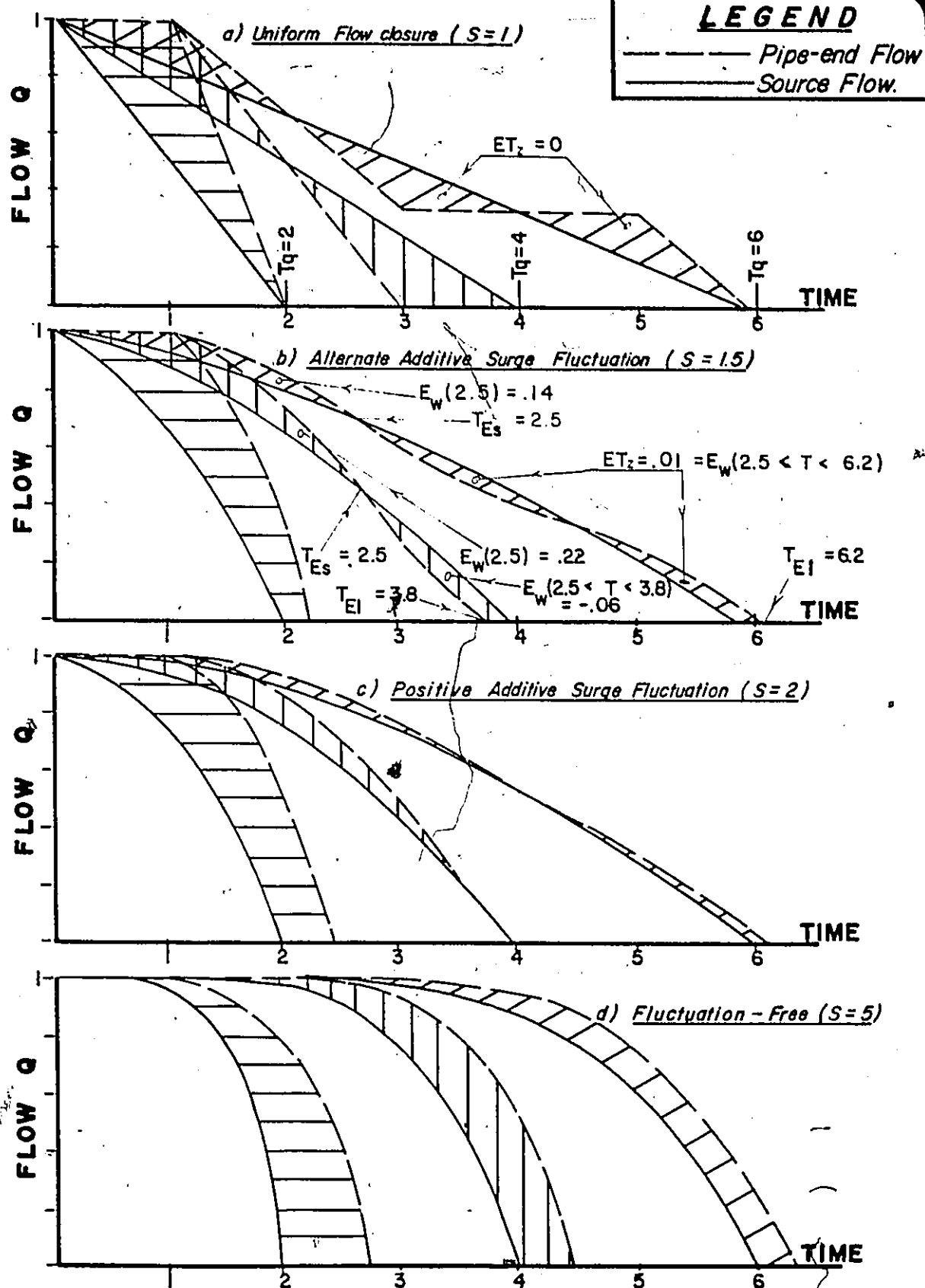
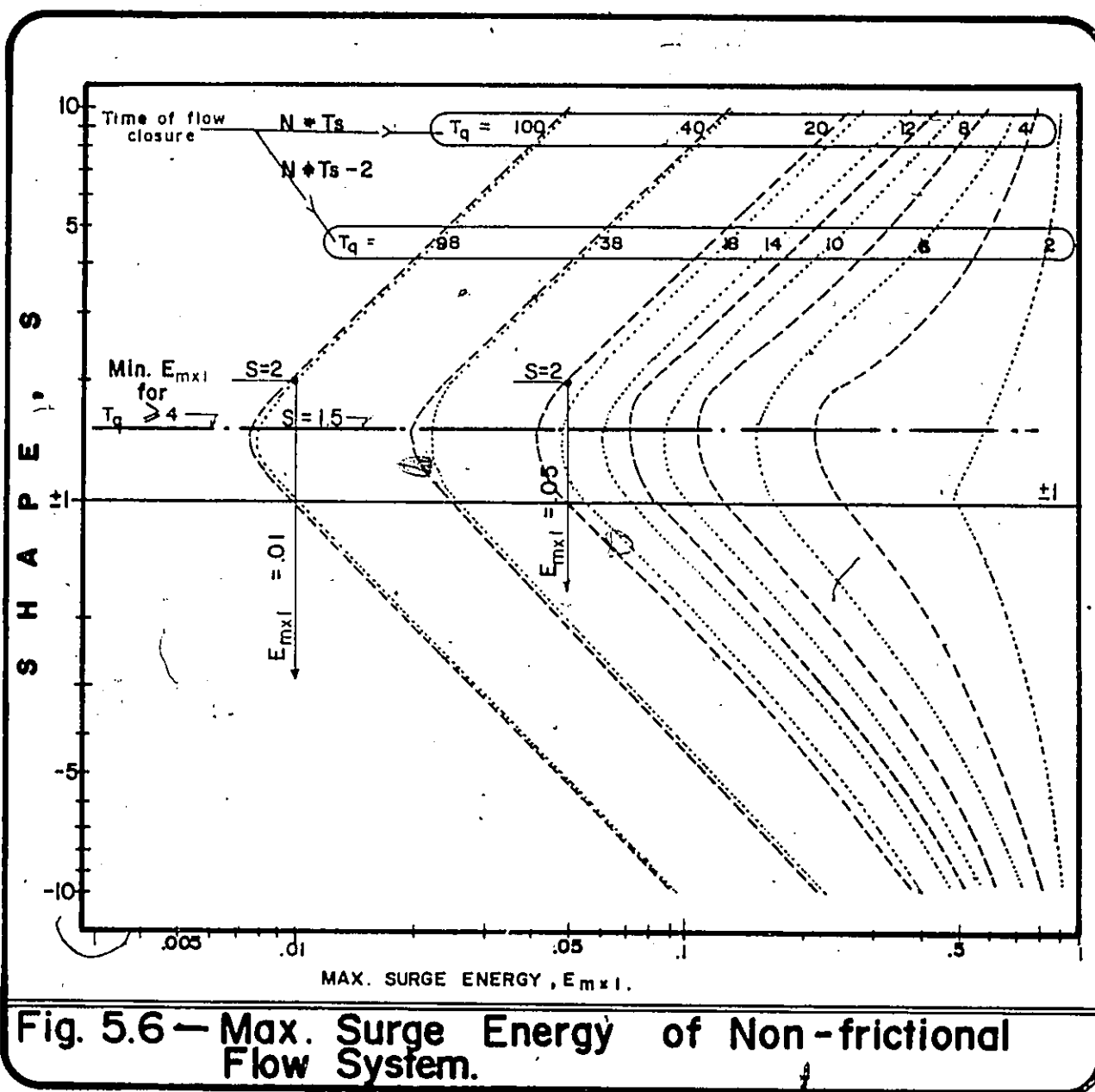


Fig. 5.5 — Flow History of Periodic Surge Fluctuations



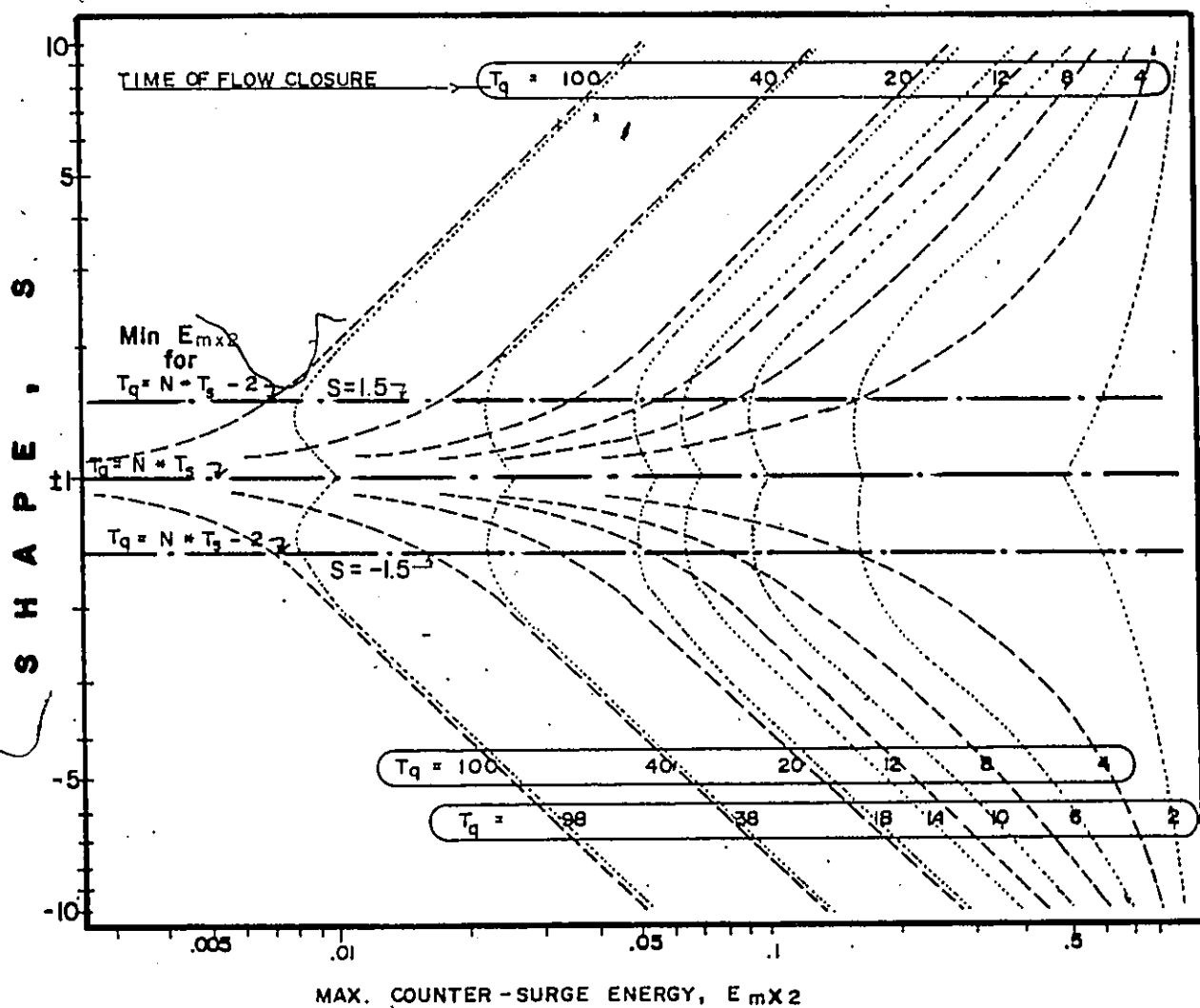


Fig. 5.7—Max Counter-surge Energy E_{mx2} of non-frictional Flow System.

6.0 ANALYSIS OF TRANSIENTS IN A STEADY FRICTION SYSTEM

The Energy Balance Concept was introduced and applied to interpret transient phenomena in a non-frictional system in the previous chapter. The adoption of this new concept continues in the present steady friction (factor) system, aided with the traditional pressure response approach.

Previous transient analysis (Ch. 5.0) also demonstrated how periodic fluctuations affect the transient behavior in a non-frictional system. In this chapter the investigation focuses on the mechanical transfer phenomenon in frictional systems having a steady friction factor (f).

The transient characteristics affected by the existence of frictional energy can be identified by comparing the results of the previous analysis of non-frictional and the present analysis of frictional systems.

6.1 Energy Relation in Transient Flow

The analysis in this section focuses on the development of energy relations in a frictional pipe system.

6.1.1 Scope of analysis

Cases representing typical flow closing characteristics in a frictional system are presented in this section for discussion. Previous illustrations—of non-frictional systems have been used as background information for the analysis of steady friction systems.

Figs. 6.1, 6.2, 6.3 and 6.4 present the transient flow and energy history for an instantaneous closure ($S = \infty$), and for flows with uniform ($S = 1$), decreasing ($S = -2$) and increasing ($S = 2$) decelerations.

A relatively rapid flow closure $T_c = 4$ is first selected, because of its significant impact on the transient energy.

Four friction heads are presented, namely $H_f = 0, 0.1, 1$ and 5 . These represent hydraulic conditions in a steady piping system varying from an ideal fluid to very high friction conditions. Among these, $H_f = 1$ is selected for more detailed analysis due to the significance of the frictional effect for this case.

The time and magnitude of critical transient properties (e.g. E_{mx1} , E_{mx2} , E_T) can be observed from the figures and have been summarized in Table 6.1.

Though limited in number, the cases illustrated in the first two sections (Sec. 6.1 and 6.2) of this chapter cover the typical flow closing characteristics with wide-ranging friction conditions. These should be sufficient to explain the most important types of behavior for transient pipe flow in frictional systems.

6.1.2 Steady state condition in a pipe

Fig. 6.1a is the schematic of a piping system under various friction head conditions. By assuming a linear hydraulic grade line (H.G.L.) along the pipeline the internal energy of the piping system is:

$$E(T) = E_f = 0.5 H_f \quad 6.1$$

where the friction head (H_f) can be determined directly from the Darcy-Weisbach friction equation.

The assumption of linear H.G.L. is realistic for a uniform and homogeneous pipeline. The term, E_f , is defined as the (steady) friction energy in this research. For the special case of a non-frictional system, $E(T) = E_f = 0$, which is consistent with the expression of an ideal fluid.

Under steady state conditions, the energy outflux (E_{out}) is the same as previously mentioned for the non-frictional system (Eq. 5.3) and this is independent of the steady

friction energy. This independence is due to the absence of surge wave reflection. Restated, the flow energy outflux at time T_1 is:

$$E_{w\pi}(T_1) = \int_0^{T_1} Q(T) dT \quad 6.2$$

where its value is independent of the friction energy (E_f) in the system. Here $E_w = E_{w\pi}$ and $Q = Q_{\pi}$.

In a frictional system the energy commonly termed friction dissipation (ED_f) continues to dissipate energy by the exchange of heat. At the time when a pipe length of water has been conveyed through the pipe-end boundary line [i.e. $T_1 = (1/v_o)/(1/a) = a / v_o$, in a non-dimensional form], the energy loss (ED_f) will be accumulated as:

$$\begin{aligned} \{ & \int_0^{a/v_o} Q(X, T) dT \\ \{ & ED_f(a/v_o) = \\ \{ & \\ \{ & \\ \{ & = E_f \end{aligned} \quad 6.3$$

where the steady flow rate, $Q(X, T) = Q_o = 1$ is independent of distance (X) and time (T). The time ($T_1 = a/v_o$) is in the order of 1000, in general practice.

The energy loss (friction dissipation) is resupplied continuously from the upstream source (e.g. by pump or reservoir) during the steady flow condition. This results in a constant internal energy, $E(T)$, as expressed by Eq. 6.1. It should be noted that the friction dissipation (ED_f), in the form of heat exchange, is the only cause of energy loss in steady pipe flows. Other causes, e.g. so-called local loss, are considered to be relatively small, and are outside the scope of analysis of this research.

6.1.3 Friction dissipation in a transient flow

1) Relevance of friction dissipation ---

Upon an instantaneous flow closure, a constant rate of flow outflux is in effect, until $T = T_{mx1} = 1$ (see Fig. 6.1b). During this period, the transient behavior is not influenced by the effect of wave reflection.

The value of friction dissipation (ED_z) within the surge period ($0 < T < T_{mx1}$) can be approximated as:

$$\begin{aligned}
 \{ \quad ED_z(T_{mx1}) &= \int_0^1 \int_0^1 Q(X,T) \, dX \, dT \\
 \{ \\
 \{ \\
 \{ &= \int_0^1 (0.5 * Q_0) \, dT & 6.4 \\
 \{ \\
 \{ &= 0.5 * v/a * E_z \\
 \{ \\
 \{ &= 0.0005 * E_z
 \end{aligned}$$

where the average transient flow in pipe is analogy to half full during the surge period (i.e. $Q(T) = 0.5 * Q_0$) and the velocity ratio (v/a) is usually in the order of 0.001.

The analysis of friction dissipation (ED_z) can also be undertaken for other hydraulic conditions in a similar manner. Results are summarized in Table 6.2. It is found that the values of friction dissipation (ED_z) are insignificant for general transient conditions where $T_0 < 100$.

2) Illustration ---

1) Given: A pipe system with the same geometries and flows as that in Sub. 5.2.3 is considered. The fluid

conveyed is water under hydraulically smooth condition.
This implies,

$$\begin{aligned}
 R &= v \cdot d / \nu = (1 \text{ m/s} \cdot 1.13 \text{ m}) / 1.13 \cdot 10^{-6} \text{ m}^2/\text{s} = 10^6 \\
 f &= 0.0115 \text{ for friction factor} \\
 h_f &= f \cdot (l/d) \cdot (v^2/2g) \\
 &= 0.0115 \cdot (1000 \text{ m} / 1.13 \text{ m}) \cdot (1^2 \text{ m}^2/\text{s}^2 / 20 \text{ m/s}^2) \\
 &= 0.5 \text{ m} \\
 H_f &= h_f/h_0 = 0.5 \text{ m} / 100 \text{ m} = 0.005
 \end{aligned}$$

11) Friction dissipation in an instantaneous closure within 100 seconds would be:

$$\begin{aligned}
 &0.5 \gamma \cdot h_f \cdot A \cdot (0.1 \text{ l}) \\
 &= 0.5 \cdot 10^4 \text{ N/m}^3 \cdot 0.5 \text{ m} \cdot 1 \text{ m}^2 \cdot 100 \text{ m} \\
 &= 2.5 \cdot 10^5 \text{ Joules} \\
 &\ll e_m(t_{mx1}) = 10^5 \text{ Joules}
 \end{aligned}$$

where time $t = 100 \text{ s} = 100 t_c$ (i.e. $T = 100$) is the upper bound of transient time in this research.

This illustration further confirms that the friction dissipation (ED_f) cannot be a relevant factor in a transient pipe flow condition.

6.1.4 Concept of energy balance in a pipe flow

Based on the energy balance characteristics, the transient energy at T_1 can be determined for normal flow closure as:

$$\begin{aligned}
 \{ E_m(T_1) &= E(T_1) - E_w(T_1) \\
 \{ & \\
 \{ &= [E_f - ED_f(T_1)] - [E_{wE}(T_1) - E_{wS}(T_1)] \\
 \{ & \\
 \{ &= E_f - \left[\int_0^{T_1} Q_E(T) dT - \int_0^{T_1} Q_S(T) dT \right]
 \end{aligned} \tag{6.5}$$

where $ED_f(T_1) = 0$.

The relation of Eq. 6.5 can be divided into two components for consideration. These are:

- i) The internal energy $E(T)$; and
- ii) the mechanical energy transfer $E_w(T)$.

6.1.4.1 Internal energy in a frictional system

In a non-frictional system ($H_f = 0$), the internal energy, $E(T)$ is a constant and zero value during the whole transient period.

However, the actual magnitude of $E(T)$ in a frictional system ($H_f > 0$) depends on the initial steady friction energy (E_f), as well as the unsteady friction dissipation (ED_f). Since the latter energy is usually insignificant, the internal energy can be considered to be a constant value. This means that:

$$\begin{aligned} E(T_1) &= E_f - ED_f(T_1) \\ &= E_f \end{aligned} \quad 6.5a$$

6.1.4.2 Mechanical energy transfer in a frictional system

In an instantaneous flow closure ($T_d = 0$), the netflux is equal to the outflux in a pipe system. This means that:

$$E_w(T) = E_{wm}(T)$$

However, for a normal flow closure ($T_d > 0$), the mechanical energy transfer becomes:

$$\begin{aligned} \{ E_w(T_1) &= E_{wm}(T_1) - E_{ws}(T_1) \\ \{ & \\ \{ & \\ \{ &= \int_0^{T_1} Q_m(T) dT - \int_0^{T_1} Q_s(T) dT \\ \{ & \end{aligned} \quad 6.5b$$

6.1.5 Discussion of energy relation

The analysis conducted in this section concluded⁴ that the effect of friction dissipation (ED_z) is so small as to be negligible for general transient flows.

In a frictional system the existence of both non-frictional and frictional energy should be the governing factors in the mechanism of energy balance (Eq. 6.5b). One of the problems arising herein is to determine the relevance of the friction affecting this (non-dissipating) energy transfer phenomenon. This is discussed in the next section.

6.2 Friction Transfer in Transient Flow

Energy transfer related to the existence of friction is defined as friction transfer (ET_z). For clarity it can be further sub-defined separately, one definition is given for surge ($0 < T < T_{mx1}$) and one for the decaying period ($T > T_{mx1}$).

6.2.1 Friction transfer during surge period ($0 < T < T_{mx1}$)

An obvious impact of the existence of friction is the increase in netflux (E_w) during the surge period. To assess the netflux between frictional and non-frictional conditions, a term friction surge transfer (ET_{zs}), is defined in this research as:

$$ET_{zs} = E_w(T_{mx1}, H_z > 0) - E_w(T_{mx1}, H_z = 0) \quad 6.6$$

where $0 < T < T_{mx1}$ and $T_{mx1}(H_z = 0) < T_{mx1}(H_z > 0)$.

6.2.1.1 Instantaneous flow closure ($S = \infty$)

The transient history of an instantaneous flow closure event has been traced for various frictional conditions and is shown in Figs. 6.1b and 6.1c.

1) Friction transfer and maximum surge energy for $H_z < 1$ ---

i) Friction surge transfer (ET_{zs}) --- Without the effect of wave reflection the flow energy netflux (E_w) is exactly the same as for the non-frictional condition. Based on the relation given by Eq. 6.6, this implies that:

$$ET_{zs}(H_z < 1) = 0 \quad 6.7$$

where $T_{mx1} = 1$.

ii) Maximum surge energy (E_{mx1}) --- Accordingly, the maximum surge energy (E_{mx1}) generated by an instantaneous flow closure is linearly proportional to the steady friction head (H_z). This can then be expressed as:

$$\begin{aligned} E_{mx1} &= E_z - E_w(T_{mx1}) \\ &= E_z - 1 \\ &= 0.5 * H_z - 1 \end{aligned} \quad 6.8$$

where $H_z < 1$.

2) Friction surge transfer (ET_{zs}) with $H_z > 1$ ---

i) Friction surge transfer (ET_{zs}) --- Following the instant of flow closure the pressure at the upstream source, after dropping a Joukowsky head, is still higher than the head at the pipe-end (i.e. $h_z - h_e > 0$). Even though the flow rate at the pipe-end (Q_m) is rapidly decreasing, the flow energy netflux is still outward at $T > 1$. At this point the friction surge transfer in this high friction system is:

$$ET_{zs}(H_z > 1) > 0 \quad 6.9$$

where $T_{mx1} > 1$.

For the case of $H_f = 5$, the friction surge transfer is:

$$ET_{f=5}(H_f=5) = 2.6 - 1 = 1.6 > 0$$

where $T_{mx1} = 4.5 > 1$.

ii) Maximum surge energy (E_{mx1}) ---

Accordingly, the maximum surge energy (E_{mx1}) in a high frictional system becomes:

$$\begin{aligned} E_{mx1} &= E_f - E_w(T_{mx1}) & 6.10 \\ &= E_f - E_w(1) - E_w(1 < T < T_{mx1}) \\ &= E_f - 1 - E_w(1 < T < T_{mx1}) \\ &= F(H_f) \end{aligned}$$

where $H_f > 1$. The netflux, $E_w(1 < T < T_{mx1})$ cannot be readily estimated but is also governed by the friction head (H_f).

Based on the results of the present numerical analyses ($T_q = 0$, in Fig. 6.7), maximum surge energy (E_{mx1}) is found to be the reciprocal of friction head (H_f), expressed as:

$$E_{mx1} = -e^{-0.693} / H_f \quad 6.11$$

where $H_f > 1$.

6:2.1.2 Friction transfer for normal closure ($T_q > 0$)

Three types of flow closure are discussed below for $H_f = 1$ and $T_q = 4$. Some energy concepts defined previously in non-frictional system are also referred here for discussion.

1) Uniformly decelerating flow ($S = 1$) ---

As shown in Fig. 6.2a, in the case of $H_f=1$ the pipe-end flow closes at approximately the same time as the source flow, (i.e. $T_{m1} = T_{mx1} = 4$). It should be noted that the effect

of periodic fluctuations appears at time $T > 2$. The frictional effect on the transient during the surge period ($T < T_{mx1}$) can be divided into two components for discussion purposes.

1) Non-frictional transfer (ET_n) --- In a non-frictional pipe system ($H_z = 0$), the non-friction transfer is less than zero and can be written:

$$\begin{aligned} ET_n(H_z=0) &= E_{wn}(T_{ns} < T < T_{n1}) - E_{wn}(T_{ns} < T < T_d) \\ &= E_{wn}(2 < T < 3) - E_{wn}(2 < T < 4) \\ &= -0.25 \end{aligned}$$

where $T_{mx1} = T_{ns} = 2$.

This negative transfer value does not affect the transient behavior during the surge period ($T < T_{mx1}=2$), as the periodic fluctuation can be formed at $T < 2$.

For the present frictional system ($H_z = 1$), this non-friction transfer (located in the lower part of the shaded area in Fig. 6.2a) diminishes to zero as:

$$ET_n(H_z=1) = 0$$

where $T_{mx1} = T_{ns} = 4$.

The existence of periodic fluctuations may still affect the transients during the surge period ($0 < T < T_{mx1}=4$), though the non-friction transfer is zero. This effect is represented in the form of friction surge transfer as discussed next.

ii) Friction surge transfer (ET_{fs}) --- As shown in the upper part of the shaded area in Fig. 6.2a, the friction surge transfer in the present case ($H_z = 1$) is:

$$\begin{aligned}
 ET_{zs}(H_z=1) &= E_w(T_{mx1}, H_z=1) - E_w(T_{mx1}, H_z=0) \\
 &= E_w(4, H_z=1) - E_w(2, H_z=0) \\
 &= 0.45 \\
 &< E_z = 0.5
 \end{aligned}$$

where $T_{mx1}(H_z=1) = 4$.

With the existence of alternating fluctuations, the friction surge transfer is less than the steady friction energy (i.e. $ET_{zs} < E_z$). This fact suggests that the introduction of friction reduces the maximum surge energy as:

$$- E_{mx1}(H_z=1) = 0.20 < - E_{mx1}(H_z=0) = 0.25$$

2) Decreasing decelerating flow ($S < -1$) ---

As shown in Fig. 6.3a, when the pipe-end flow closes at $T_{z1} = 3$ the frictional effect on transients during the flow closure is divided for discussion purposes.

1) Non-friction transfer (ET_z) --- In a non-frictional system ($H_z = 0$) the surge energy is independent of the non-friction transfer, which is:

$$\begin{aligned}
 ET_z(H_z=0) &= E_{ws}(T_{zs} < T < T_{z1}) - E_{ws}(T_{zs} < T < T_d) \\
 &= E_{ws}(1.7 < T < 2.2) - E_{ws}(1.7 < T < 4) \\
 &= -0.13
 \end{aligned}$$

where $T_{zs} = T_{mx1} = 1.7 < 2$. This also shows that the periodic fluctuation can not be formed during the surge period.

In the present frictional system ($H_z = 1$), the above non-friction transfer value is drastically reduced (Fig. 6.3a) as:

$$\begin{aligned}
 ET_z(H_z=1) &= E_{ws}(2.8 < T < 3) - E_{ws}(2.8 < T < 4) \\
 &= -0.03
 \end{aligned}$$

where $T_{zs} = 2.8$ and $T_{m1} = 3$.

In both cases, the non-friction transfer does not increase surge energy after the time of flow interception (T_{zs}). However, the existence of periodic fluctuation may affect the surge transient as discussed next.

ii) Friction surge transfer (ET_{zs}) --- As shown in the top part of the shaded area in Fig. 6.3a, the friction surge transfer in the present case is:

$$\begin{aligned} ET_{zs}(H_z=1) &= E_w(T_{m1}, H_z=1) - E_w(T_{m1}, H_z=0) \\ &= E_w(2.8, H_z=1) - E_w(1.7, H_z=0) \\ &= 0.39 \\ &< E_z = 0.5 \end{aligned}$$

where $T_{m1}(H_z=1) = 2.8$.

After time $T > 2$, the alternating fluctuation appears and consequently reduces the friction surge transfer. The fact that the friction surge transfer is less than the friction energy (i.e. $ET_z < E_z$) suggests that the maximum surge energy is reduced by the introduction of friction, as shown in Fig. 6.3b, where:

$$- E_{m1}(H_z=1) = 0.27 < - E_{m1}(H_z=0) = 0.38$$

In the above two frictional systems (i.e. $S = 1$ and $S = -2$), the maximum magnitude of surge energy is affected by the alternating periodic fluctuations, since $T_{m1} > 2$. This is not the case in a non-frictional system, in which the maximum surge energy occurs prior to the occurrence of periodicity (i.e. $T_{m1} < 2$). As observed in these two frictional systems, the friction surge transfer is always less than the steady friction energy with the existence of alternating periodic fluctuations. Therefore, it can be concluded that:

$$\{ ET_{zs}(T_{mx1}) < E_z$$

$$\{$$

6.12

$$\{ E_{mx1}(H_z > 0) < E_{mx1}(H_z = 0)$$

where $S < 1$.

3) Increasing deceleration flow ($S > 1$) ---

The example case of $S = 2$ and $T_a = 4$, is adopted, representing the increasing deceleration flow with additive fluctuations. As shown in Fig. 6.4a, the pipe-end flow closes at $T_{m1} = 4.5$ for the present frictional system ($H_z = 1$). The frictional effect is divided into two conditions for the discussion purpose.

i) Non-friction transfer (ET_z) --- The non-friction transfer is zero in both non-frictional and frictional systems, since the maximum surge energy does not occur after the end of flow closure (i.e. $T_{mx1} > T_a = 4$). This means:

$$ET_z(H_z = 0) = ET_z(H_z > 0) = 0$$

where $S = 2 > 1.7$, located at the zone of additive fluctuations.

It should be noted that the effect of periodic fluctuations should exist during the surge period ($T > T_{mx1}$), after the time $T > 2$. In order to assess this effect, a comparison is first made between the increasing and decreasing deceleration flows with $H_z = 0$ and $T_a = 4$:

$$\{ ET_z(S = +2) = 0 \text{ and}$$

$$\{$$

$$\{ ET_z(S = -2) = -0.13,$$

This shows that the periodic additive fluctuations in flow with $S = 2$ has already completed its potential destruction on the transient energy during the surge

period. No further energy decay due to this non-friction transfer occurs after the end of first surge period ($T > T_{mx1}$).

This results:

$$-E_{mx1}(S=+2) = 0.25 < -E_{mx1}(S=-2) = 0.38; \text{ and}$$

$$-E_{mx1}(S=2) = E_{mx2}(S=2) = 0.25$$

where an equivalent of surge energy reduction (i.e. $E_{mx1}(S=+2) - E_{mx1}(S=-2) = -0.13$) is contained in the periodic additive fluctuations occurring during the surge period for $S = 2$.

With the introduction of friction, this destructive effect should still exist with different degree, since the maximum surge energy occurs at $T_{mx1} > T_a > 2$.

ii) Friction surge transfer (ET_{zs}) --- An obvious impact of the existence of friction is its damping effect on the periodic fluctuations. This can be visually seen for all three types of flow closure (Figs. 6.2a - 6.4a).

Since the effect of periodic fluctuations ($S > 1.7$) is additive, the potential destructive effect of the periodic fluctuations in a frictional system should therefore be reduced for flows with $S = 2$. This is opposite to those cases of alternating fluctuation. During the surge period ($T < T_{mx1}$) the destructive effect is changed from an equivalent energy reduction of -0.13 ($H_z=0$) to that of -0.04 ($H_z=1$), which results in the difference:

$$ET_{zs}(H_z=1) - E_z = 0.09.$$

In contrast to the above two cases ($S = 1$ and -2), the friction surge transfer is actually greater than the steady friction energy, resulting in (Fig. 6.4b):

$$\begin{cases} ET_{xx} = > E_x \\ \{ \\ \{ -E_{mx1}(H_x > 0) > -E_{mx1}(H_x = 0) \end{cases} \quad 6.13$$

where $S' > 1.7$.

6.2.2 Friction transfer during decaying period ($T > T_{mx1}$)

6.2.2.1 Definition of ET_x for $T > T_{mx1}$

Analysis in Ch. 5.0 indicated that in a non-frictional system the energy reversal (E_{xx}) should be twice that of the flow closure energy (E_q). However, due to the existence of friction transfer phenomena, this is not true in a frictional system.

The friction decay transfer, $ET_{xD}(T_{mx1} < T < T_{mx2})$, during the counter-surge period is defined as:

$$ET_{xD}(T) = E_q(T_{x1}) - E_{xx}(T) / 2 \quad 6.14$$

where $T_{mx1} < T < T_{mx2}$.

The magnitude of friction decay transfer for the subsequent decaying period (e.g. $T_{mx2} < T < T_{mx2}+2$, $T_{mx2}+2 < T < T_{mx2}+4$, ...) can also be estimated in the same manner by changing to the time domain.

6.2.2.2 Friction decay transfer for instantaneous closure

As shown in Fig. 6.1b, the damping effect due to the existence of friction occurs after the end of the surge period ($T > T_{mx1}$). In the presence of friction, the rectangular shaped pipe-end flow movement (at $H_x = 0$) would be changed to a parabolic one, with a decay in the energy reversal.

In the case of $H_x = 1$, the flow closure energy $E_q = 1$ remains the same as for the non-frictional condition.

However, the energy reversal is reduced to $E_{re} = 0.79$ during the counter-surge period. The friction decay transfer is thus estimated as:

$$\begin{aligned} ET_{FD}(T) &= E_Q(T_{m1}) - E_{re}(T) / 2 \\ &= 1 - 0.79 / 2 \\ &= 0.605 \end{aligned}$$

where $T_{m1} < T < T_{m2}$.

The existence of friction generally functions as a damper to accelerate the decay of transient energy. Hence, in the absence of non-friction transfer ($ET_m=0$) the effect of friction on the maximum counter-surge energy can be expressed as:

$$E_{m2}(H_z > 0) < E_{m2}(H_z = 0) \quad 6.15$$

6.2.2.3 Friction decay transfer of normal flow closure

Three cases of typical flow closure are discussed herein, including $S = 1, -2$ and 2 for $H_z = 1$ and $T_Q = 4$.

1) Uniformly decelerating flow ($S = 1$) ---

As shown in Fig. 6.2a, the flow closure energy (E_Q) and energy reversal (E_{re}) become:

$$\begin{aligned} E_Q(T_{m1}) &= E_{we}(T_{m1}) - E_{ws}(T_Q) \\ &= E_w(T_{m1}, H_z=0) + ET_{FS}(T_{m1}, H_z=1) \\ &= 0.25 + 0.45 \\ &= 0.70, \text{ and} \end{aligned}$$

$$E_{re}(T_{m1} < T < T_{m2}) = 0.36$$

Then the friction decay transfer can be determined as:

$$\begin{aligned} ET_{FD}(T) &= E_Q(T_{m1}) - E_{re}(T) / 2 \\ &= 0.70 - 0.36 / 2 \\ &= 0.52 \end{aligned}$$

where $T_{mx1} < T < T_{mx2}$.

The non-friction transfer (ET_m) has been converted from a negative value ($ET_m = -0.25$ in non-frictional system) to a zero ($ET_m = 0$ in $H_f = 1$). The non-friction transfer $ET_m(H_f=1)$ does not act well as a damper to the transient energy during the counter-surge period as it does in the non-frictional system $ET_m(H_f=0)$. This can be shown as:

$$E_q(H_f=1) = 0.7 \gg E_q(H_f=0) = 0$$

Consequently, the existence of friction actually decelerates the decay of transient energy (Fig. 6.2b), as:

$$E_{mx2}(H_f=1) = 0.16 > E_{mx2}(H_f=0) = 0$$

This can be expressed in a general form as:

$$E_{mx2}(H_f > 0) > E_{mx2}(H_f = 0) \quad 6.16$$

It should be noted that the relation of Eq. 6.16 is induced by the change of non-friction transfer process due to the introduction of friction, $ET_m(H_f > 0)$. The existence of friction decay transfer (e.g. $ET_{fD} = 0.52$) still acts as a transient damper, if the transient energy exists at the end of flow closure period (i.e. $E_q = 0.7 > 0$). This can be seen as:

$$E_{mx2}(H_f=1) = 0.16 < E_{mx1}(H_f=1) = 0.2$$

It can be written in a general form as:

$$E_{mx2}(H_f > 0) < E_{mx1}(H_f > 0) \quad 6.16a$$

2) Flow with decreasing deceleration ($S < -1$) ---

As shown in Fig. 6.3a, the flow closure energy (E_q) and energy reversal (E_{rm}) are:

$$\begin{aligned}
E_Q(T_{E1}) &= E_{WE}(T_{E1}) - E_{WE}(T_Q) \\
&= E_W(T_{MX1}, H_F=0) + E_{TF}(T_{MX1}, H_F=1) + E_{TF}(H_F=1) \\
&= 0.38 + 0.39 - 0.03 \\
&= 0.74, \text{ and}
\end{aligned}$$

$$E_{FE}(T_{MX1} < T < T_{MX2}) = 0.44$$

The friction decay transfer becomes:

$$\begin{aligned}
E_{TFD}(T) &= E_Q(T_{E1}) - E_{FE}(T) / 2 \\
&= 0.74 - 0.44 / 2 \\
&= 0.52
\end{aligned}$$

where $T_{MX1} < T < T_{MX2}$.

For this case of $H_F=1$ the non-friction transfer has been diminished to $E_{TF} = -0.03$ from the $E_{TF} = -0.13$ at $H_F=0$. Consequently, the damping effect during the counter-surge period decreases due to the change of this non-friction transfer as:

$$E_Q(H_F=1) = 0.74 \gg E_Q(H_F=0) = 0.25$$

On the other hand, the transient decay increases due to the the introduction of friction as the friction decay transfer becomes:

$$E_{TFD}(H_F=1) = 0.52 \gg E_{TFD}(H_F=0) = 0$$

In this case of $S = -2$ and $T_Q = 4$, the overall effect of the frictional energy during the counter-surge period is positive on transient damping. This can be seen as:

$$E_{MX2}(H_F=1) = 0.18 < E_{MX2}(H_F=0) = 0.25$$

which suggests the dominance of friction decay transfer over the change of non-friction transfer during the counter-surge period.

The existence of friction on reducing the counter-surge energy can be found in the general expression given by Eq. 6.15.

3) Flow with increasing deceleration ($S > 1$) ---

As shown in Fig. 6.4a, the flow closure energy (E_Q) and energy reversal (E_{RE}) are:

$$\begin{aligned} E_Q(T_{E1}) &= E_{WE}(T_{E1}) - E_{WS}(T_Q) \\ &= E_W(T_{MX1}, H_F=0) + ET_F(T_{MX1}, H_F=1) \\ &= 0.25 + 0.59 \\ &= 0.84, \text{ and} \end{aligned}$$

$$E_{RE}(T_{MX1} < T < T_{MX2}) = 0.58.$$

The friction decay transfer during the counter-surge period is:

$$\begin{aligned} ET_{FD}(T) &= E_Q(T_{E1}) - E_{RE}(T) / 2 \\ &= 0.84 - 0.58 / 2 \\ &= 0.55 \end{aligned}$$

where $T_{MX1} < T < T_{MX2}$.

In this case of $S = 2$ and $T_Q = 4$ the non-friction transfer (ET_F) in the counter-surge period does not affect the decay of transient energy, for both non-frictional and frictional conditions. The friction decay transfer, $ET_{FD}(T_{MX1} < T < T_{MX2})$ is the only course for the decay of transient energy, as governed by the relation given by Eq. 6.15.

6.2.3 Discussion of friction transfer

An energy transfer mechanism related to the friction, namely friction surge (ET_{FS}) and decay (ET_{FD}) transfer has been established to assess the existence of friction as it affects transient behavior during the surge ($T > T_{MX1}$) and decay ($T > T_{MX1}$) period respectively.

It is well acknowledged that the transient energy is usually reduced by the inclusion of friction, for the surge (Eq. 6.12) counter-surge and decaying (Eq. 6.15) periods. This general property of friction transfer has been appropriately interpreted by the concept of energy balance.

However, results opposite to the above can be found as expressed by Eqs. 6.13 and 6.16 for the surge and counter-surge periods respectively. In both periods the maximum transient energy (E_{mx1} or E_{mx2}) can actually increase due to the existence of friction. The effect of periodic fluctuations is the cause of this unconventional (but logical and common in many practical applications) transient behavior.

In this section the friction transfer phenomenon has been analyzed on basis of some typical cases. It remains a question as to how this phenomenon is to be applied to other frictional and flow closing conditions. This is undertaken in the next section.

6.3 Parameters Governing Friction Transfer

As directed by the previous dimensional analysis (Eq. 4.5), the relation between this new transient property (ET_z) and its governing parameters can be established below as:

$$ET_z = F(H_z, S, T_a, T) \quad 6.17$$

A systematic analysis is conducted in this section to evaluate the relevance of each individual parameter in the equation. This analysis can be achieved by varying the value of only one parameter while keeping the other parameters constant.

The present analysis of friction transfer can be divided into two groups:

i) Surge period (ET_{ss} for $T < T_{mx1}$) --- where the analysis focuses on the effect of friction transfer on the maximum surge energy; and

ii) Decay period (ET_{sd} for $T > T_{mx1}$) --- where the attention is on the decay of the elastic energy of the system.

6.3.1 Relevant parameters to surge energy

For the analysis of friction transfer during the surge period the number of governing parameters in Eq. 6.17 can be reduced to three because the time (T) of transient is confined to the surge period. This can be expressed as:

$$ET_{ss} = F(H_f, S, T_d) \quad 6.18$$

where $0 < T < T_{mx1}$.

The results of analysis based on this functional relation are summarized in Fig. 6.5.

6.3.1.1 Friction head (H_f) as a relevant parameter

The relationship of Fig. 6.5a can be expressed as:

$$ET_{ss} = F(H_f) \quad 6.18a$$

where limited analysis of S and T_d has been conducted.

1) Instantaneous flow closure ---

No variation in the friction surge transfer (ET_{ss}) can be observed in a system with $H_f < 1$. However, for $H_f > 1$ the friction surge transfer increases rapidly with friction (H_f).

The friction surge transfer remains less than its steady friction energy under any condition. The relation of $ET_{ss}(H_f) < E_s$ ensures that the maximum surge energy

decreases with the friction in an instantaneous flow closure as indicated in Eq. 6.12.

2) Normal flow closure ---

The values of $S = 2$ and $T_q = 4$ are selected as constant flow closing characteristics. As observed in the Fig. 6.5a, the friction surge transfer exceeds the steady friction energy, i.e. $ET_{ss} > E_s$ for $H_x < 1.25$. This results in the maximum surge energy increasing with friction, as indicated in Eq. 6.13.

In the discussion of two other relevant parameters, T_q and S in Eq. 6.18, $H_x = 1$ is selected for the significance of its frictional effect.

6.3.1.2 Time of flow closure (T_q)

The relation of Fig. 6.5b can be expressed as:

$$ET_{ss} = F(T_q) \quad 6.18b$$

where $S = 2$ and $H_x = 1$.

The increment of friction surge transfer rises rapidly when the time of flow closure increases from the instantaneous flow closure to about $T_q = 4$. At that point it begins to decrease gradually with further increments in T_q .

The relation of $ET_{ss}(T_q) > E_s$ can be observed for $T_q > 3$, providing there is an opportunity for the positive effect of friction on the increase of maximum surge energy, as indicated in Eq. 6.13.

Because of its effectiveness in transient response, the value of $T_q = 4$ is selected during the analysis of other two parameters, (H_x) and (S) in Eq. 6.18.

6.3.1.3 Shape factor of flow closure (S) ---

The relation of Fig. 6.5c is expressed as:

$$ET_{zs} = F(S) \quad 6.18c$$

where $H_z = 1$ and $T_q = 4$.

1) Friction surge transfer greater than friction energy ---

As shown in Fig. 6.5c, the friction surge transfer would exceed its steady friction energy for a shape factor:

$$ET_{zs}(S) > E_z$$

where $1.2 < S < 2.7$.

The parameter S reaches its maximum of $ET_{zs} = 0.60$, at the boundary of positive additive fluctuation, i.e. $S = 1.7$.

The damping of periodic fluctuations by friction during the surge period contributes to $ET_{zs} > E_z$ as expressed by Eq. 6.13.

The selection of $S = 2$ in the analysis of two other parameters (H_z) and (T_q) is based on the potential of $ET_{zs} > E_z$, and this is outside the range of alternating fluctuations. The existence of alternating fluctuations ($S < 1.7$) would create considerable complexity for illustration purposes.

2) Effect of fluctuation-free flow ---

The non-fluctuating flow dominates the transient behavior during the surge period for $|S| > 5$. On the other hand, due to the effect of alternating periodic fluctuations, the friction surge transfer (ET_{zs}) is usually non-symmetrical about the axis of $S = 1$. This means that:

$$ET_{zs}(+S) > ET_{zs}(-S), \text{ for } |S| < S_x \quad 6.19a$$

or

$$ET_{zs}(+S) = ET_{zs}(-S), \text{ for } |S| > S_x \quad 6.19b$$

where $S_x = 5$, for $H_z = 1$ and $T_a = 4$.

6.3.2 Relevant parameters to decaying energy ($T > T_{mx1}$)

This subsection discusses the relationship between friction decay transfer (during the decaying period) and its four governing parameters as:

$$ET_{zD} = F(H_z, S, T_a, T) \quad 6.20$$

The friction decay transfer, $ET_{zD}(T)$, is actually determined in terms of half the surge fluctuation period (e.g. $T_{mx1} < T < T_{mx2}$, $T_{mx2} < T < T_{mx2}+2$, $T_{mx2}+2 < T < T_{mx2}+4$,). The friction ranging from $H_z=0.01$ to $H_z=5$ is selected for analyses of other three parameters, representing very low to very high frictional conditions. Results of the analysis have been summarized in Fig. 6.6.

6.3.2.1 Time of decay (T) as a relevant parameter

Only the case of an instantaneous flow closure is selected for the present discussion. Hence the effect of parameters S and T_a disappears, and the friction decay transfer (ET_{zD}) depends only on the steady friction head (H_z) and the time of the transient (T), expressed as:

$$ET_{zD} = F(H_z, T) \quad 6.20a$$

where $0.01 < H_z < 5$.

As shown in Fig. 6.6a, the friction decay transfer (ET_{zD}) increases rapidly following the end of the first surge period. The $ET_{zD}(T)$ vs. T curve then becomes horizontal after a sufficiently long time. For a friction head below 1 the friction decay transfer will increase rapidly until it reaches the maximum of netflux [i.e. $E_w(T_{mx1}) = 1$].

This means that for $H_z < 2$, the transient energy is theoretically damped out for the following condition:

$$ET_{ED}(T > T_{mx1}) = E_w(T_{mx1}) \quad 6.21a$$

where $\{ E_w(T_{mx1}) = 1 > E_z \text{ for } T_q = 0, \text{ but}$

$$\{ E_w(T_{mx1}) < 1 \text{ for } T_q > 0.$$

Conversely, in the event of high friction (i.e. $H_z > 2$), the stagnant condition will soon be reached because the friction decay transfer equals to its maximum as:

$$ET_{ED}(H_z) = E_z \quad 6.21b$$

where $H_z > 2$.

The above analysis of time (T) of decay indicates that the largest damping capability exists at the beginning of the decay period. Accordingly, the friction decay transfer during the counter-surge period ($T_{mx1} < T < T_{mx2}$) is selected for the later analysis of time (T_q) and shape (S) of flow closure (i.e. Eqs. 6.20b in Fig. 6.6b and Eq. 6.20c in Fig. 6.6c).

6.3.2.2 Time of flow closure (T_q) as a relevant parameter

In this analysis, flow closure with $S = 2$ is used in order to eliminate the effect of the alternating periodic fluctuations. With $S > 1.7$, the effect of non-friction transfer is non-existing during the counter-surge period. Hence Eq. 6.20 can be simplified to:

$$ET_{ED}(T) = F(H_z, T_q) \quad 6.20b$$

where $T_{mx1} < T < T_{mx2}$ and $0.01 < H_z < 5$.

As shown in Fig. 6.6b, the friction decay transfer (ET_{ED}) exceeds its steady friction energy (E_z) if the time (T_q) of

flow closure is small. This suggests that an efficient energy damping occurs in a rapid closure. On the other hand, a very slow damping process dominates the decaying period following the slow flow closure.

In order to observe an effective decaying process of transient energy (i.e. $ET_{\pm D}$ vs. T), the instantaneous flow closure (T_a) is selected.

6.3.2.3 Flow shape factor (S) as a relevant parameter

The evaluation of the shape of flow closure is based on $T_a = 4$, in which a greater effect of transient variation can be observed. Eq. 6.20 can then be simplified to:

$$ET_{\pm D}(T) = F(H_z, S) \quad 6.20c$$

where $T_{mx1} < T < T_{mx2}$ and $0.01 < H_z < 5$.

As indicated in Fig. 6.6c, the smallest effect of friction decay transfer occurs for the case of uniform flow closure, while the most efficient energy damping occurs for the case of an instantaneous flow closure ($S = +\infty$ and $-\infty$).

Comparing the increasing deceleration and decreasing deceleration flows in Fig. 6.6c, the following relationship can be considered as an approximation:

$$ET_{\pm D}(+S) = ET_{\pm D}(-S) \quad 6.22$$

where $|S| > 1.7$.

The absence of alternating periodic fluctuation during the counter-surge period is the reason for its symmetrical similarity.

To obtain a most efficient damping effect, the instantaneous flow closure ($S = +\infty$ or $-\infty$) is selected.

6.3.2.4 Friction (H_z) as a relevant parameter

The effect of friction on the decay of transient energy can be observed in all three parts of Fig. 6.6. Obviously, the capability of transient energy decay increases rapidly with friction in a form of a mechanical transfer process.

6.3.3 Discussion of governing parameters

It has been validated from this section that the friction transfer mechanism in a pipe system is governed by four relevant parameters, namely, time of transient (T), friction (H_z), time (T_d) and shape (S) of flow closure. The energy transfer concepts previously developed in the analysis of various cases have been used for the present interpretation.

The existence of friction affects the transient behavior in two categories. These are: 1) friction surge transfer during the surge period (ET_{zs}); and 11) friction decay transfer during the decaying period (ET_{zd}).

During the present analysis of each specific variable for testing its relevance, the remaining governing variables in the functional relation have been taken as constants. As such, only a limited number of events could be reviewed. A comprehensive transient analysis of frictional pipe systems need to be carried out in which the broad ranged parameters can be inter-related. This will be conducted in the next section.

6.4 Maximum Transient Energy

In general practice information related to maximum transient energy is most desired. With this in mind a comprehensive analysis is conducted to evaluate and quantify the relevance of parameters as directed by the dimensional reasoning in Eq. 4.5b. Restated this is:

$$E_{mx} = F(H_z, S, T_d)$$

6.23

where the characteristics of friction transfer (ET_x) has been represented by the maximum transient energy (E_{mx}).

6.4.1 Scope of analysis

An analysis has been conducted in this section with a wide range of frictional and flow closing parameters for critical transient events.

The range of relevant parameters covers:

- i) $0 < H_x < 10$;
- ii) $-10 < S < 10$; and
- iii) $0 < T_a < 100$.

Three maximum transient events are presented, namely:

- i) Maximum surge energy, E_{mx1} ;
- ii) Maximum counter-surge energy, E_{mx2} ; and
- iii) Peak decaying energy, E_{101} (at $T = 101$).

where the time $T = 101$ is selected herein as an upper bound of the decaying period, in the present research.

Apparently, it is difficult and perhaps confusing to construct a family of family curves containing comprehensive information on all three parameters. In this research, two plots are prepared for each (E_{mx1} , E_{mx2} and E_{101}) of maximum transient events as:

- i) Three cases of the flow closure with $S = 1, -2$ and 2 , are plotted in the figure, representing flows with uniform, decreasing and increasing deceleration respectively.

- ii) The second figure is constructed to indicate the appropriate frictional and flow closing conditions for achieving the same magnitude of the maximum transient energy.

The combination of these two figures should be adequate to present information for the investigation of each specific maximum transient energy as expressed by Eq. 6.23.

For the sake of simplicity the non-frictional transient condition can be obtained from the analysis with $H_z = 0.01$ without inducing significant error.

6.4.2 Maximum surge energy

The maximum surge energy in a frictional system can be expressed as:

$$E_{mx1} = F(H_z, S, T_q) \quad 6.24$$

Results of analysis on the relationship given by Eq. 6.24 can be summarized as follows (see Fig. 6.7 or/and 6.8):

6.4.2.1 Time of flow closure (T_q) as a relevant parameter.

As shown in Figs 6.7 and 6.8, the greater the time of flow closure (T_q), the smaller the maximum surge energy (E_{mx1}) that is induced, with the upper bound of maximum surge energy occurring at $T_q = 0$.

Referring to Fig. 6.5b it is evident that the friction surge transfer (ET_{fs}) is almost constant for time of flow closure (T_q) greater than 4. Then ET_{fs} decreases rapidly for $T_q < 4$. This non-effective or decreasing friction surge transfer property indicates that the non-frictional transient (elastic) behavior during the surge period begins to dominate as the time when the flow closure decreases. At $T_q = 0$, the existence of friction (for $H_z < 1$) is irrelevant to the netflux [i.e. $E_w(H_z < 1) = 1$]. Therefore, the variation of E_{mx1} vs. T_q follows the same direction as the transient behavior in a non-frictional system, in a rapid flow closing event.

6.4.2.2 Shape of flow closure (S) as a relevant parameter

As shown in Fig. 6.8, following the curve of optimal E_{mx1} , generally the higher the absolute value of the shape factor, $|S|$, the greater the maximum surge energy $|E_{mx1}|$ observed.

This can be explained in a similar manner as above. Referring to Fig. 6.5c, the friction surge transfer (ET_{fs}) decreases with the increasing $|S|$, with the highest ET_{fs} at $S = S_x = 1.7$. Consequently non-friction transient (elastic) behavior dictates the relationship between E_{mx1} and S .

6.4.2.3 Friction (H_f) as a relevant parameter

Analysis of friction affecting maximum surge energy has been done with the consideration of flow closing properties (S and T_d). The analysis consists of:

1) Non-existence of periodic fluctuations ---

As shown in all three shapes (S) of flow closure with $T_d < 2$ (Fig. 6.7), maximum surge energy decreases with increasing friction.

It should be noted that for $T_d < 2$ the effect of periodic fluctuations does not exist. Consequently, the relation given by Eq. 6.12 applies.

2) Transient energy increasing with friction ---

As shown in Fig. 6.7 the maximum surge energy actually increases with friction for $S > 1$. The relaxation of periodic additive fluctuations due to the existence of friction is the cause. This event occurs however at sufficiently high friction head.

3) Irrelevance of friction ---

For the condition of inadequate friction head the increasing rate of maximum surge energy is so slow that the frictional

effect can be neglected. This is shown in both Figs. 6.7 and 6.8, for $S > 1$.

4) Transient energy decreasing with friction ---

For the shape (S) of flow closure less than 1, a clear sign of maximum surge energy decreasing with the increment of friction can be observed in Figs. 6.7 and 6.8.

5) Dominance of friction ---

Maximum surge energy is entirely governed by friction, independent of the characteristics of flow closure (S and T_q). This can be observed for high friction head (Figs. 6.7 and 6.8). For this condition the large friction surge transfer has damped out all the periodic fluctuation effect and becomes the dominating energy for the mechanical transfer phenomenon.

6) Optimal shape factor (S^*) ---

The optimal shape of flow closure decreases with the increment of friction, as shown in Fig. 6.8.

Referring to Fig. 6.5c it is evident that the combination of non-friction transfer conversion and relaxation during different periods determines the exact magnitude of S^* . This occurs at $S = S^* < 1.5$.

Results of analysis herein can be used for both optimizing and classifying the friction effects on maximum surge energy. This will be discussed more in detail in Appendix A.4 and A.5.

6.4.3 Maximum counter-surge energy

Maximum counter-surge energy is governed by the same three common parameters as below:

$$E_{mx2} = F(H_z, S, T_q)$$

6.25

Fig. 6.9 illustrates the results of analysis for three shapes of flow closure. Fig. 6.10 locates the regions where a reduction from maximum surge to maximum counter-surge energy can exceed 0.10 to 0.25. Here the counter-surge energy reduction (ER_{mx2}) is evaluated by:

$$ER_{mx2} = 1 - E_{mx2}/E_{mx1} \quad 6.26$$

Results interpreted from Figs. 6.9 and 6.10 are summarized below:

6.4.3.1 General characteristics

Basically, all the transient properties related to the above maximum surge energy are kept in the present counter-surge condition. Some outstanding features related to E_{mx2} are however discussed below.

6.4.3.2 Transient energy increasing with friction

As shown in Figs. 6.9 and 6.10, a rapid rise of counter-surge energy with friction can be observed in the vicinity of uniform flow closure ($S = 1$), for $T_a = N * T_s$.

This transient behavior is caused by the reduction of negative non-friction transfer ($ET_{\pm}$) due to the inclusion of friction, and this has been expressed in Eq. 6.16.

6.4.3.3 Shape factor (S) related to energy reduction

As shown in Fig. 6.10, the energy reduction is much greater for flows with $S > 1$, expressed below:

$$ER_{mx2}(+S) \gg ER_{mx2}(-S) \quad 6.27$$

There is no significant difference in friction decay transfer ($ET_{\pm D}$) between increasing and decreasing deceleration flows (Fig. 6.6c). However, the non-friction

transfer (ET_m) contributes to a large part of the energy decay for flows with $S < 1$ during the counter-surge period.

6.4.3.4 Time closure (T_q) related to energy reduction

As shown in Fig. 6.10, energy reduction (ER_{mx2}) increases with the decreasing time of flow closure, with the greatest value at the instantaneous flow closure (T_q). This finding follows the same relation of friction decay transfer (ET_{FD}), as shown in Fig. 6.6b

6.4.4 Peak decaying energy

Following the first counter-surge period the transient energy in the frictional system still oscillates with a period of $T_s = 4$, and will eventually be damped out depending on the magnitude of friction. The effect of non-friction transfer ceases at $T > T_{mx2}$.

6.4.4.1 Peak decaying energy (E_{101}) at $T = 101$

Peak decaying energy is governed by the same three common parameters, and can be expressed as:

$$E_{101} = F(H_s, S, T_q) \quad 6.28$$

where $T = 101$.

Fig. 6.11 summarizes the analysis of peak decaying energy (E_{101}) at $T = 101$ for $S = 2, 1$, and -2 .

A similar pattern for the energy relationship can be observed in this peak decaying energy as previously described for the maximum counter-surge condition (Fig. 6.9). The major difference is the larger magnitude of the energy reduction in the region of high friction head but small time of flow closure.

6.4.4.2 Peak decaying energy reduction (ER_{101}) at $T = 101$

Fig. 6.12 summarizes the hydraulic conditions with an energy reduction level of 0.50. Two energy reduction guidelines are classified for this analysis.

1) Energy reduction based on E_{mx1} ---

The guideline for estimating the peak decaying energy reduction is based on:

$$ER_{101} = 1 - E_{101}/E_{mx1} \quad 6.29$$

As shown in Fig. 6.12, the comparison between increasing and decreasing deceleration flows follows the same general relation given by Eq. 6.27. This means that:

$$ER_{101}(+S) \gg ER_{101}(-S) \quad 6.30$$

As discussed previously concerning Eq. 6.27 a large portion of the energy decay in Eq. 6.30 is related to the non-friction transfer (ET_m), during counter-surge period.

2) Energy reduction based on E_{mx2} ---

The peak decaying energy reduction can be determined from another form as:

$$ER_{101}^* = 1 - E_{101}/E_{mx2} \quad 6.31$$

As shown in Fig. 6.12, for a flow with $S > 1$, an almost identical pattern of energy reduction based on E_{mx2} (Eq. 6.31) can be found compared to that based only on the E_{mx1} (Eq. 6.29). This means that:

$$ER_{101}^*(S) = ER_{101}(S) \quad 6.32$$

where $S > 1$.

This is due to the fact that friction decay transfer (ET_{FD}) is the only cause of transient energy damping during the counter-surge period.

On the other hand, for a flow with $S < -1$,

$$ER_{101}^*(S) \ll ER_{101}(S) \quad 6.33$$

where $S < -1$.

The difference between Eqs. 6.32 and 6.33 is due to the existence of non-friction transfer (ET_N) during the counter-surge period. The axis of the symmetrical energy reduction pattern (ER_{101}^*) between the increasing and decreasing deceleration flows is located along the optimal flow closing performance curve, which decreases with increasing friction.

6.4.5 Discussion of maximum transient energy

A comprehensive analysis has been undertaken to assess and quantify the relevance of governing parameters for each specific maximum transient energy.

The results of analysis can be effectively interpreted from the non-friction and friction transfer phenomena, developed from the concept of energy balance.

6.5 Conclusions

1) Theoretical Concepts ---

i) The application of the *Energy Balance Concept* in this chapter has proved to be a powerful tool to interpret the transient flow phenomenon in a frictional closed conduit (Eq. 6.5).

ii) The functional relation derived from dimensional reasoning has been proved effective in providing research direction to evaluate phenomena between transient properties (M) and each of the governing

parameters. The four relevant parameters are: i) friction (H_z); ii) time (T_a) and iii) shape (S) of flow closure, and iv) the time of transient (T). The transient properties consist of maximum surge energy (E_{mx1}), maximum counter-surge energy (E_{mx2}) and peak decaying energy (E_T).

iii) The friction dissipation (ED_z), which is in the form of heat exchange, has traditionally been considered as the source of energy decay in transient flows. Though it may still be the prime contributor to energy losses to a slow flow motion (e.g. U - tube and pulsating mass oscillation flows), it does not contribute any significant energy damping in the general transient condition during and after the flow closure.

iv) A new concept, namely the friction transfer (ET_z) has been introduced in this chapter to investigate its effect on transient behavior. The transient energy in a frictional system can actually be damped out by a form of mechanical energy exchange.

v) The magnitude of periodic alternate fluctuations decreases with increasing friction, thus affecting the effect of non-frictional energy transfer (ET_n) during the surge and counter-surge period.

For the case of a highly frictional system (e.g. $H_z \gg 1$), the same transient energy (E_{mx}) would be induced, independent of any flow closing characteristics (S and T_a).

2) Effect of friction during the surge period ($T < T_{mx1}$) ---

The existence of friction in association with periodic fluctuations affects the surge behavior in several ways.

1) Instantaneous flow closure ($S \Rightarrow \infty$) --- In the absence of periodic fluctuations, the friction surge

transfer (ET_{fs}) has no effect on the transients for $H_f < 1$. Maximum surge energy is simply determined as:

$$E_{mx1} = 1 - E_f.$$

ii) Decreasing deceleration flows ($S < -1$) --- The existence of friction converts the non-friction transfer from counter-surge to surge period, resulting in a friction surge transfer (ET_{fs}) which is smaller than its steady friction energy (E_f). This also means that the friction reduces maximum surge energy.

iii) Increasing deceleration ($S > 1$) --- The effect of periodic fluctuations during the surge period can be reduced with the inclusion of friction, resulting in a friction surge transfer greater than its steady friction energy. The friction can increase the maximum surge energy for this condition.

For most frictional and flow closing conditions the relationship between above increasing and decreasing deceleration flows implies that:

$$E_{mx1}(+S) > E_{mx1}(-S)$$

where $H_f < 0.05$ for $T_q > 4$, and $H_f > 0.012$ for $T_q < 100$ as shown in Fig. A.3.

iv) The optimal shape factor (S^*) for providing the best hydraulic performance decreases with increasing friction (H_f) and time of flow closure (T_q), as shown in Fig. 6.8.

v) The effect of friction on maximum surge energy (E_{mx1}) can be classified into four zones: the dominant, positive, negative, and nil zones, depending largely on the flow closing characteristics.

The last two points will be discussed more in detail in Appendix A.4 and A.5.

3) Effect of friction during the decay period ($T > T_{mx1}$) ---

i) Friction decay transfer ($ET_{\pm D}$) --- The difference in friction decay transfer ($ET_{\pm D}$) between increasing and decreasing deceleration flows can be considered small during the decaying period. This implies that:

$$ET_{\pm D}(+S) = ET_{\pm D}(-S)$$

ii) Non-friction transfer ($ET_{\pm}$) --- However, the existence of non-friction transfer ($ET_{\pm}$) contributes an additional energy decay in the region of $S < 1$, during the decaying period.

iii) Peak decaying energy (E_T) --- The maximum rate of energy decay (including E_{mx2} and E_{101}) for flows with decreasing deceleration is usually less than that for increasing decelerations. This implies that:

$$E_T(+S) > E_T(-S), \text{ where } T > T_{mx1}.$$

iv) Uniformly decelerating flows ($S = 1$) --- In the vicinity of $S = 1$ and $T_a = N * T_s$, the existence of friction may increase the transient energy from the non-frictional condition.

In contrast to current theories on the effect of friction the maximum transient energy (including surge (E_{mx1}), counter-surge (E_{mx2}) and decaying (E_T) energy) may actually increase with the existence of friction. This condition can be explained neither from the traditional friction dissipation concept ($ED_{\pm}$), nor from the conventional pressure response method of interpretation.

4) Practical applications ---

In addition to the theoretical background, these transient properties occurring during the flow closing period are also

potentially important to engineering practice. Examples of possible applications are:

i) Rigid water column theory (RWCT) --- An entirely new theory has been established to replace the classic RWCT. Its effectiveness in applications to the hypothetical flow patterns as well as to other common surge control devices has also been demonstrated. This will be presented, in Appendix B.

ii) Optimal maximum surge energy --- Operating hydraulic conditions have been derived first for hypothetical flows and then for common surge control devices. This will be presented in Appendix A.4.

iii) Friction classification --- Regions for the frictional effect on the maximum surge energy have been established for various flow conditions and for all four common control devices. This will be presented in Appendix A.5.

Table 6.1 Summary in Steady Friction (factor) Analysis

Fric. H_f	Shape S	Time T_G	Maximum transient energy					
			Surge		Counter-surge		peak-decay	
			E_{mx1}	T_{mx1}	E_{mx2}	T_{mx2}	E_T	T
0	-	0	-1	1	1	3	-1	5
.1			-.95	1	.86	3	-.80	5
1			-.50	1	.29	3	-.21	5
5			-.10	4.5	.07	6.5	-.05	8.5
0	1	4	-.25	2	0	---	0	---
.1			-.23	2.1	.03	5.5	-.30	7.5
1			-.20	4	.16	6	-.14	8
5			-.10	7	.07	9	-.05	11
0	-2	4	-.39	1.7	.26	4	-.26	6
.1			-.37	1.8	.24	4	-.23	6
1			-.26	2.8	.18	5	-.15	7
5			-.10	6	.07	8	-.05	10
0	2	4	-.25	4	.25	6	-.25	8
.1			-.28	4	.27	6	-.26	8
1			-.34	4.5	.24	6.5	-.20	8.5
5			-.10	7.5	.07	9.5	-.05	11.5

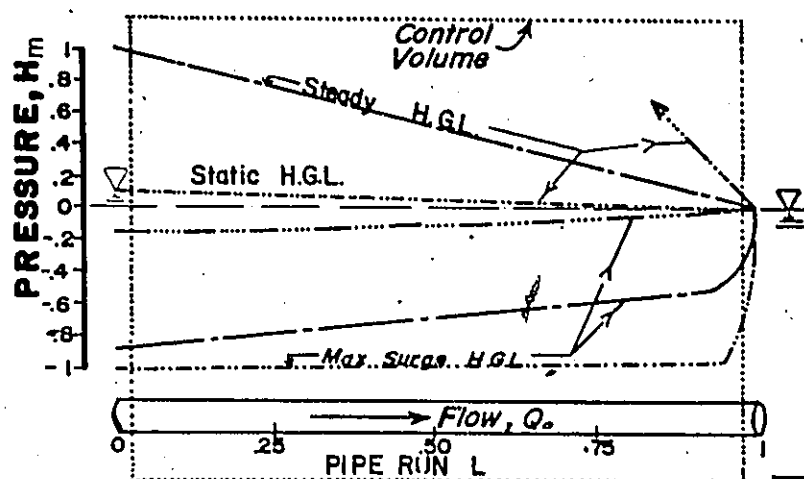
Table 6.2 Summary of Friction Dissipation (ED_s)

H_s	E_s	T_a	S	$0 < T < T_{mx1}$	$T_{mx1} < T < T_{mx2}$
1	0.5	0	-	0.00025	0.0004
5	2.5	0	-	0.0035	0.00001
1	0.5	100	1	0.025	0.00015

Table 6.3 Summary of Friction Transfer, ET_s

H_s	S	T_a	Surge Transfer $0 < T < T_{mx1}$		Decay Transfer			
			ET_{sg}	E_d	$T_{mx1} < T < T_{mx2}$	$T_{mx2} < T < T_{mx2} + 2$	ET_{sg}	ET_{sd}
1	-	0	0	1	0.79	0.61	0.50	0.75
5	-	0	1.6	2.60	0.14	2.53	0.06	2.57
1	1	4	0.45	0.70	0.36	0.52	0.30	0.55
1	-2	4	0.38	0.74	0.44	0.52	0.32	0.58
1	2	4	0.59	0.84	0.58	0.55	0.44	0.62

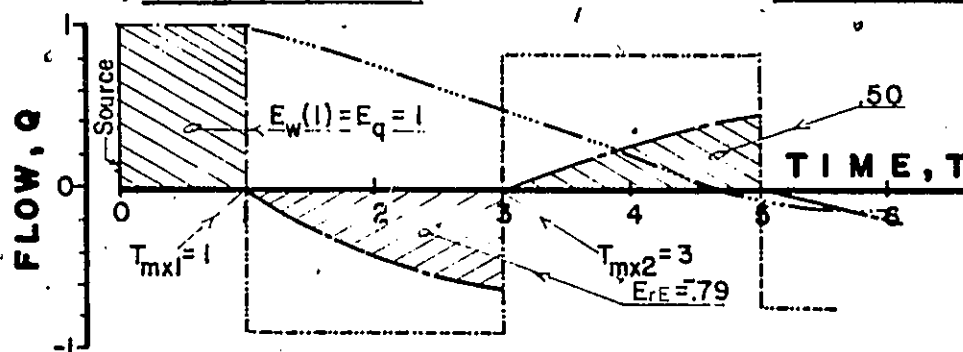
a) Schematic of a FRICTIONAL PIPELINE



LEGEND

—	$H_f = .1$
- - -	$H_f = 1$
· · ·	$H_f = 5$

b) Flow (Pipe-end)



c) Energy

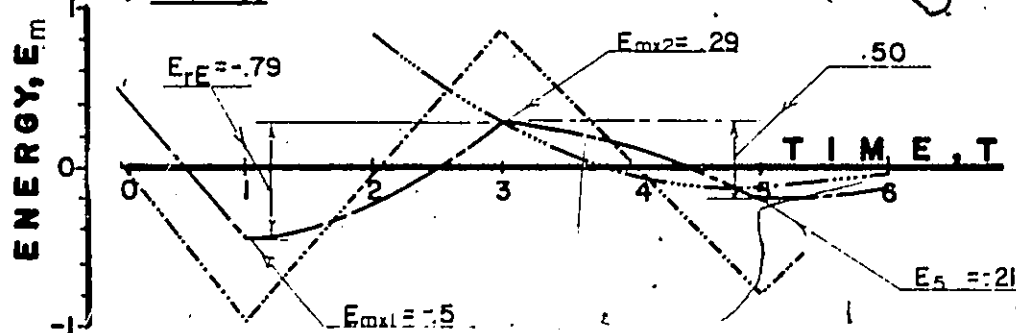
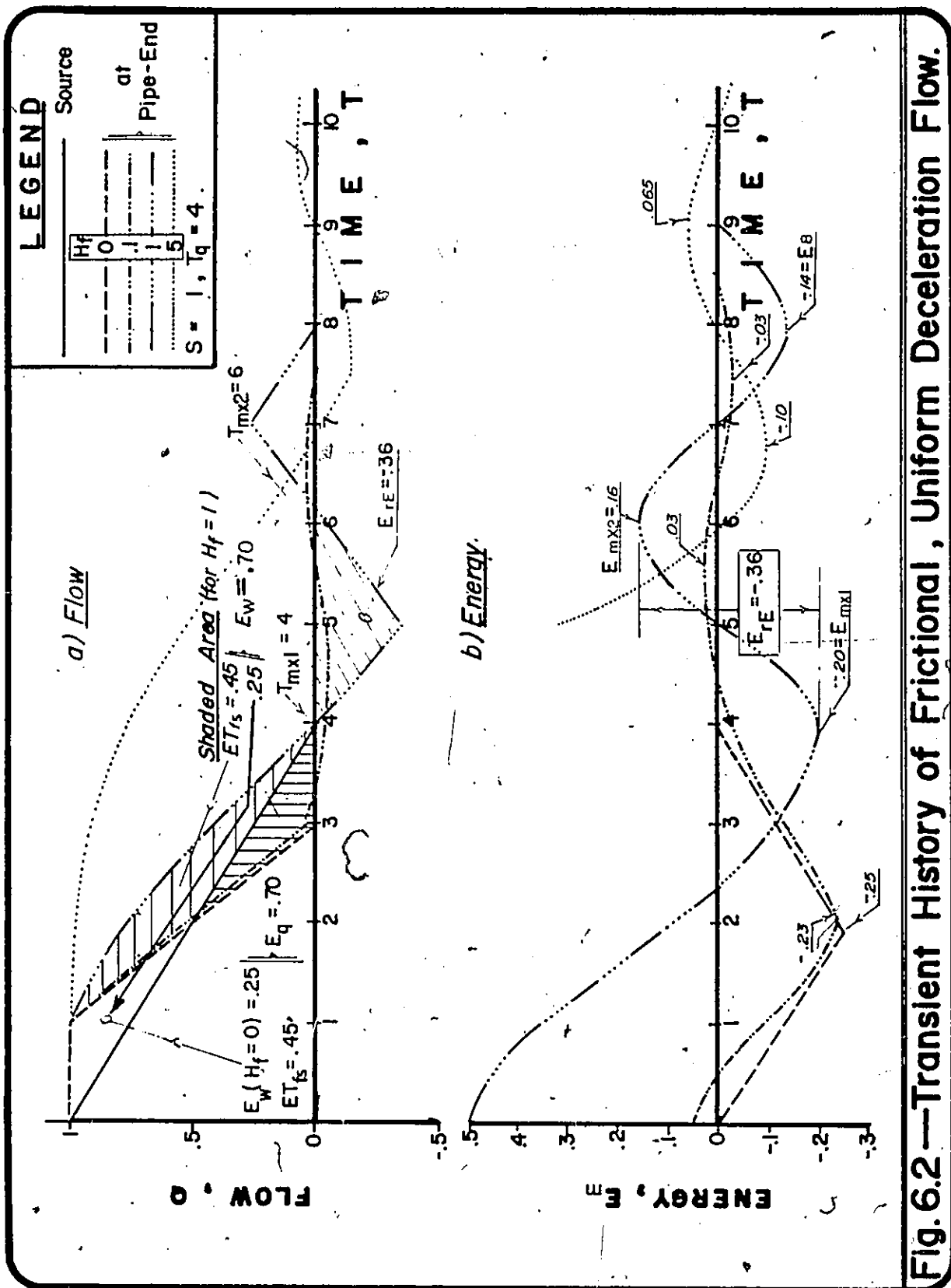


Fig. 6.1—Transient History of Frictional, Instant Flow Closure.



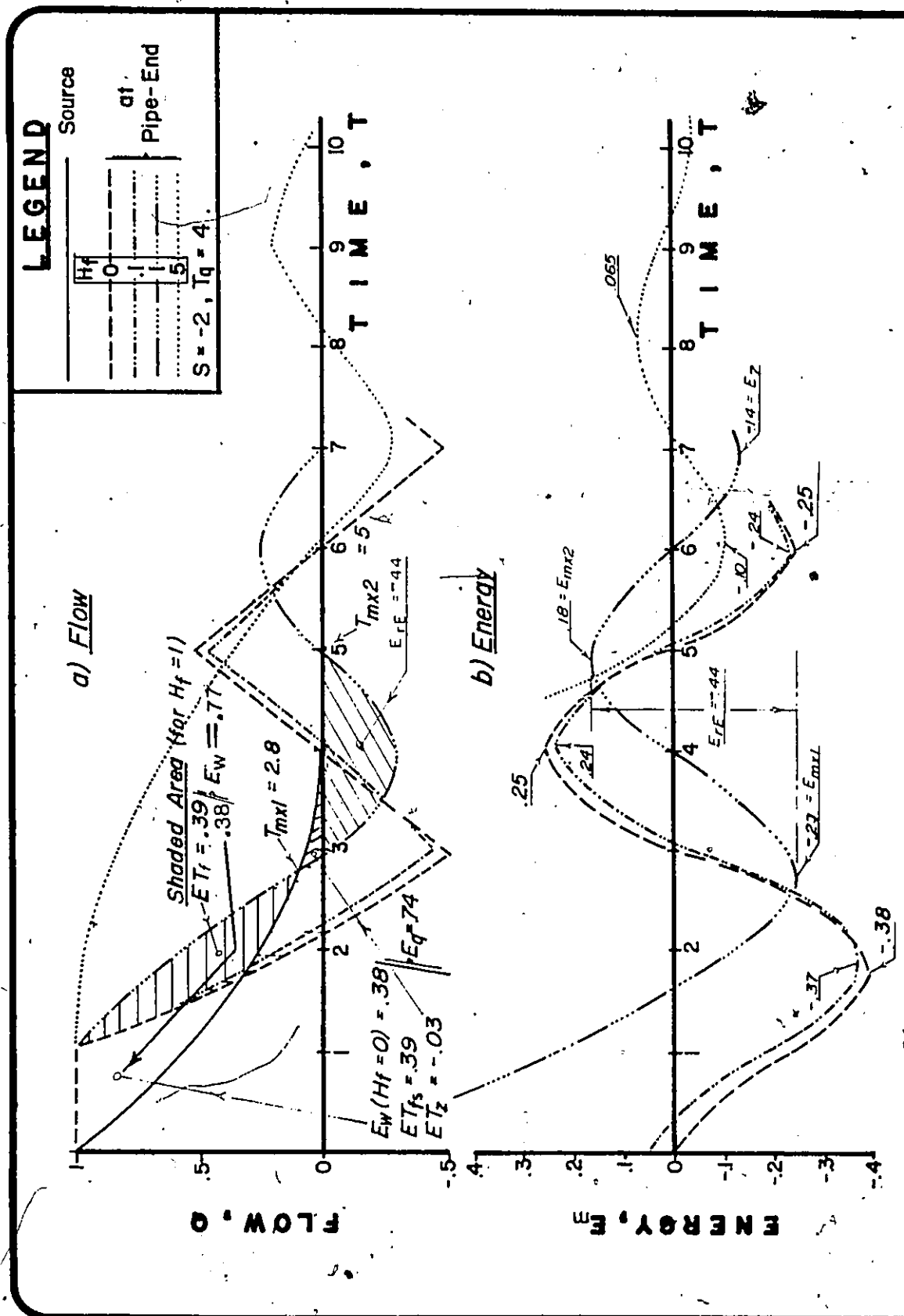
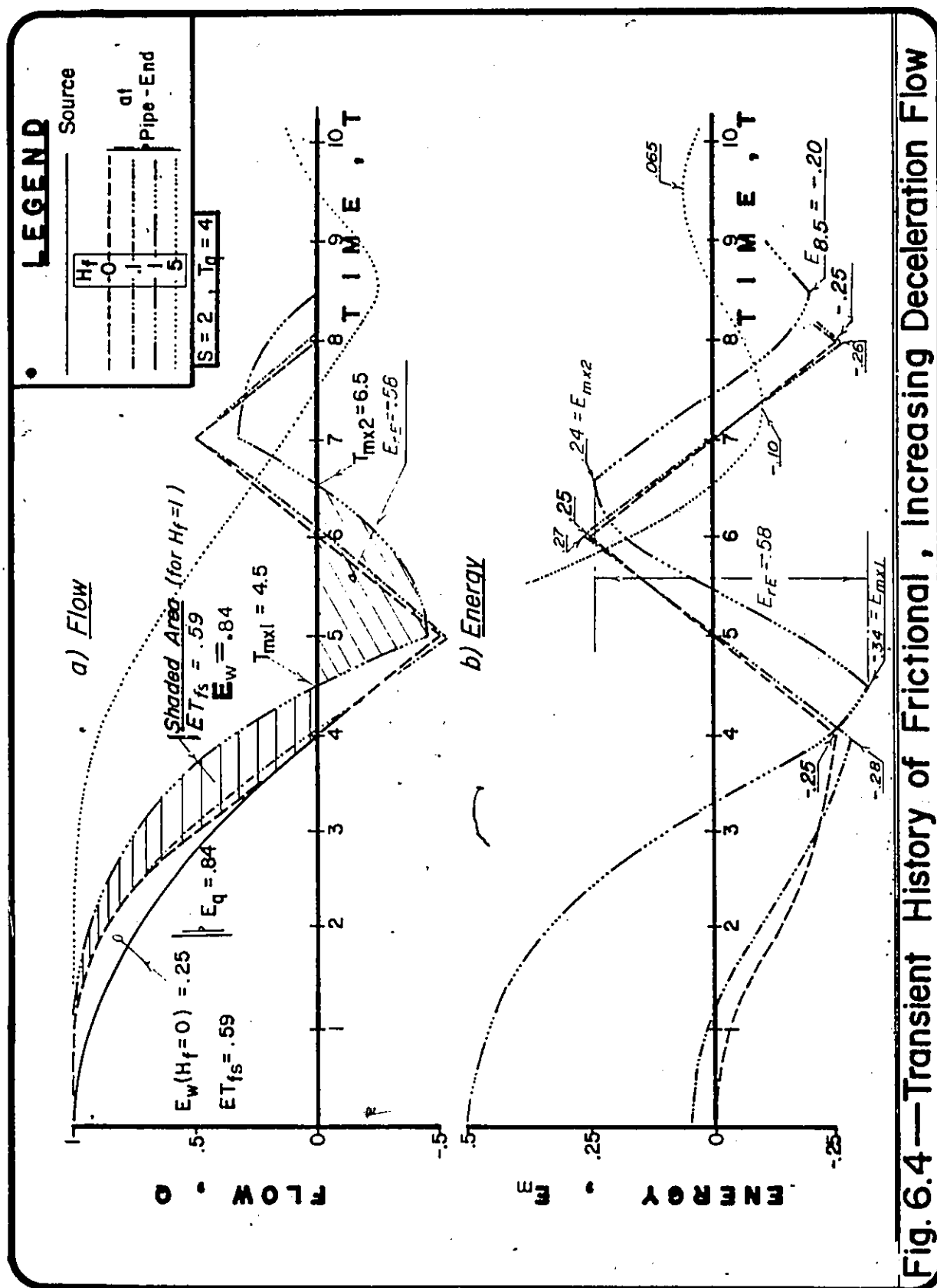


Fig. 6.3—Transient History of Frictional, Decreasing Deceleration Flow



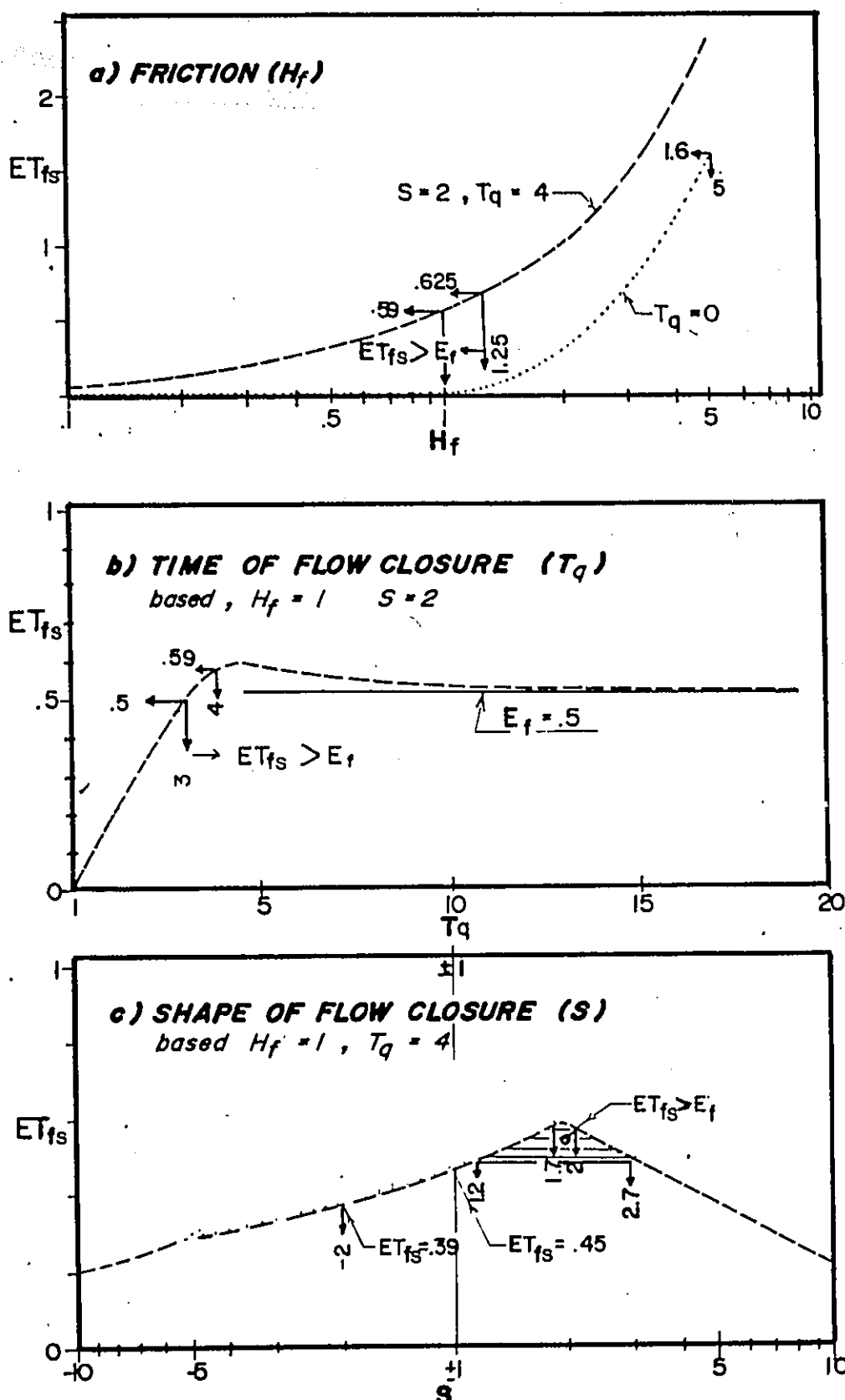
FRICTION SURGE TRANSFER ET_{fs}


Fig. 6.5—Frictional Transfer During Flow Closing Period.

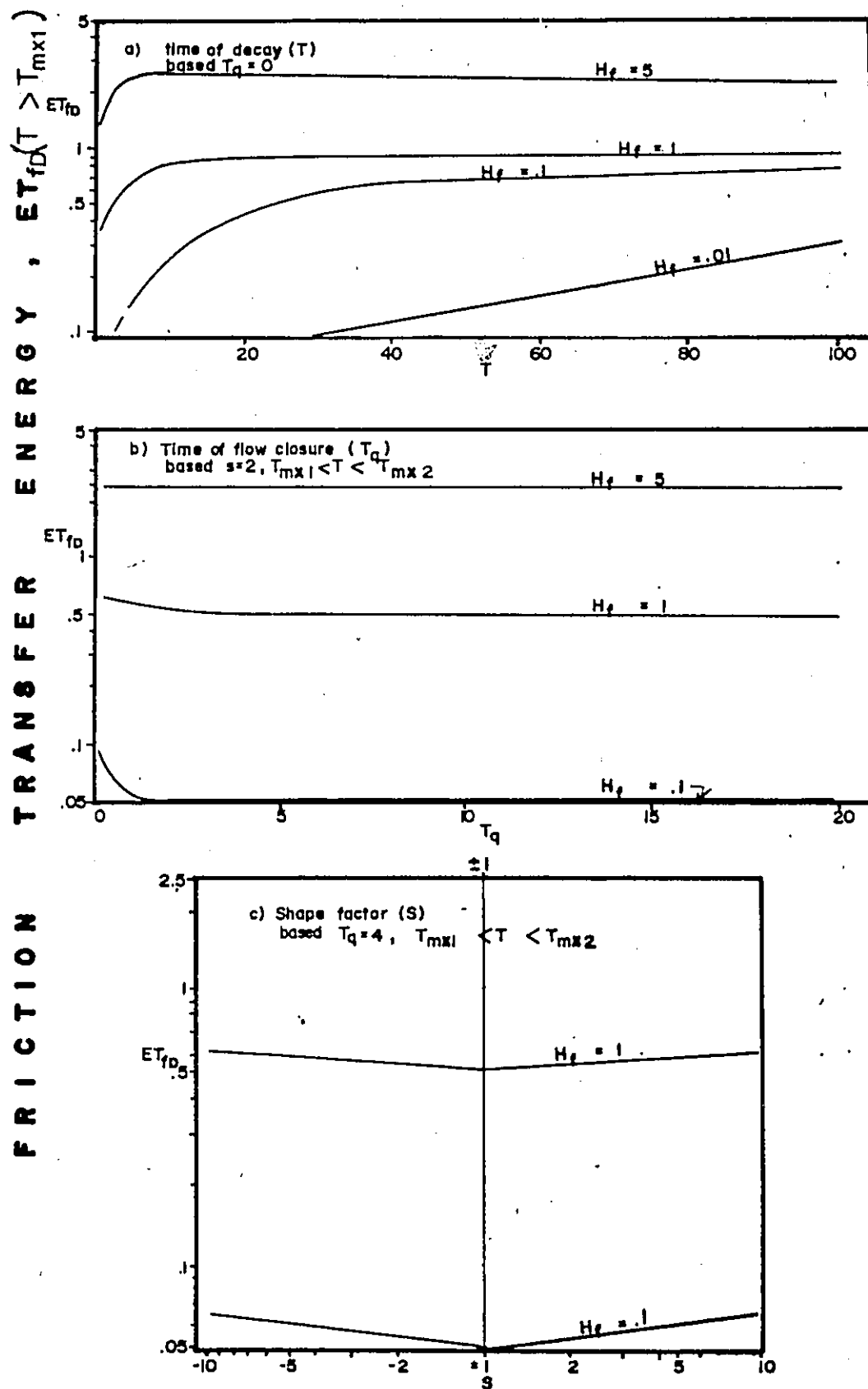


FIG. 6.6— Frictional transfer During the Decaying Period.

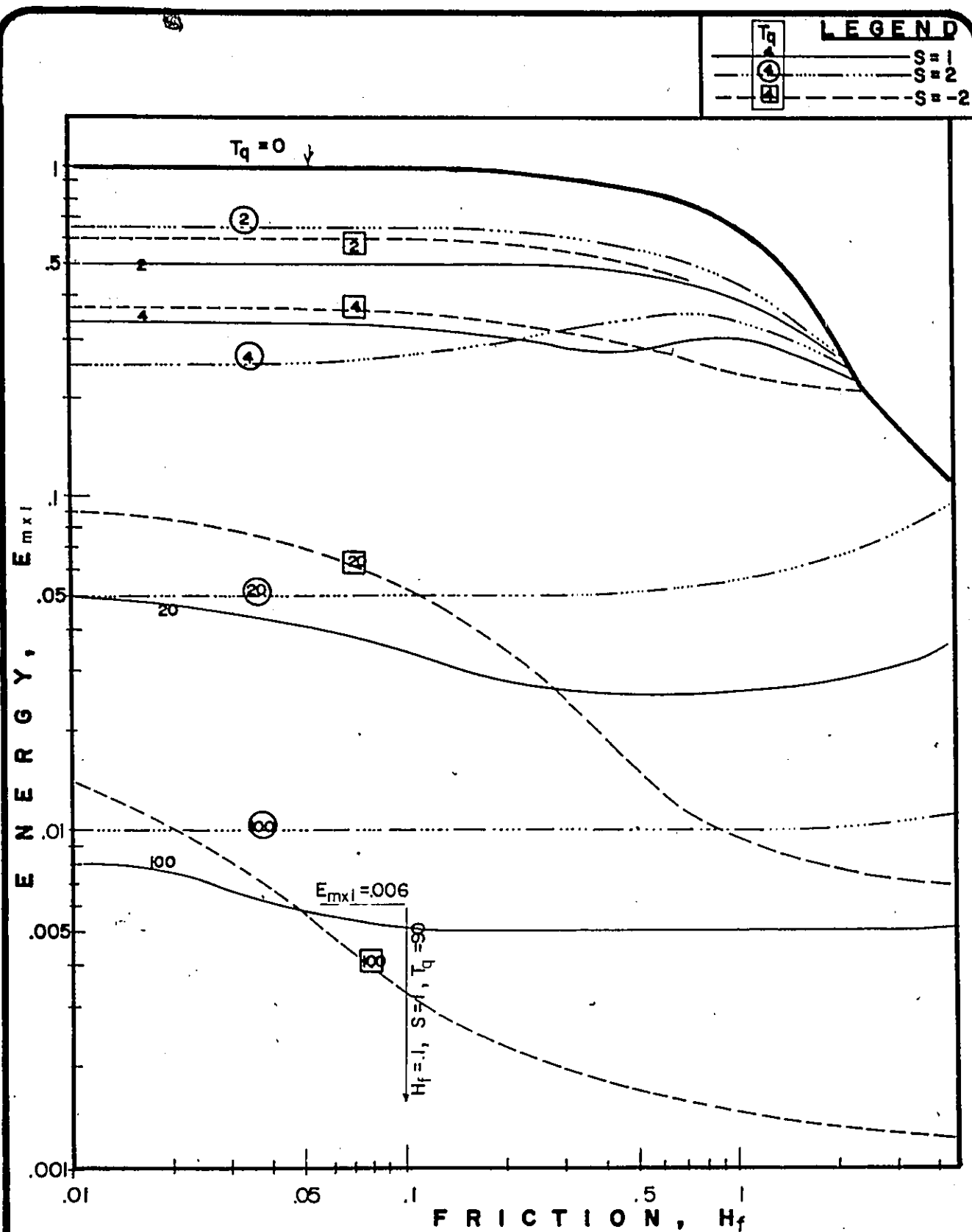


Fig. 6.7 — Maximum Surge Energy induced by Typical Flow Closures.

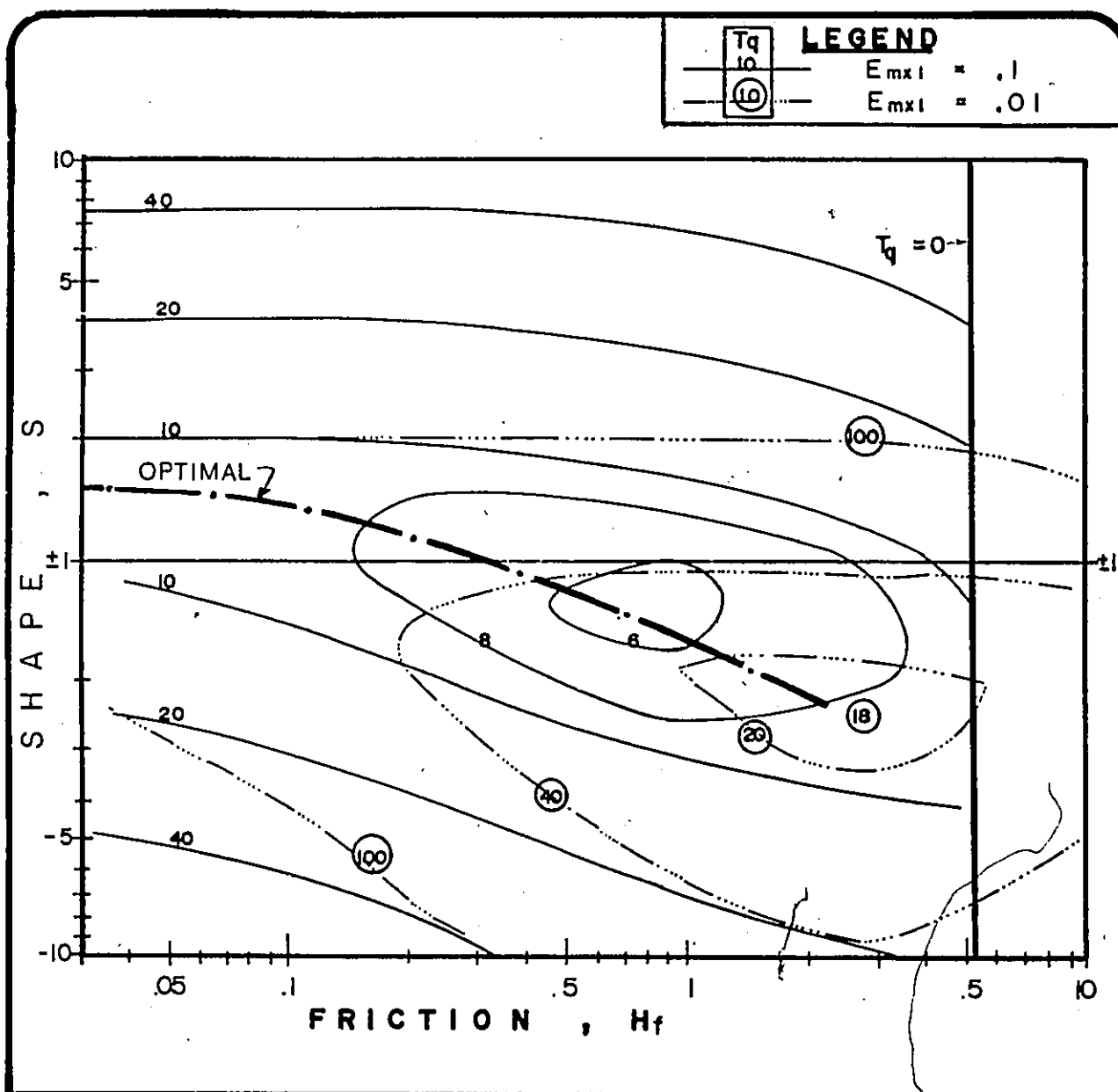


Fig. 6.8 — Conditions for same level of Maximum Surge Energy.

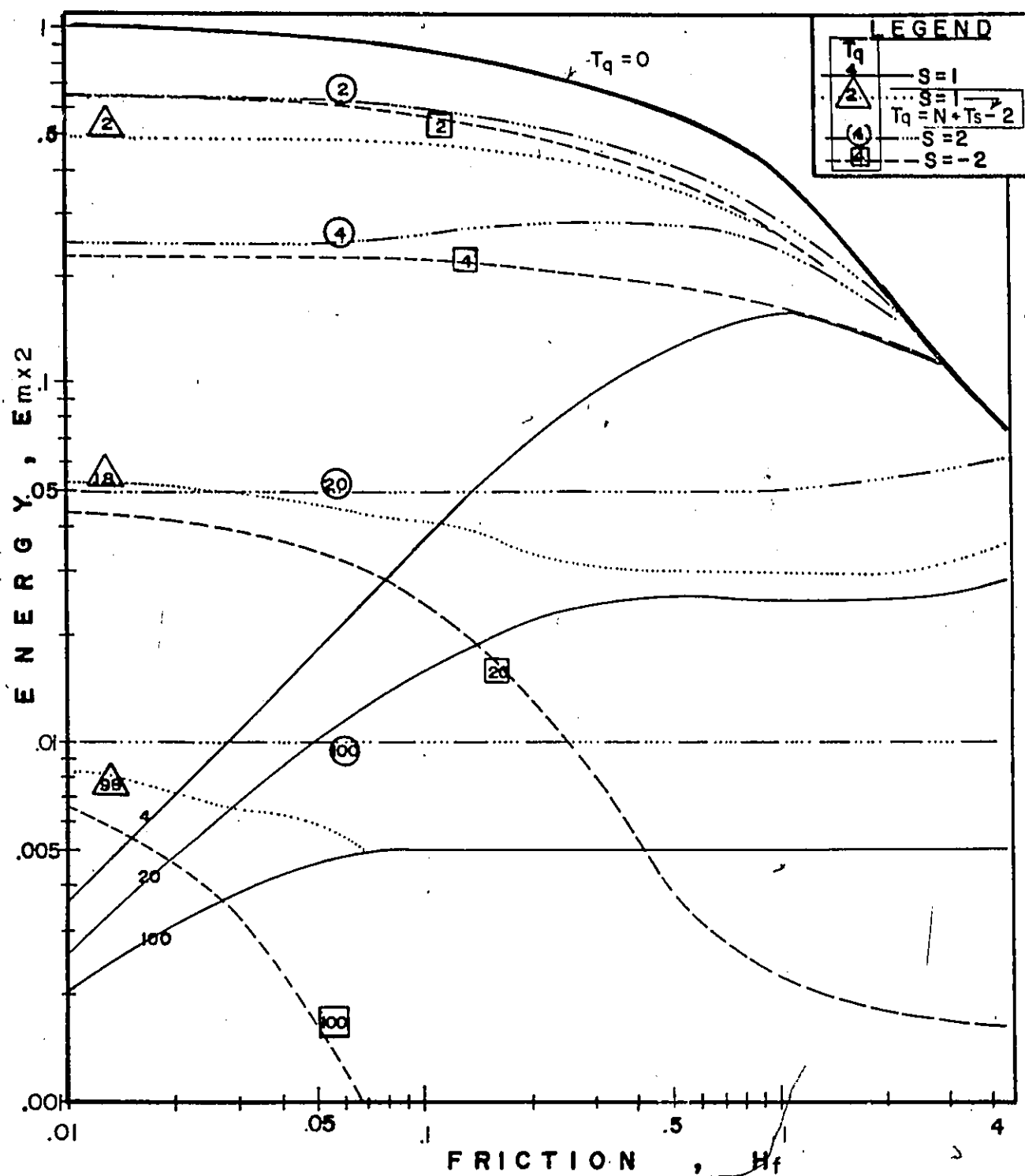


Fig. 6.9 — Maximum Counter-surge Energy induced by Typical Flow Closures.

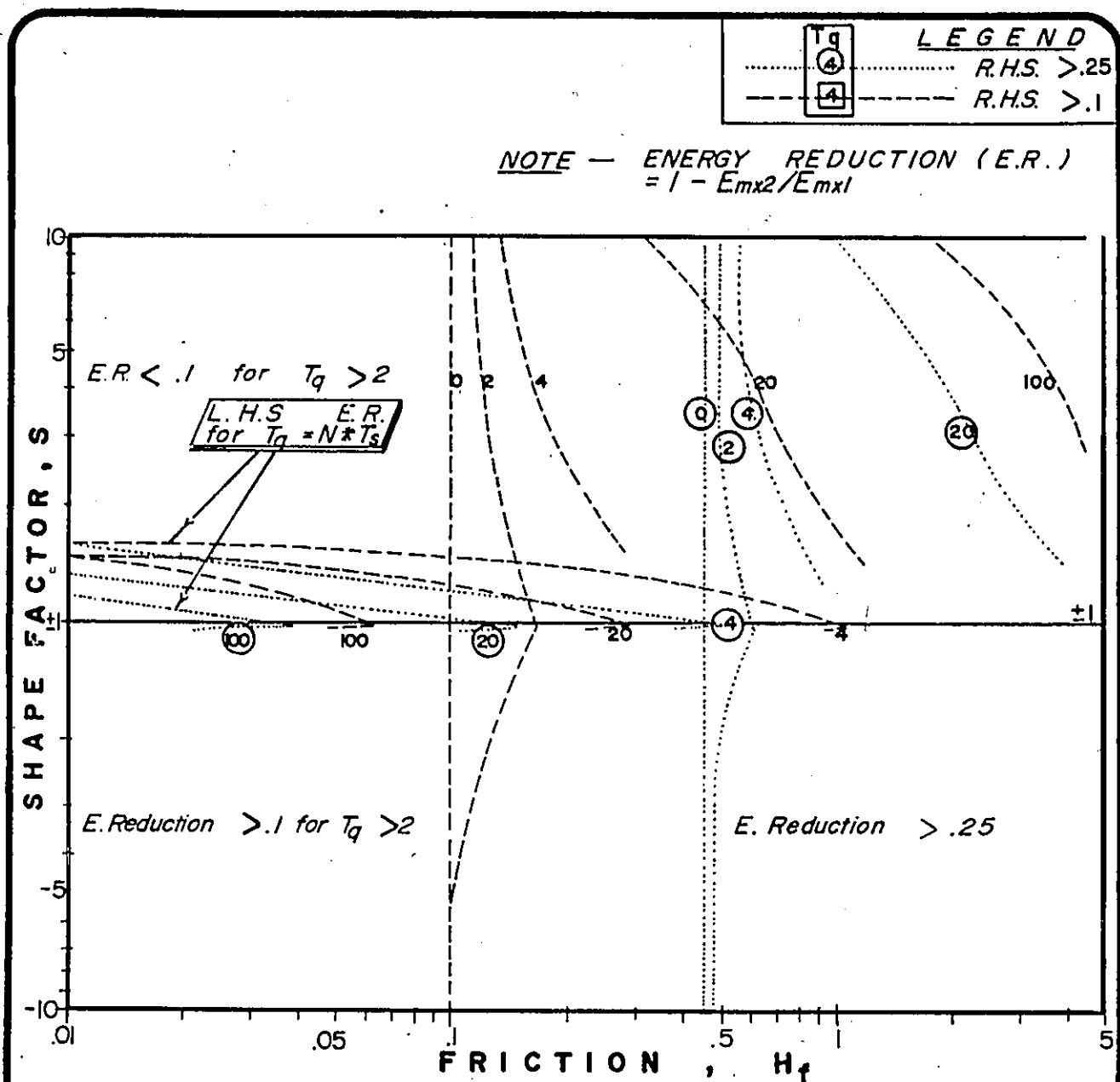


Fig. 6.10—Hydraulic Conditions required for same Energy Reduction during Counter-surge Period.

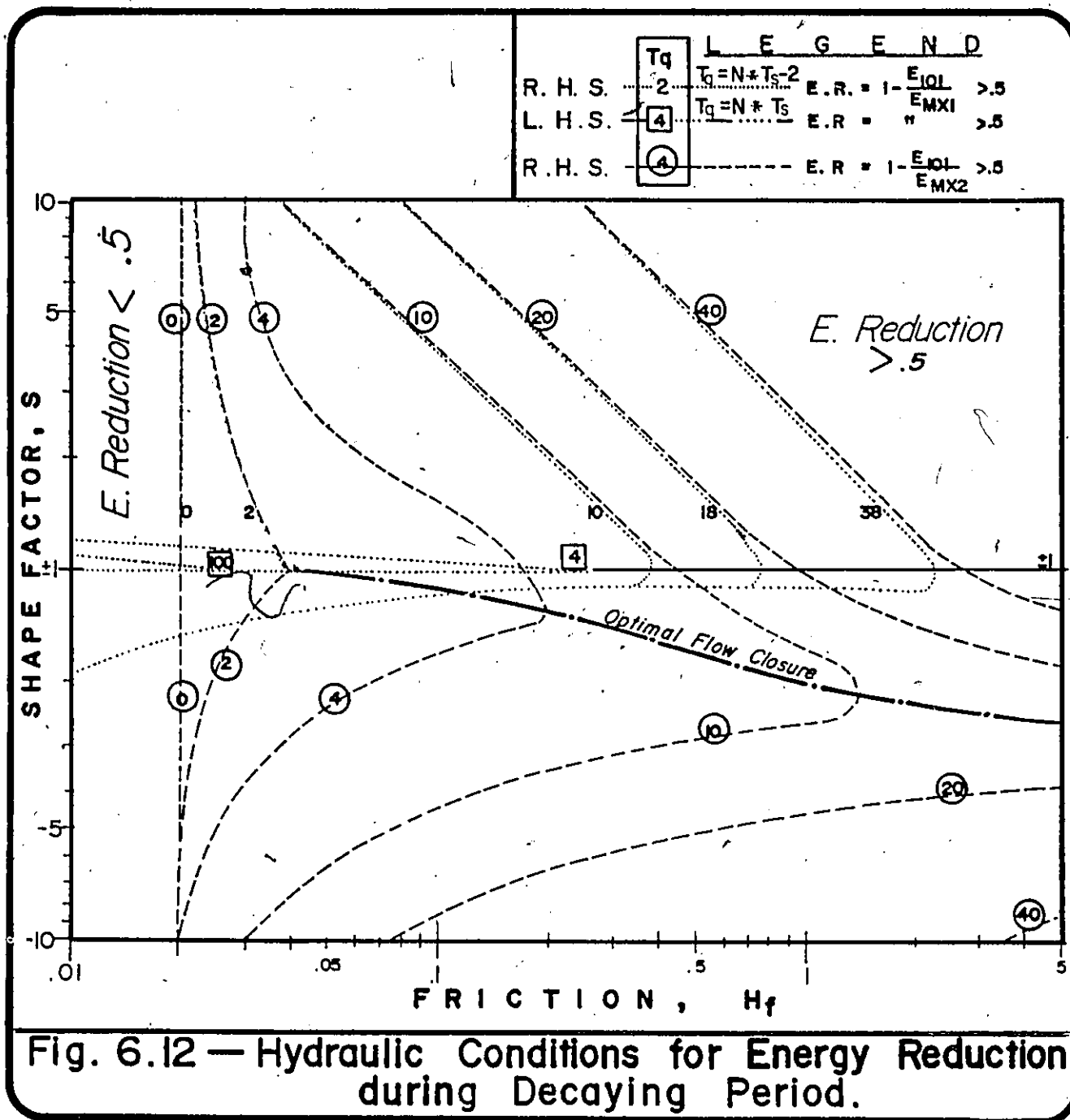


Fig. 6.12 — Hydraulic Conditions for Energy Reduction during Decaying Period.

7.0 ANALYSIS OF TRANSIENTS IN QUASI-STEADY FRICTION SYSTEM

The concepts of mechanical energy transfer (non-friction and friction) have been introduced in the previous chapter, to interpret transient phenomena in a steady friction (factor) system. These concepts are applied in the present chapter in quasi-steady friction (factor) system.

The main objective of this chapter is to evaluate the transient characteristics affected by the existence of a quasi-steady friction factor (f_q). It is felt that a thorough analysis of this most elementary among current unsteady friction models is necessary prior to proceeding toward more advanced unsteady friction models.

7.1 Scope of Analysis

7.1.1 Cases of transient history used for analysis

Three cases of transient history with $H_f = 0.1, 1$ and 5 have been simulated for the case of instantaneous flow closure, representing moderate to high frictional conditions (Fig. 7.1).

Flow histories with $S = 1, -2$ and 2 are plotted in Figs. 7.2, 7.3 and 7.4, representing typical flows with uniform, decreasing and increasing deceleration respectively. Two cases of friction head (i.e. $H_f = 0.1$ and 1) have been analyzed for normal flow closing and for the single case of the time of flow closure ($T_q = 4$).

Table 7.1 summarizes the results of critical transient properties (e.g. E_{mx1} , E_{mx2} and E_{1o1}) for the above cases.

7.1.2 Flow regimes adopted for analysis

Four initial flow regimes have been defined for the transient flow simulations, including:

- i) laminar flow ($R < 2000$);
- ii) transition zone of flow having low turbulence ($R=1 \times 10^4$);
- iii) transition zone of normal turbulent flow ($R=1 \times 10^6$); and
- iv) flow with steady friction factor (f)

The constant friction factor (f) is adopted as a bench mark for the assessment of the quasi-steady friction (f_q) model in this chapter.

The values of Reynolds Number referred to in this chapter are those of the initial steady state conditions, unless otherwise noted.

Since a smooth pipe assumption has been adopted in this research, only one f_q vs. R curve in the Moody diagram is considered for the turbulent flow condition. Therefore, the quasi-steady friction factor characteristics existing during the transient period can be uniquely represented by a function of the appropriate Reynolds Number, as expressed below:

$$f_q(T_1, X_1) = F(R(T_1, X_1)) \quad 7.1$$

where the parameters T_1 and X_1 are referenced to a specific time during the transient period and to a specific point along the system, respectively.

3) Maximum transient energy ---

Comparison of maximum values of critical transient events between constant (f) and quasi-steady (f_q) friction factors has been done for a comprehensive range of frictional ($0.01 < H_f < 10$), flow closing characteristics ($-10 < S < 10$, $0 < T_q < 100$) and flow regime ($2000 < R < 10^6$). Results of this comparison are summarized and presented in two forms:

1) Flow characteristics --- Three shapes of flow closure with $S = -2, 1$ and 2 are selected for illustration of the maximum transient energy, representing decreasing, uniform and increasing deceleration flows.

11) Reduction of maximum transient energy --- Regions have been established to define isoquants of maximum transient energy reduction due to a quasi-steady friction factor for various independent variables.

4) Application ---

The cases of transient history in a quasi-friction system are interpreted in detail using mechanical energy transfer concepts. Though limited by the extent of analysis, these case illustrations are intended to provide insight into the transient phenomena affected by the unsteadiness of a quasi-steady friction system. These two concepts are applied in Secs. 7.2 and 7.3.

The functional relationship established from the dimensional analysis are used as a guide for the research direction, indicating the potential parameters governing transient behavior. This application is demonstrated in Secs. 7.3 and 7.4.

7.2 Energy Relation in a Quasi-steady Friction System

7.2.1 Friction dissipation in transients $ED_z(f_q)$

The friction dissipation (ED_z) proved to be insignificant in a steady friction (factor) system as discussed, in the previous chapter. In a quasi-steady friction system, the friction dissipation (ED_z) is:

$$ED_z(T, f_q) = \int_0^T \int_0^1 f_q(X, T) * Q(X, T) dX dT \quad 7.2$$

where the quasi-steady friction factor, $f_q(X;T) > f$, and is increasing with a decrease in the flow.

Consequently, the quasi-steady friction dissipation, $ED_x(f_q)$ should be greater than the steady friction dissipation $ED_x(f)$.

7.2.1.1 $ED_x(f_q)$ in an instantaneous flow closure

During the surge period of an instantaneous flow closure, the flow rate in line is either the same as for the steady case, or, is zero. Hence, the existence of quasi-steady friction factor does not affect the dissipation since:

$$ED_x(T, f_q) = ED_x(T, f)$$

where $T < T_{max1}$.

7.2.1.2 $ED_x(f_q)$ in a normal flow closure

For the case of uniform flow closure in a laminar flow condition with an upper bound of time of flow closure ($T_q = 100$), the quasi-steady friction dissipation may be increased by a factor of two, since:

$$\begin{aligned} ED_x(f_q) &= \int_0^{100} \int_0^1 f_q(X,T) * Q(X,T) dX dT \\ &= \int_0^{100} \int_0^1 dX dT \\ &= T_q * (a/v * E_x) \\ &= 0.10 * E_x \\ &= 2 * ED_x(f) \end{aligned}$$

Where $f_a = F(1/R) = F(1/Q)$, resulting in $f_a * Q = Q^0$. The steady $ED_x(f)$ is then equal to $0.05 * E_x$, as previously determined.

As an extension of Table 6.2 the friction dissipation in a quasi-steady friction system has been determined for various hydraulic conditions and these are summarized in Table 7.2.

A comparison between the two hydraulic conditions in Table 7.2 indicates that the friction dissipation (ED_x) is still far too small to significantly affect the transient flow phenomenon even with the existence of quasi-steady factor. Hence, it is concluded that the energy balance equation established previously for the steady friction condition is valid for application to this quasi-steady friction (factor) system.

7.2.2 Concept of energy balance in transient flow

In the neglect of friction dissipation, the transient energy for a normal flow closure can be expressed in terms of the steady friction condition. This can be restated as:

$$E_m(T_1) = E_x - E_w(T_1)$$

$$= E_x - [E_{wm}(T_1) - E_{ws}(T_1)]$$

7.3

$$= E_x - \left[\int_0^{T_1} Q_m(T) dT - \int_0^{T_1} Q_s(T) dT \right]$$

where the frictional dissipation (ED_x) is neglected and the processes of mechanical transfer (ET_m and ET_s) are carried out by the flow flux movement.

7.2.3 Discussions of energy relation

The analysis in this section suggests that the friction dissipation ($ED_x(f_a)$) in a quasi-steady friction system is

still too small to be of concern in a transient simulation for $T_q < 100$.

The main theme of this chapter is to evaluate how the relevance of the quasi-steady friction (f_q) affects the mechanical energy transfer phenomenon.

7.3 Quasi (steady) Friction Transfer in Transients

7.3.1 General characteristics

7.3.1.1 Definition of quasi friction transfer

An impact due to the existence of quasi-steady friction factor is the reduction of net energy flux (E_w) during the transient period. A new term quasi friction transfer (ET_{qs}) is defined in this research to evaluate the net energy flux between the steady (f) and quasi-steady (f_q) frictional conditions. This is:

$$ET_{qs}(T) = ET_s(f, T) - ET_s(f_q, T) \quad 7.4$$

Following the procedures of the previous steady friction system the present analysis of quasi friction transfer (ET_{qs}) is divided into two parts below:

i) Quasi surge transfer which is defined as:

$$ET_{qsS} = ET_{ss}(f, T) - ET_{ss}(f_q, T) \quad 7.4a$$

where $0 < T < T_{mx1}$, and

ii) Quasi decay transfer which is defined as:

$$ET_{qsD} = ET_{sd}(f, T) - ET_{sd}(f_q, T) \quad 7.4b$$

where $T_{mx1} < T < T_{mx2}$, $T_{mx2} < T < T_{mx2+2}$,

7.3.1.2 Relevance of quasi friction transfer

The cases illustrated in Figs. 7.1 - 7.4 have been adopted for the present analysis, with the concept of energy balance as an interpretation tool. By screening from these very limited cases, it appears that the existence of quasi-steady friction factor is not relevant unless:

- i) The friction head (H_f) is adequately high; and
- ii) the Reynolds Number (R) is sufficiently low.

For the discussions below, the friction head $H_f > 1$ and laminar flow ($R < 2000$) are selected.

7.3.2 Quasi friction transfer in a surge period ($T < T_{mx1}$)

Referring to Eq. 6.6 for the definition of friction surge transfer (ET_{fs}) during the surge period ($T < T_{mx1}$), the friction surge transfer can be determined for both steady and quasi-steady friction conditions as:

$$ET_{fs}(f) = E_w(f, H_f > 0) - E_w(f, H_f = 0) \quad 7.5a$$

$$ET_{fs}(f_q) = E_w(f_q, H_f > 0) - E_w(f_q, H_f = 0) \quad 7.5b$$

By combining Eqs. 7.4a and 7.5a, the quasi surge transfer becomes:

$$ET_{qfs} = E_w(f, T) - E_w(f_q, T) > 0 \quad 7.6$$

where $0 < T < T_{mx1}$.

The relation given by Eq. 7.6 can then be used directly for the evaluation of quasi-steady friction as it affects maximum surge energy.

7.3.2.1 Instantaneous flow closure ($S' = \infty$)

For the present discussion two conditions can be subdivided, depending on the magnitude of friction heads.

1) Quasi surge transfer for $H_z < 1$ ---

The netflux during the first surge period is independent of the existence of friction, as stated previously for a steady friction system. Therefore, the unsteadiness of the friction factor (f_q) cannot be a relevant parameter during the surge period, as shown in Fig. 7.1a and 7.1b. This can be expressed as:

$$E_w(0 < T < T_{mx1}) = 1, \text{ for any } R$$

and consequently, the quasi surge transfer is:

$$ET_{qzs} = 0$$

2) Quasi surge transfer for $H_z > 1$ ---

The transient history induced by an instantaneous flow closure with $H_z = 5$ has been traced in Fig. 7.1c. Three stages can be classified:

1) $Q_z(f_q) = Q_z(f)$ --- During the first wave travel time, no surge wave reflection occurs in the system. Hence, the outflow rate at the pipe-end is a constant as shown below:

At $0 < T < 1$,

$$Q_z(T, f_q) = Q_z(T, f) = 1$$

Accordingly, the (partial) quasi friction transfer is zero during this period:

$$ET_{qzs}(T) = 0$$

where $0 < T < 1$.

ii) $Q_m(f_q) < Q_m(f)$ --- After the surge wave is first reflected, the rate of outflux will decrease. With the existence of a quasi-steady friction factor, the flow outflux decreases faster at the early stage of wave reflection than for the modeling a constant friction factor. As shown in Fig. 7.1c,

at $1 < T < T_{qz}=1.6$,

$$Q_m(T, f_q) < Q_m(T, f) < 1$$

where the term T_{qz} is defined here as the time of quasi flow intersection, when the pipe-end flow (Q_m) with constant friction factor (f) varies from greater to smaller than for Q_m with a quasi-steady friction factor (f_q).

Hence, the second segment of quasi surge transfer within this period is greater than zero because:

$$ET_{qzs}(T) = E_w(T, f) - E_w(T, f_q) = 0.38 > 0$$

where $1 < T < T_{qz}=1.6$.

iii) $Q_m(f_q) > Q_m(f)$ --- Toward the end of surge period, the above situation reverses because it takes a longer time for the flow with a quasi-steady friction factor to drop to zero. As shown in Fig. 7.1c,

at $T_{qz} < T < T_{mx1}=T_{x1}=7$,

$$1 > Q_m(f_q) > Q_m(f)$$

Consequently, the last part of quasi surge transfer is below zero because:

$$ET_{qzs}(T) = E_w(T, f) - E_w(T, f_q) = -0.29 < 0$$

where $T_{qz} < T < T_{mx1}=7$.

For the case of $R < 2000$, $H_s = 5$ and $T_q = 0$, the quasi-friction transfer over the whole surge period is:

$$\begin{aligned} ET_{qzs} &= ET_{qzs}(1) + ET_{qzs}(1 < T < 4) + ET_{qzs}(4 < T < 7) \\ &= 0 + 0.38 - 0.29 \\ &= 0.09 \end{aligned}$$

The relation expressed in the quasi surge transfer should also imply the same reduction in maximum surge energy due to the unsteadiness of the quasi-friction effect. For the present case ($R < 2000$),

$$-E_{mx1}(f_q) = 0.01 < -E_{mx1}(f) = 0.10$$

where the surge energy has been reduced by 0.09.

7.3.2.2 Quasi surge transfer⁴ in normal flow closure

Three typical flow closures are discussed here for $R < 2000$, $H_s = 1$, and $T_q = 4$.

1) Uniform deceleration flow ($S = 1$) ---

As shown in Fig. 7.2b, the time of quasi-steady flow interception is:

$$T_{qs} = 3.6 < T_q < T_{mx1} = 4.4$$

The quasi surge transfer during the surge period consists of two components as:

$$\begin{aligned} ET_{qzs} &= ET_{qzs}(1) + ET_{qzs}(1 < T < 3.6) + ET_{qzs}(3.6 < T < 4.4) \\ &= 0 + 0.155 - 0.045 \\ &= 0.11 \end{aligned}$$

Accordingly, maximum surge energy has been reduced (as shown in Table 6.1):

$$-E_{mx1}(f_q) = 0.09 < -E_{mx1}(f) = 0.20$$

2) Decreasing deceleration flow ---

As shown in Fig. 7.3b;

$$T_{qs} = 2.9 > T_{mx1}(f_q) = 2.7$$

The quasi friction transfer (ET_{qs}) is contributed by only one energy component because:

$$\begin{aligned} ET_{qs} &= E_w(f, T_{mx1}) - E_w(f_q, T_{mx1}) \\ &= 0.14 \end{aligned}$$

where $T_{mx1}(f) = 2.8$ and $T_{mx1}(f_q) = 2.7$.

As shown in Table 6.1; maximum surge energy now becomes: ---

$$-E_{mx1}(f_q) = 0.11 < -E_{mx1}(f) = 0.25$$

3) Increasing deceleration flow ---

As shown in Fig. 7.4b, the time of quasi-steady flow interception occurs just before the time of maximum surge energy because:

$$T_{qs} = 4.4 < T_{mx1}(f_q) = 4.5$$

Consequently, the last energy component is very small and can be expressed as:

$$\begin{aligned} ET_{qs} &= ET_{qs}(1) + ET_{qs}(1 < T < 4.4) + ET_{qs}(4.4 < T < 4.5) \\ &= 0 + 0.15 - 0.005 \\ &= 0.14 \end{aligned}$$

As shown in Table 6.1, the maximum surge energy in the quasi-steady friction system is:

$$-E_{mx1}(f_q) = 0.20 < -E_{mx1}(f) = 0.34$$

7.3.3 Quasi friction transfer in decaying period ($T > T_{mx1}$)

Referring to Eq. 6.14 for the definition of friction decay transfer ($ET_{\pm D}$) during the decaying period, the steady and quasi-steady friction decay transfer can be determined as:

$$ET_{\pm D}(f) = E_Q(f) - E_{xx}(f) / 2 \quad 7.7a$$

$$ET_{\pm D}(f_Q) = E_Q(f_Q) - E_{xx}(f_Q) / 2 \quad 7.7b$$

where the quasi decay transfer $ET_{\pm D}$ should be below zero because the damping capability of a quasi-steady friction system, i.e. $ET_{\pm D}(f_Q)$, is expected to be higher than for the case of steady $ET_{\pm D}(f)$.

It should be noted that Eqs. 7.4b and 7.7 can be applied to counter-surge periods ($T_{mx1} < T < T_{mx2}$) as well as to other subsequent peak decaying periods (e.g. $T_{mx2} < T < T_{mx2+2}$,).

7.3.3.1 Instantaneous flow closure

An example of $R < 2000$, $H_z = 1$ and $T_Q = 0$ is adopted here for the analyses of instantaneous flow closure. The (steady) friction decay transfer was previously determined (6.2.2.2) as:

$$ET_{\pm D}(f) = 0.605;$$

and the flow closure energy remains constant for:

$$E_Q(f) = E_Q(f_Q) = E_w(T_{mx1}) = 1$$

As shown in the shaded area of Fig. 7.1b, the energy reversal is:

$$\begin{aligned} E_{xx}(f_Q) &= E_{mx2}(f_Q) - E_{mx1}(f_Q) \\ &= 0.14 - (-0.50) \\ &= 0.64 \end{aligned}$$

Hence, the friction decay transfer for the quasi-steady condition is:

$$\begin{aligned} ET_{SD}(f_q) &= E_q(f_q) - E_{xx}(f_q) / 2 \\ &= 1 - 0.64 / 2 \\ &= 0.68 \end{aligned}$$

Hence,

$$\begin{aligned} ET_{QSD}(T) &= ET_{SD}(f, T) - ET_{SD}(f_q, T) \\ &= 0.605 - 0.68 \\ &= -0.075 \end{aligned}$$

where $T_{mx1} < T < T_{mx2}$.

Consequently, the maximum counter-surge energy has been significantly reduced due to the effect of the quasi-steady friction factor (as indicated in Table 6.1). This can be expressed as:

$$E_{mx2}(f_q) = 0.14 < E_{mx2}(f) = 0.29$$

7.3.3.2 Quasi decay transfer in normal flow closure

The values of friction transfer $ET_x(f)$ used below were previously determined in the analysis of a steady friction system.

1) Uniformly decelerating flow ($S = 1$) ---

As shown in Fig. 7.2b, the flow closure energy is:

$$\begin{aligned} E_q(f_q) &= E_q(f) - ET_{SS}(T_{mx1}) \\ &= 0.70 - 0.11 \\ &= 0.59; \text{ and} \end{aligned}$$

The energy reversal is the shaded area below the $Q = 0$ axis in the Fig. 7.2b as:

$$\begin{aligned}
 E_{xx}(f_q) &= E_{mx2}(f_q) - E_{mx1}(f_q) \\
 &= 0.01 - (-0.09) \\
 &= 0.10
 \end{aligned}$$

Hence, the friction transfer for the quasi-steady friction system becomes:

$$\begin{aligned}
 ET_{sd}(f_q) &= E_q(f_q) - E_{xx} / 2 \\
 &= 0.59 - 0.10 / 2 \\
 &= 0.54
 \end{aligned}$$

Therefore, the quasi friction transfer becomes:

$$\begin{aligned}
 ET_{qsd}(T) &= ET_{sd}(f, T) - ET_{sd}(f_q, T) \\
 &= 0.52 - 0.54 \\
 &= -0.02
 \end{aligned}$$

where $T_{mx1} < T < T_{mx2}$.

As shown in Table 6.1, maximum counter-surge in this quasi-steady friction has almost been damped out:

$$E_{mx2}(f_q) = 0.01 \ll E_{mx2}(f) = 0.16$$

2) Flow with decreasing deceleration ($S < -1$) ---

The effect of quasi-steady friction factor on counter-surge transients is shown in Fig. 7.3b. Following the same procedures above, the following analysis below has been undertaken:

$$\begin{aligned}
 E_q(f_q) &= E_q(f) - ET_{ss}(T_{mx1}) \\
 &= 0.74 - 0.14 \\
 &= 0.60
 \end{aligned}$$

$$\begin{aligned}
 E_{xx}(f_q) &= E_{mx2}(f_q) - E_{mx1}(f_q) \\
 &= 0.01 - (-0.11) \\
 &= 0.12
 \end{aligned}$$

$$\begin{aligned}
 ET_{\pm D}(f_q) &= E_q(f_q) - E_{xx} / 2 \\
 &= 0.60 - 0.12 / 2 \\
 &= 0.54
 \end{aligned}$$

$$\begin{aligned}
 ET_{q\pm D}(T) &= ET_{\pm D}(f, T) - ET_{\pm D}(f_q, T) \\
 &= 0.52 - 0.54 \\
 &= -0.02
 \end{aligned}$$

where $T_{mx1} < T < T_{mx2}$.

$$E_{mx2}(f_q) = 0.01 \ll E_{mx2}(f) = 0.18$$

3) Flow with increasing deceleration ($S > 1$) ---

Fig. 7.4b shows the decaying transients for various flow regime conditions. Similarly, the effect of quasi-steady friction factor can be determined as:

$$\begin{aligned}
 E_q(f_q) &= E_q(f) - ET_{ss}(T_{mx1}) \\
 &= 0.84 - 0.14 \\
 &= 0.70
 \end{aligned}$$

$$\begin{aligned}
 E_{xx}(f_q) &= E_{mx2}(f_q) - E_{mx1}(f_q) \\
 &= 0.01 - (-0.20) \\
 &= 0.21
 \end{aligned}$$

$$\begin{aligned}
 ET_{\pm D}(f_q) &= E_q(f_q) - E_{xx} / 2 \\
 &= 0.70 - 0.21 / 2 \\
 &= 0.60
 \end{aligned}$$

$$\begin{aligned}
 ET_{q\pm D}(T) &= ET_{\pm D}(f, T) - ET_{\pm D}(f_q, T) \\
 &= 0.55 - 0.60 \\
 &= -0.05
 \end{aligned}$$

where $T_{mx1} < T < T_{mx2}$.

$$E_{mx2}(f_q) = 0.01 \ll E_{mx2}(f) = 0.24$$

7.3.4 Summary of quasi friction transfer

The effects of quasi-steady friction factor on transients have been carried out and is summarized in Table 7.3. These are discussed below:

i) Surge period ($T < T_{mx1}$) --- For the present cases of normal flow closure, i.e. $R < 2000$, $H_z = 1$ and $T_q = 4$, significant addition of energy decay has been induced during the surge period. This is shown from the quasi surge transfer ET_{qss} in Table 7.2.

ii) Decaying period --- For the counter-surge or the subsequent decaying periods, the effect of quasi-steady friction factor is not significant, as compared between friction decay transfer (ET_{sd}) and quasi decay transfer (ET_{qsd}) of Table 7.2 during these periods.

However, significant reductions in maximum counter-surge energy, i.e. $E_{mx2}(f_q)$ vs. $E_{mx2}(f)$ and peak decaying energy, i.e. $E_T(f_q)$ vs. $E_T(f)$ (see Table 7.1 vs. Table 7.2) have been observed. The cause of reduction is primarily due to the decrease in flow closure energy, $E_q(f_q)$ vs. $E_q(f)$ (see Table 7.2 vs. Table 6.2) occurring during the flow closing period ($0 < T < T_q$).

7.3.5 Discussion of quasi friction transfer

The adoption of the mechanical energy transfer concept can also be used effectively to interpret transient behavior with the inclusion of the quasi-steady friction factor.

In a normal flow closure condition ($T_q > 0$), the increase of damping capability due to the quasi-steady friction occurs mainly during the flow closing period ($T > T_q$) when the flow is large. The effect of the quasi-steady friction factor is small after the end-of flow closure (at $T < T_q$).

In this section the analysis of quasi friction transfer (ET_{qs}) phenomenon has been undertaken for a limited number of cases. The relevance of each governing parameter will be analyzed in the next section.

7.4 Parameters Governing Quasi Friction Transfer

Analysis from the previous section clarified one important concept related to the unsteadiness of quasi-friction affecting transient characteristics. The term quasi friction transfer (ET_{qs}) was introduced to evaluate the reduction of transient energy due to the existence of quasi-steady friction.

As established by dimensional reasoning (Eq. 4.6) the quasi friction transfer is governed by the following five parameters:

$$ET_{qs}(T) = F(H_s, S, T_q, R) \quad 7.8$$

As noted in the previous discussion the analysis of this quasi friction transfer can be divided into two groups, as follows:

- i) Quasi surge transfer (ET_{qss}) --- where the main theme of analysis is the reduction of maximum surge energy ($0 < T < T_{mx1}$); and
- ii) Quasi decay transfer (ET_{qsd}) --- where the decay of elastic energy after the surge period is the focus ($T > T_{mx1}$).

7.4.1 Quasi friction transfer to surge energy

The functional relationship between ET_{qss} and its governing parameters (Eq. 7.8) can be simplified as follows for a normal flow closure:

$$ET_{qss} = F(H_s, S, T_q, R) \quad 7.9$$

This functional relationship (Eq. 7.9) has been summarized in Fig. 7.5a - 7.5c. Each part of the figure reflects one variable (i.e. H_f , T_q and S) in the equation while the effect of the quasi-steady friction factor under the laminar flow condition (in terms of $R < 2000$) is presented throughout all three parts of Fig. 7.5. Normally the effect of the quasi-steady friction factor is very small for $R > 1 \times 10^4$, and can hardly be shown in a plot with linearly scaled energy units, as shown in part (a) of Figs. 7.1 - 7.4.

During the analysis of each individual parameter as discussed below, constant values of other parameters are selected to represent the most typical or critical hydraulic conditions.

7.4.1.1 Friction head (H_f) as a relevant parameter

The relations illustrated in Fig. 7.5a can be expressed as:

$$ET_{qzs} = F(H_f) \quad 7.9a$$

where $T_q = 0$ and 4, $S = 2$ and $R = 2000$.

i) Instantaneous flow closure ($T_q = 0$) --- The quasi surge transfer (ET_{qzs}) stays at zero at $H_f > 0$. Then it increases with friction until it reaches its maximum value at $H_f = 2.5$.

ii) Normal flow closure ($T_q = 4$) --- The quasi friction transfer increases continuously with friction until $H_f = 1.5$, when the maximum reduction of surge energy occurs.

iii) Relevance of friction (H_f) --- The effect of quasi-steady friction factor is however very minimal for $H_f < 0.1$, even for the laminar flow condition. This indicates the importance of high friction head (H_f) in the evaluation of quasi-steady friction factor effects.

7.4.1.2 Time of flow closure (T_q) as a relevant parameter

The relationship given by Fig. 7.5b can be expressed as:

$$ET_{qzs} = F(T_q) \quad 7.9b$$

where $H_s = 1$, $S = 2$ and $R = 2000$.

i) ET_{qzs} increases with T_q --- For the present condition, the quasi surge transfer increases with time of flow closure until $T_q = 4$.

ii) ET_{qzs} decreases with the increase of T_q --- For $T_q > 4$, the past accumulated wave reflections reduces the importance of the quasi-steady friction factor affecting the surge energy.

7.4.1.3 Shape of flow closure (S) as a relevant parameter

The results of Fig. 7.5c can be expressed as below:

$$ET_{qzs} = F(S) \quad 7.9c$$

where $H_s = 1$, $T_q = 4$ and $R = 2000$.

i) ET_{qzs} vs. S --- The quasi surge transfer generally decreases with the increasing shape factor $|S|$, with minimum occurring at the instantaneous closure [i.e. $ET_{qzs}(S=\infty) = 0$].

ii) Symmetrical property --- the following relation of symmetry is a valid approximation:

$$ET_{qzs}(+S) = ET_{qzs}(-S) \quad 7.10$$

where the maximum magnitude of quasi surge transfer (ET_{qzs}) occurs at $|S| = 2$.

For flows with $S = 1$, the effect of periodic fluctuation condition ($ET_{\pm}$) is the most prominent compared with other shapes of flow closure. Hence, this results in a decrease of $ET_{\pm}(S)$ from $|S| = 2$ to 1.

7.4.2 Quasi friction transfer to decaying energy

Each part of Fig. 7.6 reflects the effect of one specific variable. In total, three of the five parameters in Eq. 7.8 have been represented (i.e. T , T_q and S). The effects of various friction heads (i.e. $H_f = 0.1$ and 1) can be observed throughout all three parts of the Fig. 7.6. Since the effect of the quasi-steady friction factor on the decay of transient energy is normally small for turbulent flow conditions, only the event of laminar flow is presented in the Figure 7.6.

7.4.2.1 Time of transient (T) as a relevant parameter

The relation given in Fig. 7.6a can be expressed as:

$$ET_{q\pm D}(T) = F(T) \quad 7.11a$$

where $T > T_{mx1}$, $0.01 < H_f < 1$, $S = \infty$, $T_q = 0$ and $R < 2000$.

In this figure the quasi decay transfer ($ET_{q\pm D}$) has to be interpreted from the difference of friction decay transfer between the steady $ET_{\pm D}(f)$ and quasi-steady $ET_{\pm D}(f_q)$ conditions. During the early decaying period, the quasi decay transfer ($ET_{q\pm D}$) increases with friction (H_f). Then $ET_{q\pm D}$ continues to increase with time, particularly for low friction head conditions. However, as the quasi decay transfer ($ET_{q\pm D}$) approaches the flow closure energy (E_q), towards the end of decaying period, this quasi decay transfer ($ET_{q\pm D}$) begins to diminish.

7.4.2.2 Time of flow closure (T_q) as a relevant parameter

The given relation in Fig. 7.6b is shown below:

$$ET_{Q\pm D}(T) = F(T_Q) \quad 7.11b$$

where $T_{mx1} < T < T_{mx2}$, $0.1 < H_f < 1$, $S = 2$ and $R < 2000$.

The magnitude of the quasi decay transfer ($ET_{Q\pm D}$) rapidly decreases with increasing time of flow closure (T_Q). For example, the effect of quasi-steady friction on transient energy is hardly noticeable for $T_Q > 4$.

7.4.2.3 Shape of flow closure (S) as a relevant parameter

The results of analysis in Fig. 7.6c can be expressed as:

$$ET_{Q\pm D}(T) = F(S) \quad 7.11c$$

where $T_{mx1} < T < T_{mx2}$, $0.1 < H_f < 1$, $T_Q = 4$ and $R < 2000$.

i) $ET_{Q\pm D}$ vs. S --- The magnitude of the quasi decay transfer ($ET_{Q\pm D}$) increases with the shape factor $|S|$, with the instantaneous flow closure ($S = \infty$) giving the maximum $ET_{Q\pm D}$.

ii) Symmetrical property --- Approximately the same value of $ET_{Q\pm D}$ can be obtained for the same $|S|$, independent of the increasing or decreasing deceleration flow. This conforms to the symmetrical properties expressed in Eq. 7.10.

7.4.3 Discussion of governing parameters

The concept of quasi friction transfer ($ET_{Q\pm}$) has been adopted in this section to evaluate the relevance of governing parameters. The analysis has been conducted separately for two conditions, namely the quasi surge ($ET_{Q\pm S}$) and decay ($ET_{Q\pm D}$) transfer for the surge ($T < T_{mx1}$) and the decaying ($T > T_{mx1}$) period respectively.

It appears that high friction head (H_f) and low Reynolds Number (R) are necessary conditions for the parameters to be relevant. Furthermore, a long time (T) of transient and a

rapid flow closure ($|S| = \infty$ or $T_q = 0$) are also required for achieving an efficient damping process.

After the present test of the parameters' relevance, the analysis proceeds to evaluate the effect of quasi-steady friction on transients in a wide-ranged and inter-related manner. This will be carried out in the next section.

7.5 Maximum Transient Energy

7.5.1 Evaluation criteria for maximum transients

In practice, the effect of quasi-steady friction (f_q) on transients is evaluated on a relative basis, in which quasi energy reduction (ER_{qs}) is defined as below:

$$ER_{qs} = | (E_{mx}(f) - E_{mx}(f_q)) / E_{mx}(f) | \quad 7.12$$

where the numerator of Eq. 7.12, $E_{mx}(f) - E_{mx}(f_q)$, is on the absolute basis of energy reduction.

1) Maximum surge energy (E_{mx1}) ---

Here the quasi surge reduction is determined from:

$$\begin{aligned} ER_{qs1} &= | (E_{mx1}(f) - E_{mx1}(f_q)) / E_{mx1}(f) | \quad 7.12a \\ &= ET_{qs1} / E_{mx1}(f) \end{aligned}$$

where $E_w(f) - E_w(f_q) = E_{mx1}(f) - E_{mx1}(f_q) = ET_{qs1}$.

2) Peak decaying energy (E_T) ---

Here the quasi decay reduction is estimated from:

$$ER_{qsD}(T) = | (E_T(f) - E_T(f_q)) / E_T(f) | \quad 7.12b$$

where $T > T_{mx1}$. Here $E_T(f) - E_T(f_q) > -ET_{qsD}$. Part of the additional energy decay, $E_T(f) - E_T(f_q)$, is contributed by the change of flow closure energy, $E_q(f) - E_q(f_q)$ which occurs during the flow closing period ($T < T_q$).

7.5.2 Maximum surge energy

Maximum surge energy in a quasi-steady friction system is determined by four relevant parameters, as established from dimensional reasoning (Eq. 4.6b):

$$E_{mx1} = F(H_z, S, T_q, R) \quad 7.13$$

Common effects induced by the existence of a quasi-steady friction factor on the maximum surge energy are discussed in this subsection.

Fig. 7.7a, 7.7b and 7.7c show the relationships of:

$$E_{mx1} = F(H_z, T_q, R) \quad 7.13a$$

for typical flows with $S = -2, 1$ and 2 respectively.

Fig. 7.8 plots the isoquants of a constant ER_{qss} for various hydraulic conditions. This is expressed as:

$$ER_{qss} = F(H_z, S, T_q, R) \quad 7.13b$$

where $ER_{qss} = 0.1$.

7.5.2.1 General characteristics of quasi friction impact

1) Steady flow regime (R) on energy reduction ---

As shown in Figs. 7.7 and 7.8, the effect of the quasi-steady friction factor on maximum surge energy (ET_{qss}) increases with the decrease of the Reynolds Number (R). For the normal turbulent flow condition ($R = 10^6$), a large frictional energy is required for this parameter to become relevant.

As shown in Fig. 7.8 there is little effect induced from the existence of a quasi-steady friction factor for the case of increasing deceleration flow in turbulent condition with

$R > 10^4$. On the other hand, for decreasing deceleration flow some effect of quasi-steady friction may be observed for relatively high friction (H_z) or for long flow closing times (T_q).

2) Optimal energy reduction due to quasi-steady friction ---

The maximum magnitude on reducing maximum surge energy can be achieved under the following two conditions:

i) Time of flow closure (T_q) --- As shown in the upper part of Fig. 7.8, the effect of quasi-steady friction is the strongest at $T_q = 4$, for flows with $S > 1$.

ii) Shape of flow closure (S) --- As shown in the lower part of Fig. 7.8, the most efficient energy reduction due to quasi-steady friction is for the shape (S) of flow closure varying approximately between -1.25 and -2.5 with respect to laminar ($R < 2000$) and normal turbulent flow ($R = 10^4$) conditions respectively.

3) Non-symmetrical property of energy reduction ---

Comparing the upper and lower parts of Fig. 7.8, it is evident that the relative effect of the quasi-steady friction factor on maximum surge energy is stronger for a decreasing deceleration flow. This means that:

$$ER_{qss}(-S) > ER_{qss}(+S) \quad 7.14$$

The inequality of Eq. 7.14 is mainly due to the difference of maximum surge energy for different types of decelerating flows. As shown in Fig. A.3 (to be discussed in the section on optimal shape factor), $-E_{mx1}(-S) < -E_{mx1}(+S)$ is valid for adequately large time (T_q) in flow closure and friction head (H_z). The actual difference of friction transfer (ET_{qss}) is approximately the same for the two deceleration flow conditions, as indicated in Eq. 7.10.

7.5.2.2 Properties related to flow closing performance

1) Flow with decreasing deceleration ($S < -1$)

Analysis related to flow with decreasing deceleration can be shown in Fig. 7.7a for $S = -2$, or from the lower part of Fig. 7.8 for $S < -1$.

i) Shape of flow closure ($-S$) --- Generally, the higher the value of the shape factor $|-S|$, the less sensitive the quasi-steady friction factor is in reducing maximum surge energy. For an instantaneous flow closure, virtually no affect is expected from the existence of a quasi-steady friction factor ($H_z < 1$), compared to that of the frictional condition.

ii) Time of flow closure (T_q) --- The effect of the quasi-steady friction factor on maximum surge energy (i.e. ER_{qss}) increases with the time of flow closure (T_q). Instantaneous flow closure always induces the least energy reduction (ER_{qss}).

2) Flow with increasing deceleration ($S > 1$) ---

Conditions of increasing deceleration flow can be observed in Fig. 7.7c for $S = 2$, or from upper part of Fig. 7.8 for $S > 1$.

i) Friction-insensitive zone --- For flows with increasing deceleration, the impact of the quasi-steady friction factor is usually negligible during the surge period (ER_{qss}). Its effect is less than 10 % for $H_z < 0.2$ even for laminar flow, and virtually zero for the normal turbulent flow condition (less than 10 % for $H_z < 2.5$ with $R = 10^6$).

In the analysis of steady friction transients, it was observed that friction is irrelevant to surge energy for flows with $S > 1$, for low to moderate friction head (Refer to $H_z < 1$ in Figs. 6.7 and 6.8). Consequently, no

significant energy reduction due to the existence of friction should be expected inside this friction insensitive region, even with the inclusion of the unsteadiness in the quasi-steady friction factor.

- ii) Time of flow closure (T_q) --- Generally, the quasi surge reduction (ER_{qss}) is most significant at $T_q = 4$. In the earlier analysis (Fig. 7.5b), it was found that the quasi surge transfer (ET_{qss}) rapidly increased with time of flow closure until it reached a maximum at $T_q=4$. Then it began to decrease with T_q (for $T_q > 4$). On the other hand, maximum surge energy (E_{mx1}) continued to decrease with the time of flow closure in a relatively constant and moderate rate. Consequently, this different rate of ET_{qss} vs. E_{mx1} (with respect to T_q) produces two opposite effects for the quasi surge reduction (ER_{qss}) vs T_q .

The smallest value of quasi surge reduction (ER_{qss}) occurs for an instantaneous flow closure for laminar and for low turbulent ($R < 10^4$) flow conditions. Conversely, the condition of an instantaneous flow closure in normal turbulent flow ($R > 10^6$) provides the largest value of ER_{qss} , in which the flow closing phenomenon is primarily governed by the friction-dominant condition ($H_s \gg 1$).

- iii) Shape of flow closure (S) --- For $T_q < 4$, the magnitude of ER_{qss} increases with shape of the flow closure (S). However, this behavior is reversed for $T_q > 4$. These two opposite effects are induced by the same reason as discussed for the behavior of quasi surge transfer (i.e. ER_{qss} vs. T_q).

7.5.3 Peak Decaying Energy

The earlier analysis of time (T) of transient as a relevant parameter (Fig. 7.6a) indicated that the effect of quasi-steady friction on the decay of transients can be better interpreted with a sufficiently long time (T).

For simplicity, only the peak decaying energy at $T = 101$ in a quasi-steady friction system is analyzed. The functional relationship between E_{101} and its governing parameters can be expressed as:

$$E_{101} = F(H_z, S, T_q, R) \quad 7.15$$

Results of analyses are plotted in Fig. 7.9 for various types of flow deceleration:

$$E_{101} = F(H_z, S, R) \quad 7.15a$$

where $S = -2, 1$ and 2 . Laminar flow ($R < 2000$) is adopted.

Three isoquants of quasi decay reduction ($ER_{q\pm\tau} = 0.25, 0.50$ and 0.75) are identified in the Fig. 7.9.

A quasi decay reduction isoquant is presented in Fig. 7.10:

$$\begin{aligned} ER_{q\pm\tau} &= F(H_z, S, T_q) \\ &= 0.5 \end{aligned} \quad 7.15b$$

where $R = 2000, 10^4$ and 10^6 .

The effect of quasi-steady friction factor on reducing this peak decaying energy can be summarized as follows:

7.5.3.1 Flow regime (R) on energy decay

A substantial reduction in peak decaying energy is achieved by the quasi-steady friction factor particularly for the laminar flow regime ($R < 2000$).

On the other hand, the effect of a quasi-steady friction factor on normal turbulent flow with $R = 10^6$ is still relatively small since $ER_{q\pm\tau} < 0.50$, for $H_z < 1$.

7.5.3.2 Shape of flow closure (S) on energy decay

As shown in Fig. 7.10, quasi energy reduction (ER_{qsd}) decreases with the increase in $|S|$ for the laminar or low turbulent flow regimes ($R < 10^4$). The following symmetrical relation may be approximated:

$$ER_{qsd}(+S) = ER_{qsd}(-S) \quad 7.16a$$

For a normal turbulent flow condition ($R = 10^6$), the above symmetrical properties do not hold since:

$$ER_{qsd}(+S) > ER_{qsd}(-S) \quad 7.16b$$

7.5.3.3 Time of flow closure (T_q) on energy decay

i) Low Reynolds Number ($R < 10^4$) --- Quasi decay energy reduction (ER_{qsd}) increases with time of flow closure (T_q) for laminar or low turbulent flow conditions.

For these conditions, ER_{qsd} is least sensitive to instantaneous flow closure ($T_q = 0$) in comparison to other flow closing performances ($T_q > 0$).

ii) High Reynolds Number ($R > 10^6$) --- The ER_{qsd} of an instantaneous flow closure becomes more sensitive to Reynolds Number, compared to other flow closing events.

In this turbulent flow condition, the effect of quasi-steady friction on the energy decay is relevant only for high friction ($H_s > 1$).

7.5.4 Discussion of quasi energy decay

Analyses have been carried out in a comprehensive manner, to investigate the effect of quasi-steady friction factor on transients.

It has been found that the unsteadiness in the quasi-steady friction factor is irrelevant for:

i) flows with increasing deceleration ($S > 1$) during the surge period ($T < T_{mx1}$); and

ii) the normal turbulent flow condition with $R > 10^4$, for both the surge ($T < T_{mx1}$) and decaying period ($T > T_{mx1}$).

7.6 Surge Control Device in a Quasi-steady Friction System

Four common surge control devices are discussed in this section.

7.6.1 Flow regime in surge control systems

7.6.1.1 Turbulent flow regime

i) Water system --- Pipe flow in typical water supply systems is usually turbulent. It is most unlikely that the quasi-steady friction model needs to be used in water systems.

ii) Highly viscous fluid system --- In an oil pipeline or some home water distribution systems (diameter less than 100 mm, in which the Reynolds Number approaches 10^4) with relatively high friction head ($H_f > 0.1$), the adoption of a quasi-steady friction model may have some practical application to describe the transient behavior.

7.6.1.2 Laminar flow regime

i) Air chamber and surge tank are applied only under turbulent conditions, and hence are beyond the present scope.

ii) Applications to valve and pump installation can be found for laminar flow conditions. These are mainly related to machinery, biological circulator and other fluidic devices.

Analysis of hypothetical flow hydraulics suggests that the effect of the quasi friction factor may be relevant only in a pump or valve system having a low Reynolds Number (R).

7.6.2 Transient behavior in surge control systems.

7.6.2.1 Quasi friction affecting maximum surge energy (E_{max})

i) Valve --- The flow closing pattern in a low head valve is equivalent to the hypothetical flow closure with increasing deceleration ($S > 1$). As such, the quasi-steady friction model is not required.

ii) Pump --- Pump flow normally closes in the form of step-wise movement upon the pump trip-out condition. Hence, this type of flow closure should induce significantly higher elastic energy which dominates the transient behavior more than quasi-steady friction.

Therefore, a quasi-steady friction model is not necessary during the surge period ($T < T_{max}$).

7.6.2.2 Quasi friction affecting peak decaying energy (E_T)

i) Valve --- A slow valve closure results in a small transient flow movement after the valve closure. Consequently the quasi-steady friction energy reduction is insignificant.

ii) Pump --- A pump without a check valve will have reversed flow through its rotating element, in which the pipe hydraulics are governed by the water levels between the source and pipe-end reservoirs.

Analyses of the quasi-steady friction model (f_q) are summarized in Table 8.2, to be discussed together with the unsteady friction model (f_u).

7.6.3 Discussion of control device in quasi friction system

It appears that the need for application of a quasi-steady friction model is limited in transient pipe systems with common surge control devices.

The valve system with a rapid closure and the pump system equipped with check valve are the necessary conditions for the consideration of a quasi-steady friction model (f_q) in a decaying transient.

7.7 Conclusions

1) Theoretical background for quasi friction analyses ---

i) Quasi friction transfer (ET_{qz}) --- The previously established mechanical energy transfer concept related to friction, has been used to interpret the effect of a quasi-steady friction factor on general transient flows.

ii) Transient flow movement (Q_m) --- The variabilities of a quasi-steady friction factor affects the transient behavior in two opposing ways. During the early flow closing period, the existence of a quasi-steady friction factor reduces the rate of outflow from the system (i.e. $Q_m(f_q) < Q_m(f)$). This flow situation reverses towards the end of the flow closure (i.e. $Q_m(f_q) > Q_m(f)$).

The netflux $E_w(T)$ is however reduced with the existence of a quasi-steady friction factor. This explains its positive effect on reducing the maximum transient energy. This means that $E_{mx}(f_q) < E_{mx}(f)$.

iii) Evaluation method --- Two criteria have been adopted for evaluating the effect of quasi friction. These are: a) quasi friction transfer (ET_{qz}); and b) quasi energy reduction (ER_{qz}).

The former is based on the actual difference in friction transfer between steady $ET_z(f)$ and quasi-steady $ET_z(f_q)$ frictional conditions. The latter is on the basis of relative comparison with E_{mx} as a denominator, and is adopted for the conclusions to be made below.

iv) Evaluation period --- The effect of quasi-steady friction on transients has been investigated in two different periods. These are: a) surge period ($T \leq T_{mx1}$); and b) decay period ($T > T_{mx1}$).

The existence of non-friction transfer (ET_z) cease to affect the transients after the end of counter-surge period ($T > T_{mx2}$).

2) Parameters governing quasi friction phenomena ---

i) Reynolds Number (R) --- Flow regime in terms of a low Reynolds Number ($R < 10^4$) is a relevant parameter reflecting the characteristics of a quasi-steady friction factor (f_q). However, the effect of quasi-steady friction diminishes for normal turbulent conditions ($R > 10^5$).

ii) Time of Transient (T) --- The effect of a quasi-steady friction factor on the decay of transient is a gradual and relatively slow process. Unless a high friction head prevails a long transient simulation is usually required to obtain a meaningful discrepancy between the two assumptions.

iii) Friction (H_z) --- The existence of adequately high friction is necessary to carry out a proper transient analysis of quasi-steady frictional systems.

iv) Flow closing characteristics (S and T_q) --- The time (T_q) and shape (S) of flow closure are relevant parameters significantly affecting the quasi-steady friction phenomena.

3) Effect of quasi friction during the surge period ---

i) Flows with increasing deceleration ($S > 1$) --- The existence of quasi-steady friction factor does not significantly affect the transient behavior, unless laminar flow and adequately high friction exist.

In general, the effect of quasi friction increases with increasing time of flow closure until $T_q = 4$. Then it begins to decrease with $T_q > 4$.

ii) Flows with decreasing deceleration ($S < -1$) --- With the adoption of quasi surge reduction (ER_{qss}), the quasi-steady friction factor becomes a sensitive parameter for this case, particularly for long flow closing times.

In general, the effect of quasi-steady friction on the maximum surge energy increases with T_q , with $T_q = 0$ as the minimum.

4) Effect of quasi friction during the decaying period ---

i) Low Reynolds Number $R < 10^4$ --- The effect of quasi-steady friction on the peak decaying energy increases with T_q .

ii) High Reynolds Number $R > 10^4$ --- For normal turbulent flow conditions ($R > 10^6$), the maximum effect due to the existence of quasi-steady friction factor occurs for an instantaneous flow closure ($T_q = 0$). For this conditions, the quasi friction is not effective.

5) If a quasi-steady friction model is applied, it should be limited to:

i) a rapid valve closure or a pump with a check valve;

ii) a low Reynolds Number flow ($R < 10^4$); and

iii) analysis of transient energy decay.

6) Incorporation with other unsteady friction models(---)

The quasi-steady friction model discussed in this chapter can be considered as a very elementary unsteady friction model. It should therefore be possible to effectively incorporate present results into other complex and sophisticated unsteady friction models for general transient flows. A thorough understanding of the transient characteristics for the quasi-friction system is very important prior to advancing to the final objective of this thesis.

Table 7.1 Summary of Quasi-steady Fiction Analysis

H _z	S	T _q	R	Maximum transient energy		
				Surge E _{m×1}	Counter surge E _{m×2}	Peak decay at T=101 E ₁₀₁
.1		0	2000	-.95	.85	-.01
1				-.50	.14	-.00
5				-.01	.00	-.00
.1			10 ⁴	-.95	.86	-.06
1				-.50	.26	-.00
5				-.07	.02	-.00
.1			10 ⁶	-.95	.86	-.14
1				-.50	.27	-.01
5				-.09	.05	-.00
.1			∞	-.95	.86	-.80
1				-.50	.29	-.21
5				-.10	.07	-.05
1	1	4	2000	-.09	.01	-.00
	-2			-.11	.01	-.00
	2			-.20	.01	-.00
	1		10 ⁴	-.18	.10	-.00
	-2			-.23	.10	-.00
	2			-.31	.17	-.00
	1		10 ⁶	-.19	.15	-.08
	-2			-.25	.15	-.08
	2			-.33	.23	-.09
	1		∞	-.20	.16	-.14
	-2			-.26	.18	-.15
	2			-.34	.24	-.20

Table 7.2 Summary of Quasi Friction dissipation, $ED_s(f_s)$

H_s	E_s	T_a	S	$0 < T < T_{mx1}$		$T_{mx1} < T < T_{mx2}$	
				$ED_s(f)$	$ED_s(f_s)$	$ED_s(f)$	$ED_s(f_s)$
1	0.5	0	-	0.00025	0.00025	0.0004	0.0005
5	2.5	0	-	0.0035	0.0045	0.00001	0.00001
1	0.5	100	1	0.025	0.050	0.00015	0.00020

Table 7.3 Summary of Quasi Friction Transfer, ET_{qs}

H_s	S	T_a	Surge Transfer		Decay Transfer			
			$0 < T < T_{mx1}$		$T_{mx1} < T < T_{mx2}$		$T_{mx2} < T < T_{mx2}+2$	
			$ET_{ss}(f)$	ET_{qss}	$ET_{sd}(f)$	ET_{qsd}	$ET_{sd}(f)$	ET_{qsd}
1	-	0	0	0	0.61	-0.07	0.75	-0.18
5	-	0	1.6	0.09	2.51	0	2.57	0
1	1	4	0.45	0.11	0.52	-0.02	0.55	-0.04
1	-2	4	0.38	0.14	0.52	-0.02	0.58	-0.01
1	2	4	0.59	0.14	0.55	-0.05	0.62	-0.03

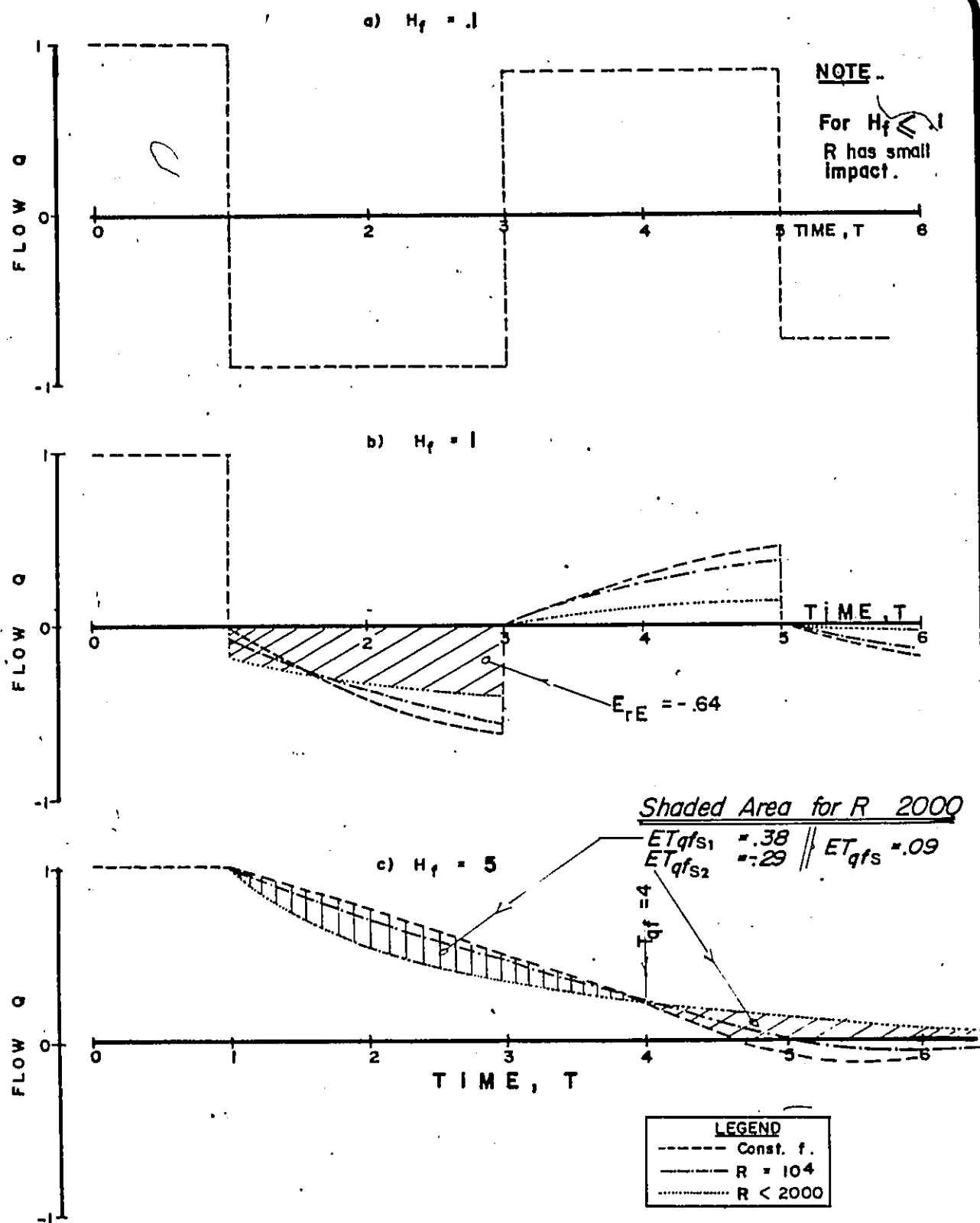


Fig. 7.1 — Transient history of instant flow closure with quasi-steady friction factor

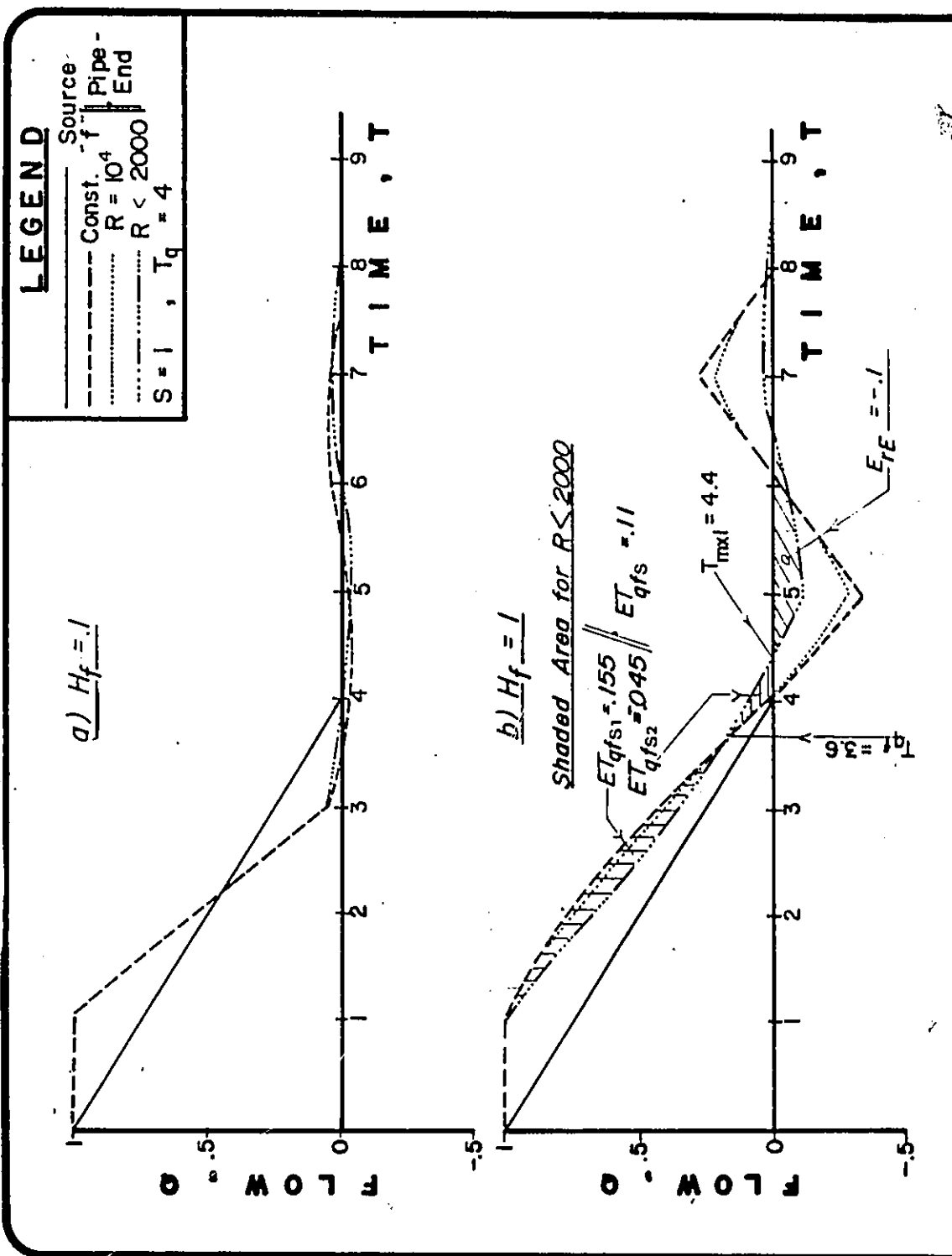


Fig.7.2 — Transient History of Uniform Deceleration Flow with Quasi-steady Friction Factor.

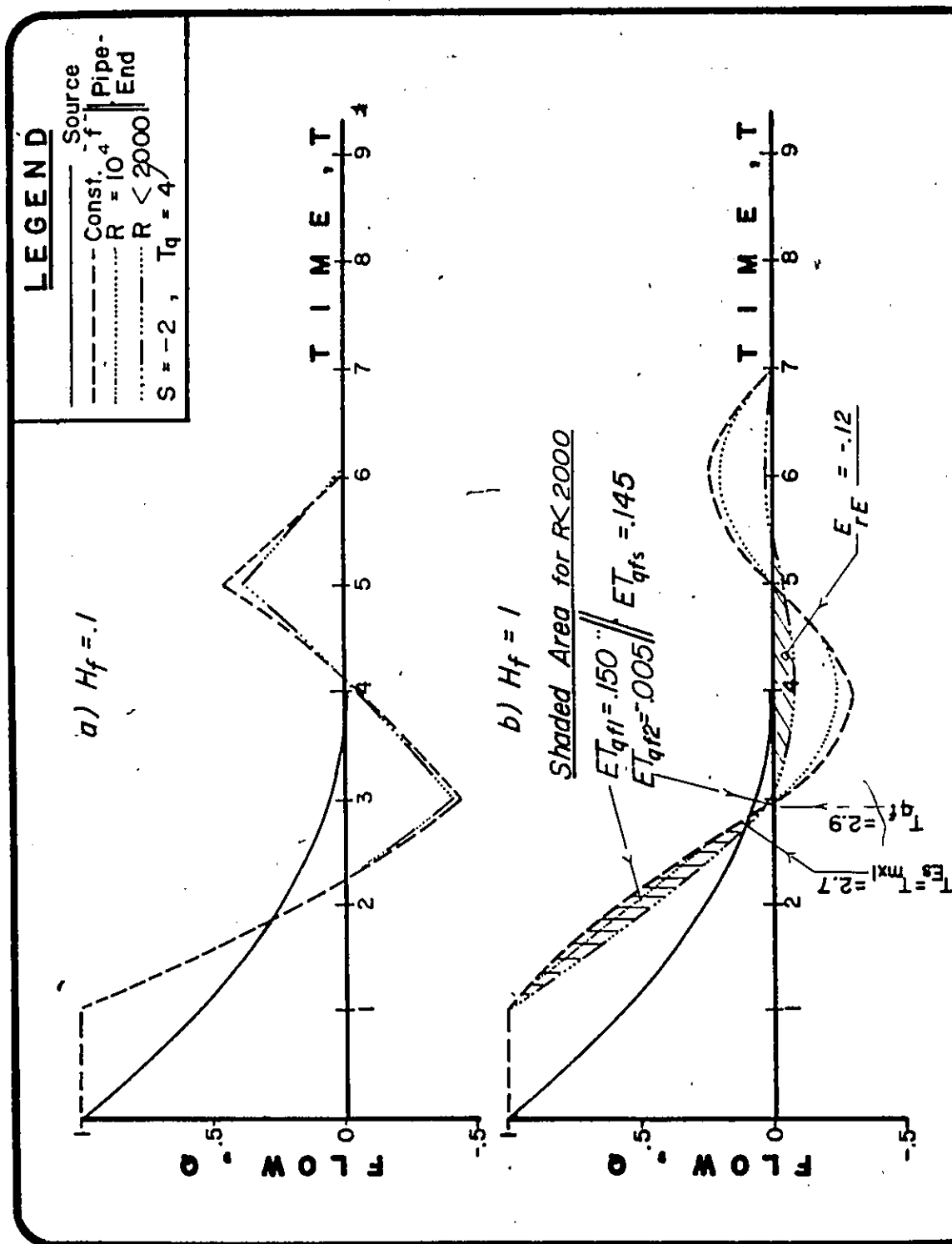


Fig. 7.3—Transient History of Decreasing Deceleration Flow with Quasi-steady Friction Factor.

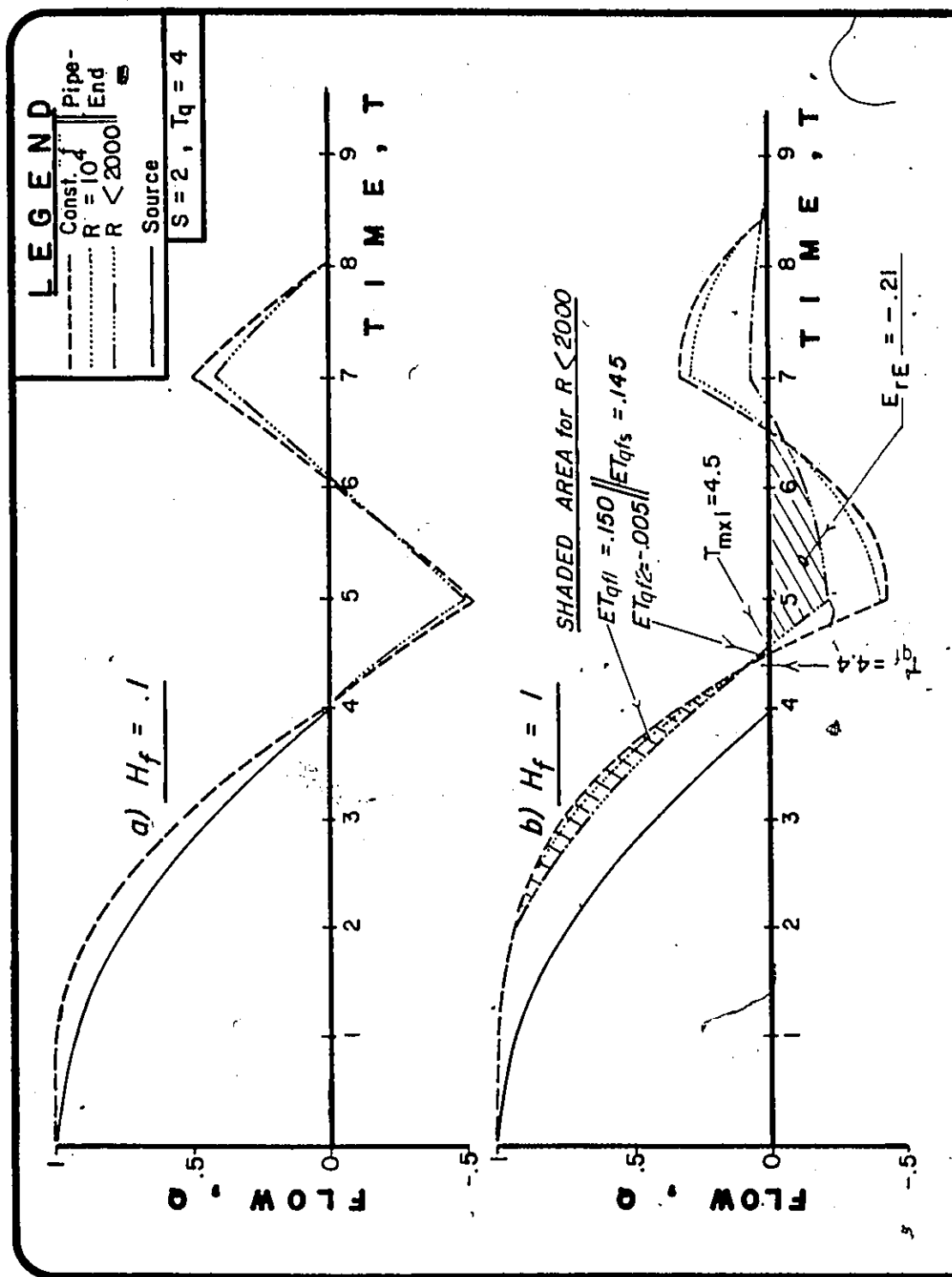
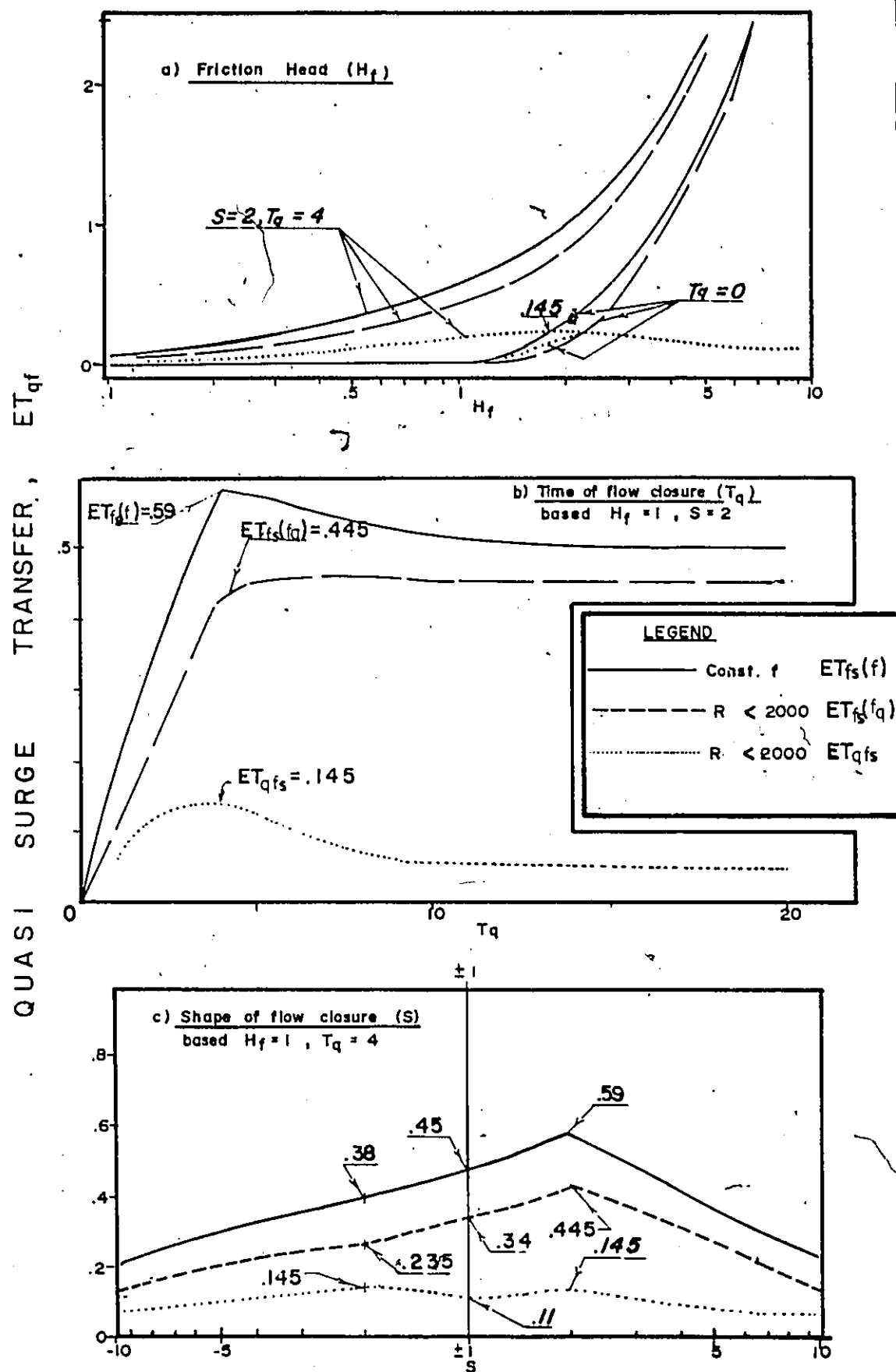


Fig. 7.4—Transient History of Increasing Deceleration Flow with Quasi-steady Friction Factor.



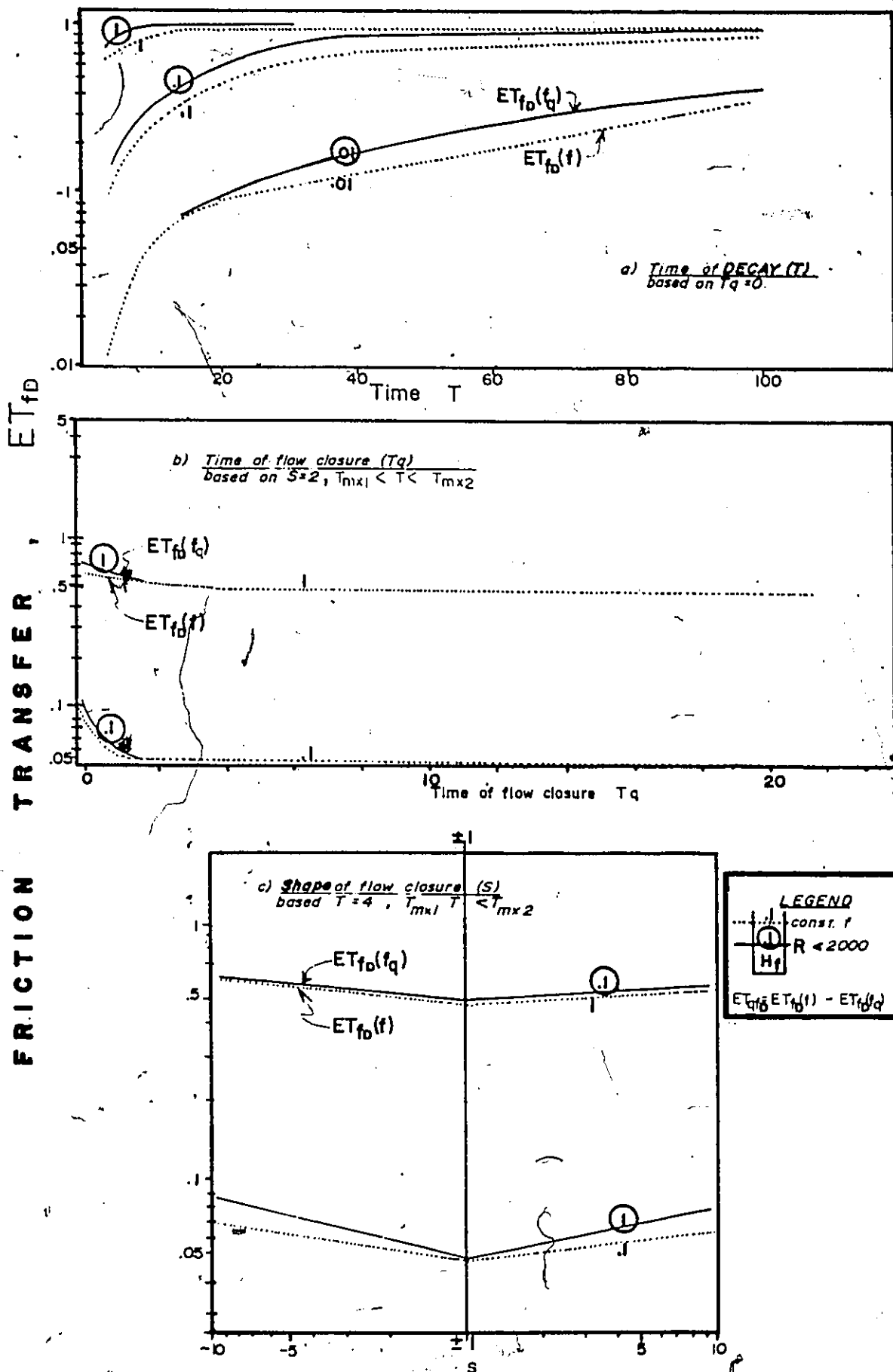


Fig. 7.6 — QUASI DECAY TRANSFER DURING DECAYING PERIOD.

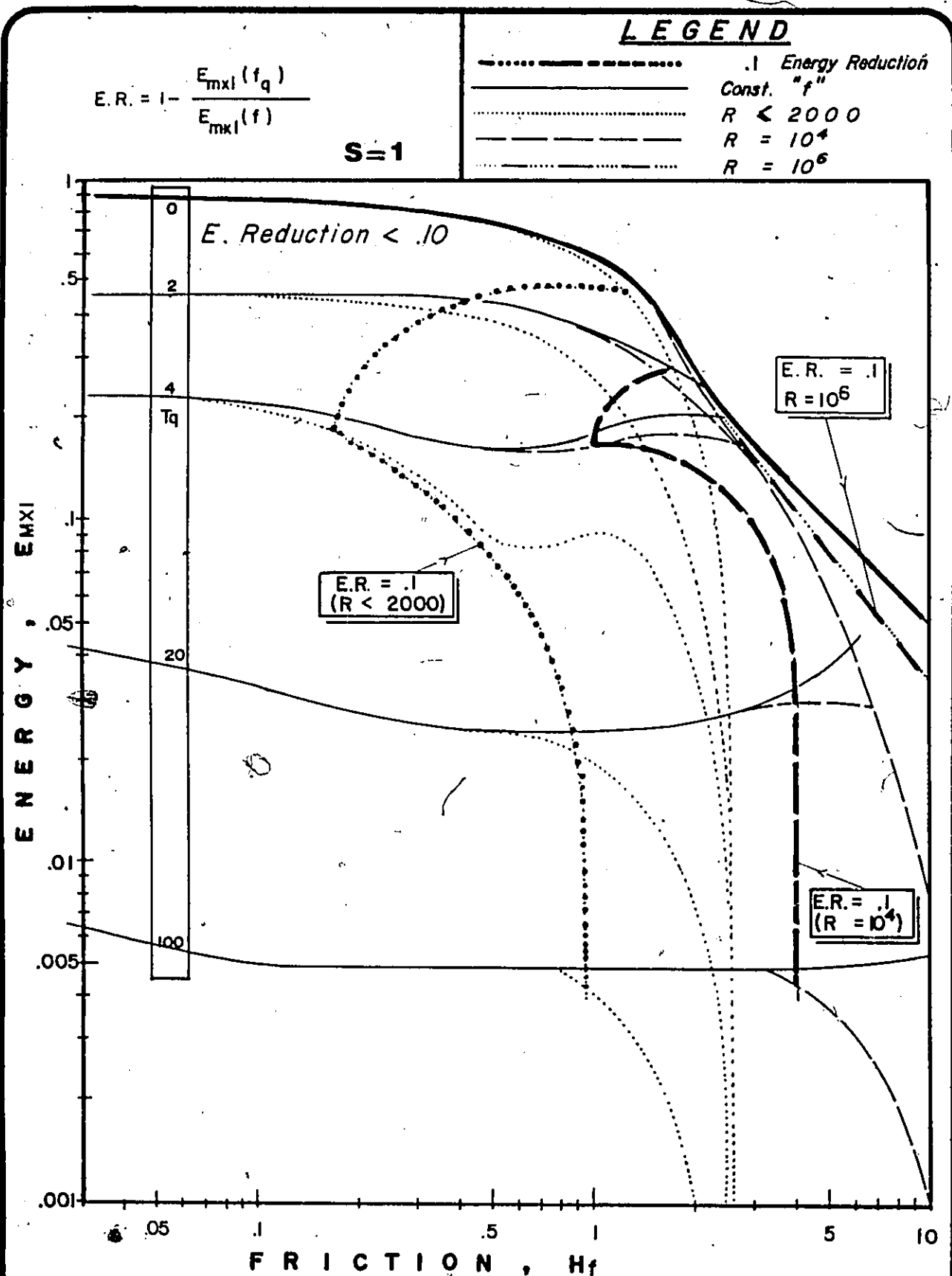
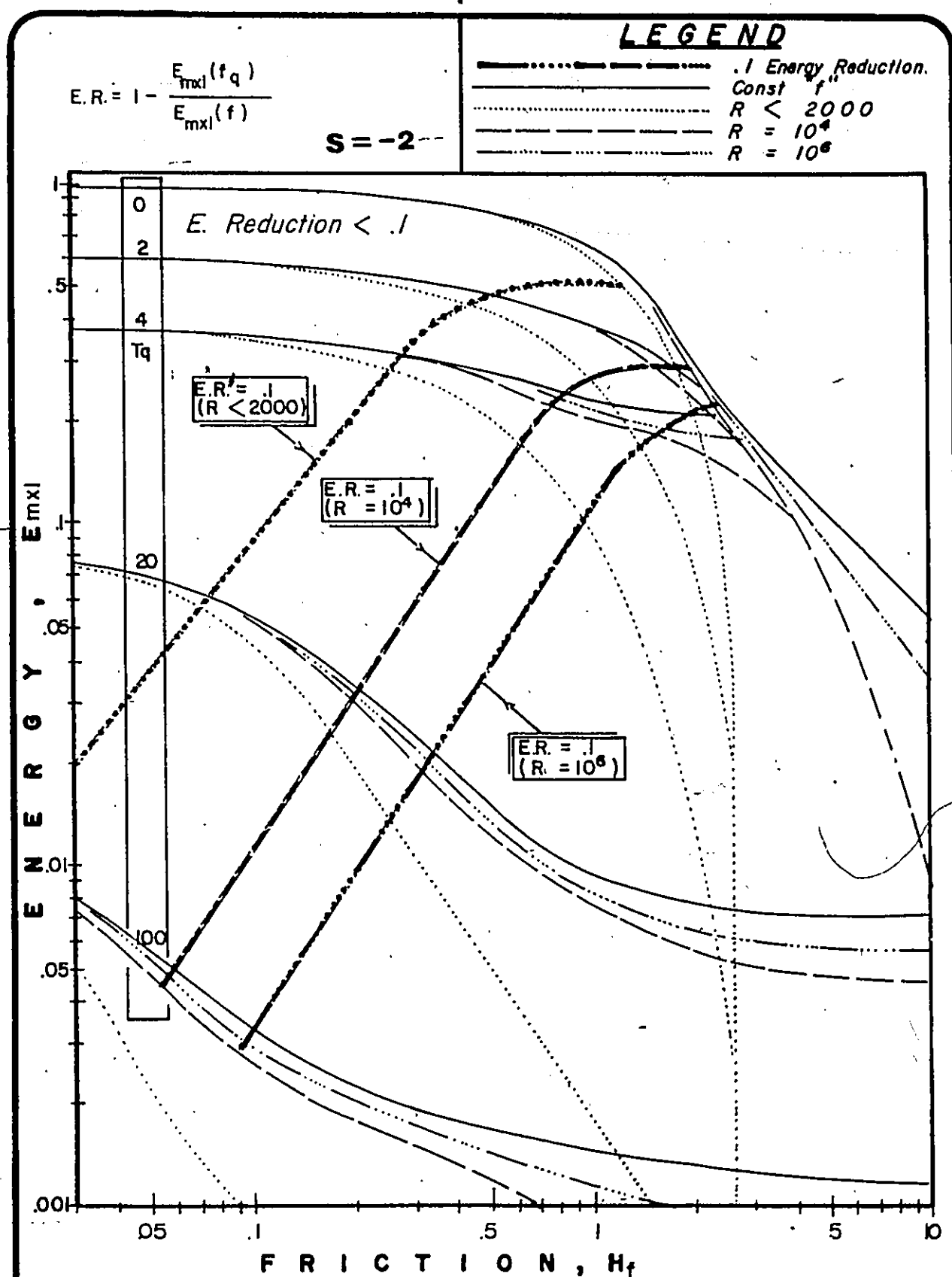


Fig. 7.7 a)–Max Surge Energy induced by Uniform Deceleration Flow with Quasi-steady Factor



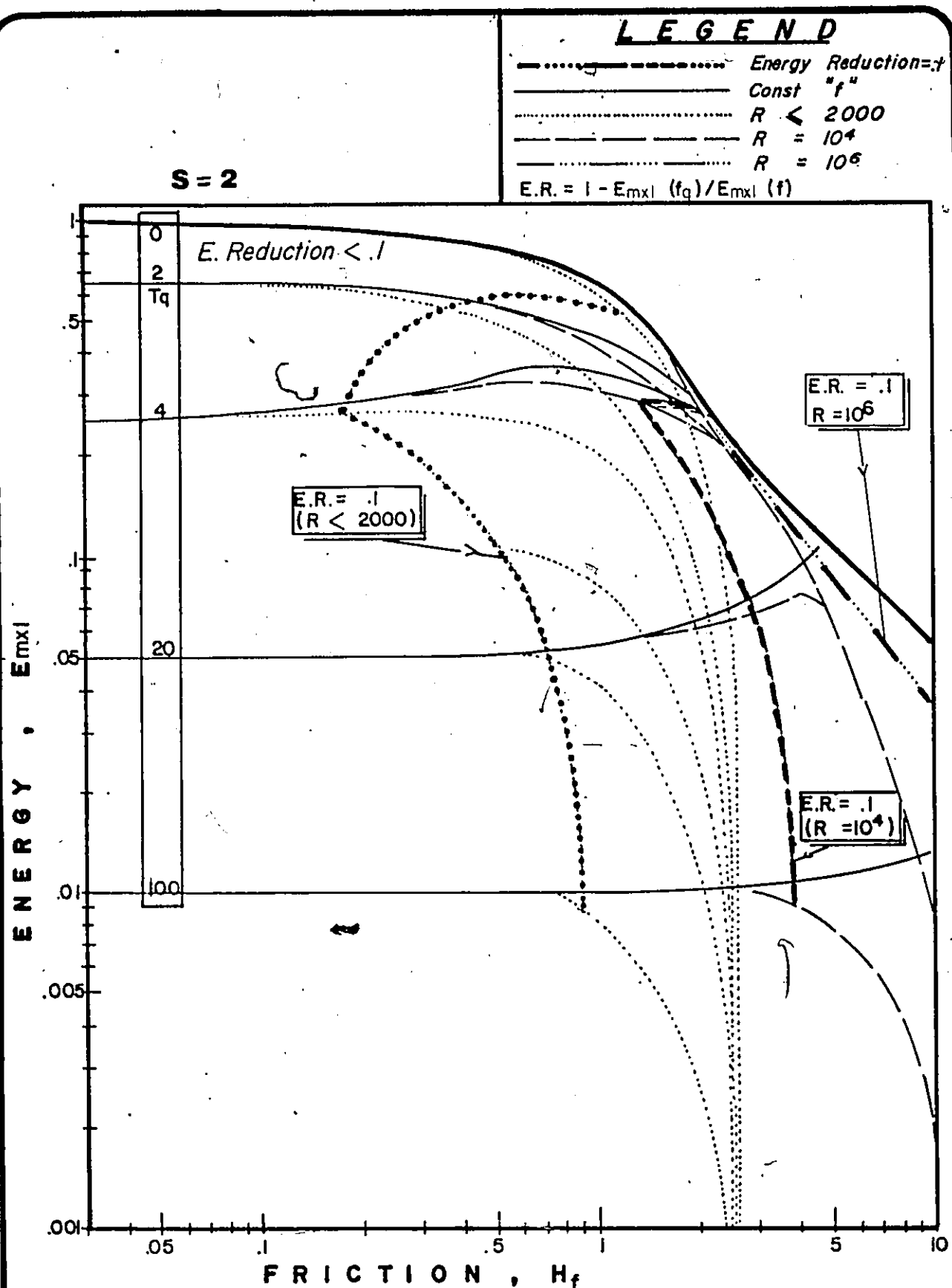


Fig. 7.7c) — Max Surge Energy Induced by Increasing Deceleration Flow with Quasi-steady Factor.

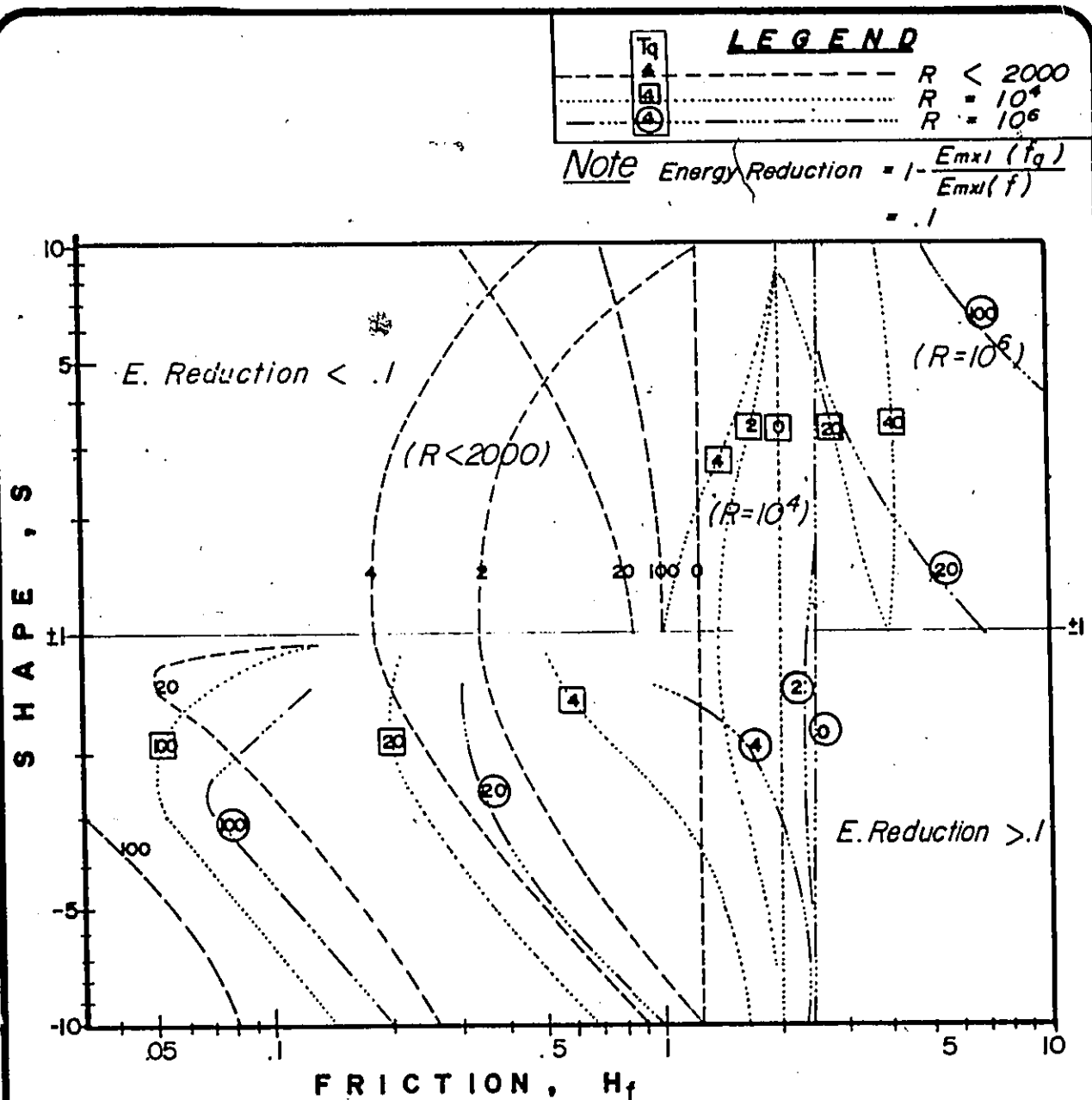
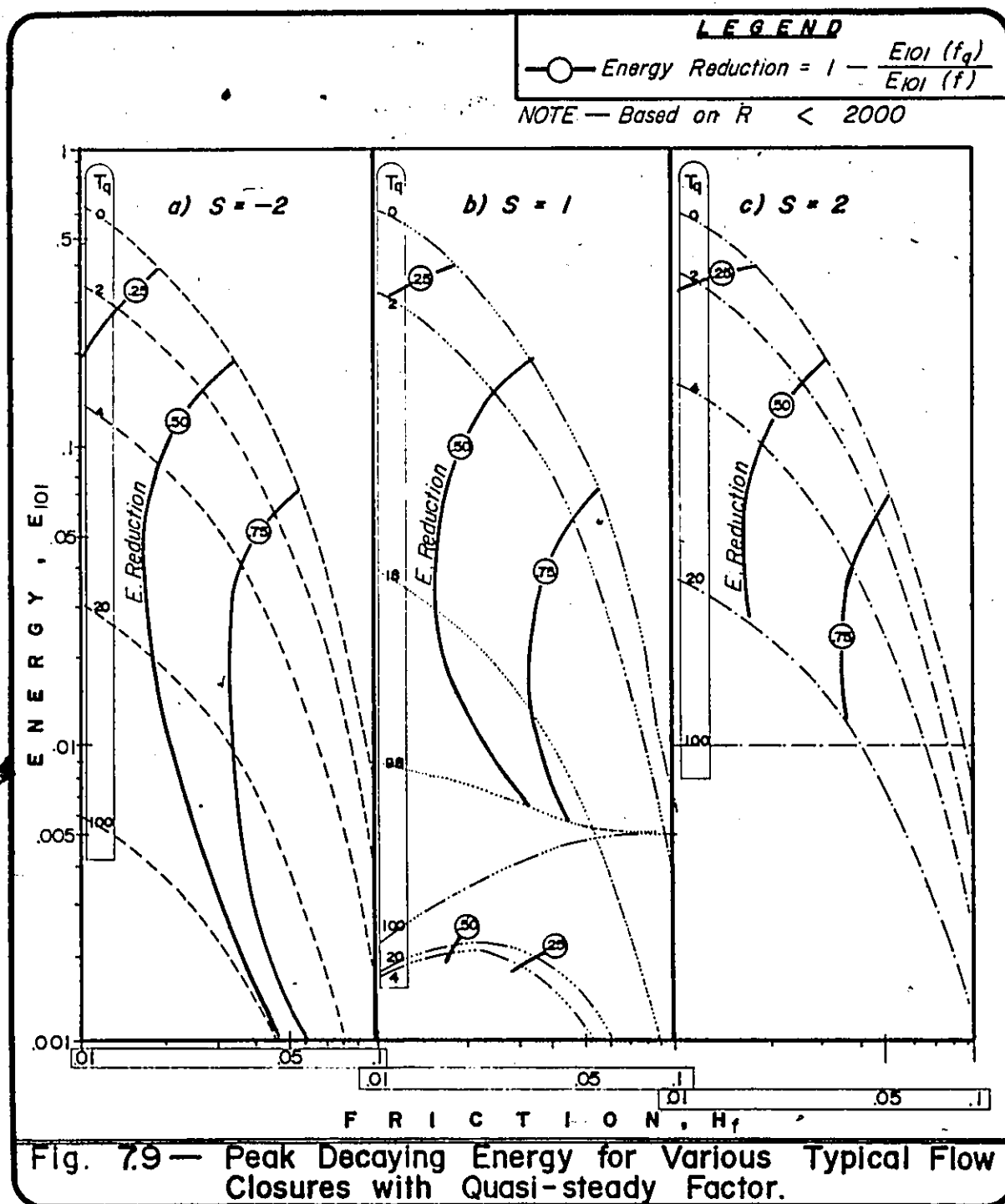
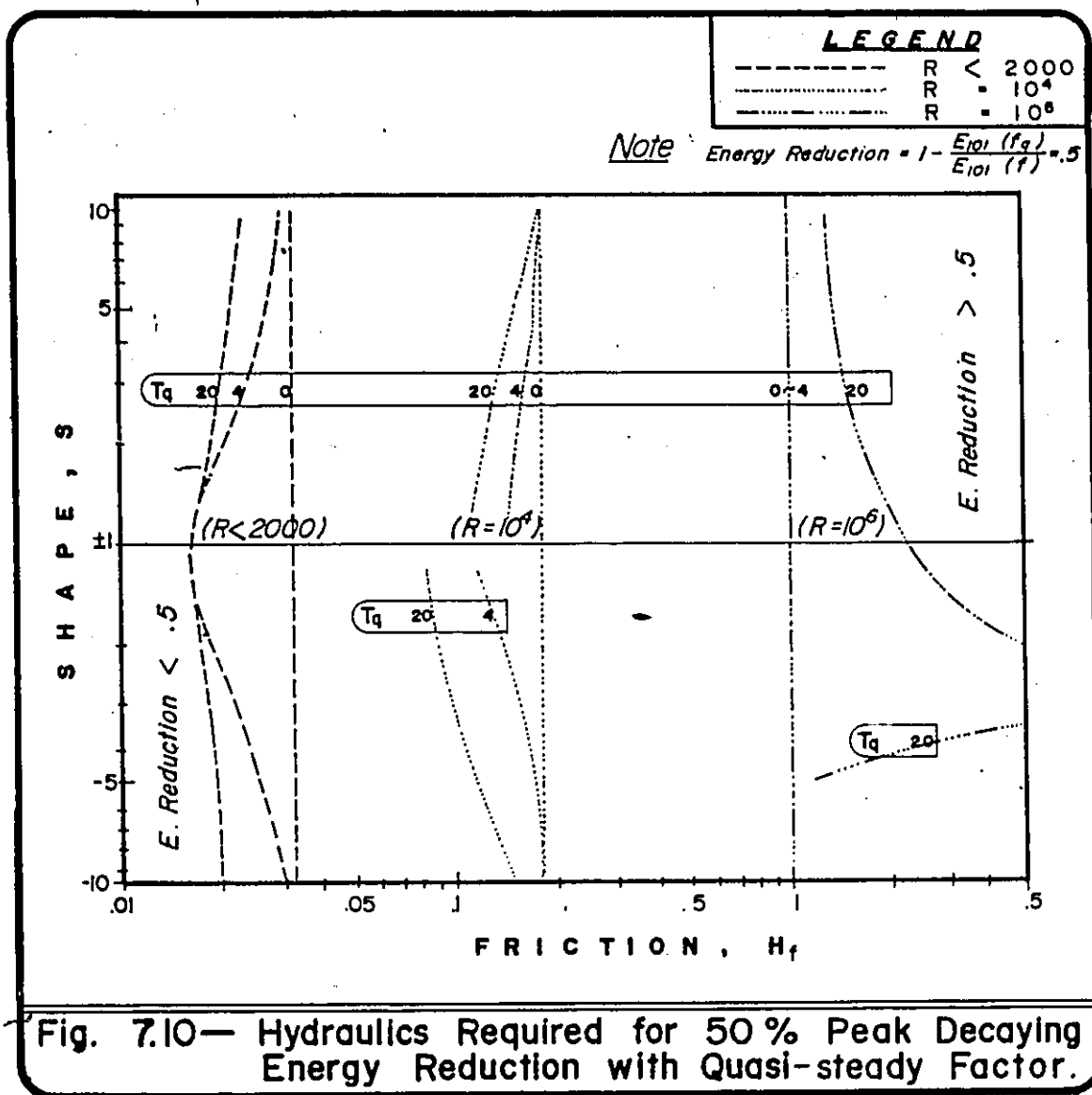


Fig. 7.8 — Conditions required for 10 % Maximum Surge Energy Reduction with Quasi-steady Factor.





8.0 ANALYSIS OF TRANSIENTS IN AN UNSTEADY FRICTION SYSTEM

The distortion of a shear stress profile as a result of rapid unsteady flow motion is the common background of current unsteady friction theories for anomalous energy losses encountered in general transient conditions. Though numerous unsteady friction theories have been proposed, a theoretically sound and practically efficient friction model is still not available for general transient pipe flows (Wylie and Streeter 1978, page 63; and Shuy and Apelt 1983).

The main objective of this chapter is to establish such an unsteady friction model. To do this, transient properties affected by the adoption of current unsteady friction theories are evaluated and the available experimental data utilized. The application of energy balance and various mechanical transfer concepts (non-friction, steady friction and quasi-steady friction) follows in the analysis of this unsteady friction system.

8.1 Current Unsteady Friction Theories

Many unsteady friction theories have been developed based on the flow velocity and acceleration, as shown below:

$$f_u = F(v, dv/dt) \quad 8.1$$

which is in a manner similar to Eqs. 2.1 and 2.2.

8.1.1 Reference from mass oscillations

Analyses of unsteady flow phenomena in U-tube mass oscillation have been conducted and documented in Appendix A.6.

The results of these analyses provide ample supportive evidence for the relationship expressed in Eq. 8.1, in mass oscillating flow systems.

A question now arises as to whether those unsteady friction theories, well established from the early analyses in U-tube system, remain valid in general transient applications. Many unsteady flow properties related to U-tube oscillation have been changed in the pipe system. This is a critical issue and it is re-addressed throughout this chapter, during discussions of various current unsteady friction models adopted for transient pipe flows.

8.1.2 Scope of analysis

The unsteady friction theories discussed below are related to pipe systems with constant pressure head at both source and pipe-end.

8.1.2.1 Hydraulic conditions

Two hydraulic conditions, namely, turbulent and laminar flows, are generally classified according to the steady state flow regime.

1) Current recognition ---

i) Laminar Flow --- It is generally asserted that unsteady laminar flows have been widely studied analytically and experimentally, and that the unsteady flow phenomena are well understood (for example Shuy and Apelt 1983; and Wylie and Streeter 1978, page 63).

ii) Turbulent flow --- It is similarly generally asserted that there is still a lack of understanding unsteady turbulent flow problems, although numerous theories have been introduced [see for the same examples of (i)].

2) Review from available literature ---

However, in contrast to these assertions, a careful review of reported research shows that (Table 2.1):

i) for laminar flow, examples of analysis of unsteady friction factor in laminar uni-directional flow conditions are almost non-existent.

ii) for turbulent flow, considerable effort has been devoted to the investigation of both uni-directional and bi-directional flows, under turbulent conditions.

8.1.2.2 Unsteady friction theories

1) Uni-directional flow system ---

Two unsteady friction theories have been discussed for this case of uni-directional flow. These are:

i) First derivative momentum (dv/dt); and

ii) second derivative momentum (d^2v/dt^2).

2) Bi-directional flow system ---

This analysis is related to the decay of transients in a frictional system after the termination of the incoming flow ($T > T_d$). Three main approaches have been proposed to tackle the unsteadiness of the friction factor for this flow condition, namely:

iii) Linear transfer friction model, which is mainly confined to the laminar flow condition;

iv) Hypothesized viscous friction model, which relates to the analysis in turbulent flow conditions; and

v) Time-dependent friction model, which was initially developed by Zielke (1966) for laminar flow, then has been later extended to turbulent flow conditions by Zielke and various other researchers.

3) Other unsteady friction theories ---

Other unsteady friction theories exist, but are not discussed in this research due to the scope of this thesis. However, it is felt that the above five categories of unsteady friction theories cover the main stream of current research, and others can most likely to be incorporated into one of these theories for the purpose of discussion.

8.1.3 Unsteady friction in uni-directional flow

This analysis focuses on the unsteady flow phenomena in frictional systems during the flow closing period ($0 < T < T_q$). Current unsteady friction theories related to this condition adopt Newton's basic equation of motion which can be expressed as:

$$\begin{aligned} h_{mx1} &= h_{av} + h_f \\ &= [l/g * dv/dt] + [f_u * l/d * v^2/2g] \end{aligned} \quad 8.2$$

where ' h_{av} ' and ' h_f ' are commonly termed inertial head and friction head respectively.

8.1.3.1 First derivative momentum model (dv/dt)

1) Theoretical background ---

Daily et al (1956) derived an expression from the first derivative of flow velocity for the unsteady friction factor. This momentum friction theory has been popular in the last three decades and its application in general transients has been found in the work of many researchers. These include Carstens and Roller (1959), Sarpkaya (1959), Ruus (1981) and Shuy and Apelt (1983). The unsteady friction factor can be estimated from the following relationship:

$$\begin{aligned} f_u/f_q &= 1 + b * (2d/f_q) * v^{-2} * dv/dt \\ &= F(v, dv/dt) \end{aligned} \quad 8.3$$

where the coefficient, 'b', was later determined semi-empirically by Carstens and Roller (1959), as a function of the Reynolds Number:

2) Illustration by examples ---

An example problem was selected within the range of (physical) experimental investigations (Table 2 of Dally 1956) and this has been numerically reproduced by the method of characteristics, with results as shown in Fig. 8.1. Basically, the test is a uniform and slow flow closure with low frictional head in a turbulent flow condition as below:

$$S = 1, T_a = 500, H_x = 0.005 \text{ and } R = 1.3 \times 10^5.$$

3) Some problems associated with this theory ---

Among the many questions which remain unsolved the primary concerns are summarized below:

1) Basic equation used in a uniform closing flow --- The pressure variations simulated from Eq. 8.2 are plotted in Fig. 8.1c. The first component (h_{av}) of the equation is governed by the flow deceleration (dv/dt) which is constant throughout the whole period of the experiment. The second component (h_x) is determined by the flow velocity (v^2), which decreases rapidly with time. Consequently, the inertial component (dv/dt) of Newton's basic equation as adopted by Dally et al, becomes redundant in the analysis of unsteady flow behavior.

The proposed relationship of Eq. 8.3, in which the unsteadiness of friction is governed by the flow deceleration (dv/dt), contradicts these authors' own basic assumption expressed in the Newton Law.

1i) Elastic properties --- The unsteady flow motion of Dally's test can hardly be considered as a rigid body. As shown in Fig. 8.1c, the effect of periodic pressure

fluctuation (period $T_s = 4$ or $t_s = 0.01$ sec.) is substantial, from the beginning to the end of the flow closing period in the test. The strong periodicity is a clear sign of non-rigid flow movement.

iii) Accuracy of measurement --- The broad range of pressure fluctuation raises a suspicion as to whether an accurate measurement can be made during the test. Any slight deviation from the center of data band will lead to a substantially different unsteady friction factor. As a result of this, scattered and conflicted results can be expected.

iv) Dimensional Reasoning --- Eq. 8.3 was derived within the duration of the established phase when the parameter (dv/dt) is constant (Fig. 7 of Daily, 1956). No more tests were performed to verify the relevance of dv/dt in the equation; for example, by changing its values systematically while keeping other parameters constant. Without conducting such tests, the validity of dv/dt as a characteristic parameter governing the unsteadiness of friction factor remains a question.

Apparently, Daily's acceptance of dv/dt as a relevant parameter came from the logic of unsteady friction theories (Eq. 8.1) established from mass oscillations.

v) Scale effect --- Substantial impact due to the scale effect can be found in the application of Eq. 8.3. A comparison is made in two different sized pipes for the determination of the unsteady friction factor (f_u):

- a) Pipe #1, for $d = 13$ mm, $v = 9.75$ m/s, $R = 1.1 \times 10^5$,
 $b = 0.449$ and $f_q = 0.018$

Eq. 8.3 can be simplified for the small sized pipe:

$$f_u/f_q = 1 + 0.001 dv/dt.$$

- b) Pipe #2, for $d = 204$ mm, $v = 0.61$ m/s, $R = 1.1 \cdot 10^5$,
 $b = 0.449$ and $f_a = 0.018$

Then for the larger sized pipe, Eq. 8.3 becomes:

$$f_u/f_a = 1 + 4.18 \, dv/dt.$$

A comparison of these two cases suggests that the unsteadiness of the friction factor does not exist in a high frictional system with rapid deceleration [small pipe in case (a)], while it becomes very significant in a low friction system with slow deceleration [large pipe in case (b)]. Perhaps no logical explanation can be found for this anomalous behavior using existing theory.

4) Discussion of first derivative momentum theory

Adopting a discrete expression for Newton's basic motion equation as a basis for the development of momentum friction theory in a non-rigid water body is not well received in this research. Conflicting results and deficiencies found in Dally's work suggest the need of a constraint in the application of this unsteady friction model in uni-directional flows.

8.1.3.2 Second derivative momentum model (d^2v/dt^2)

1) Theoretical background

The second derivative of velocity has been introduced by Denisov (1972) and Popov (1974). The application of this higher order unsteady friction theory has been broadly accepted, particularly in Eastern Europe. The unsteadiness of friction factor can be predicted as below:

$$f_u/f_a = \exp(-20 \cdot K_1) + 20 \cdot K_1 \cdot \exp(1 + K_1) \cdot (1 + K_2) / (1 + 10 \cdot K_1)$$

8.4

$$= F(v, dv/dt, d^2v/dt^2)$$

where $K_1 = 2*d/(v^2 f_a) * dv/dt$; and
 $K_2 = d^2/(32*v) * v^{-0.5} * (d^2v/dt^2)^{0.5}$

The unsteady friction factor (f_u) may be increased to a factor of 100 times over the quasi-steady value (f_a) as shown in Fig. 2 of Popov's work (1974).

2) Illustration by example ---

An illustration of laboratory work (Fig. 4 of Popov, 1974) is numerically reproduced in this research as shown in Fig. 8.2. Almost identical results are obtained from the basic motion equation (Eq. 8.2) and the present numerical model.

In this example of flow opening, the flow velocity increases rapidly with time as:

$$v = 1075 t^2$$

in which the increase of acceleration is determined by:

$$dv/dt = 2150 t$$

and the second order of flow change becomes a constant:

$$d^2v/dt^2 = 2150$$

Hence, the above second order flow variation is equivalent to a flow opening with $S = 2$.

3) Problems associated with theory ---

The following paragraphs discuss various aspects of analysis in this investigation:

- 1) Elastic properties --- Periodic fluctuations during the flow opening is small and can be neglected for simplicity. Transient behavior during the entire

unsteady flow period can be accurately predicted by Newton's basic equation of motion (Eq. 8.2).

ii) Effect of friction --- The history of unsteady pressure head (h) and its two components [friction head (h_f) and inertial head (h_{av})] have been plotted based on the basic motion equation, as shown in Fig. 8.2c.

Extremely high unsteady friction factor (f_u) was proposed by Denisov (1972) and Popov (1974) at the beginning stage of the test run and then its value decreases rapidly with time. During the whole acceleration period, the friction head (h_f) was very small in comparison with the inertial head (h_{av}). For example,

$$f_u/f_a = 50,$$

at $h_f/h_{av} \ll 0.1\%$. Then,

$$f_u/f_a = 1.1,$$

where $h_f/h_{av} < 5\%$ still holds.

It should be noted that an insignificant friction head (h_f) multiplied many times, as derived by the Denisov 1972 and Popov 1974, is still far too small to be significant.

iii) Direction of flow --- As shown by Popov (1974, Fig. 4), during the second half of the test run, the flow decelerates with characteristics identical to the corresponding accelerated flow movement. The effect of unsteady friction factor was however non-existent. It is not realistic that such a great deviation in unsteady friction behavior would occur between the two corresponding flows bearing the same shape (S) and time (T_a) of flow variations. The only difference is the direction of flow in this case (i.e. acceleration vs. deceleration).

iv) Errors in measurement --- A close examination of the experimental procedure from both articles indicates that a slight disturbance during the low flow period can easily create a huge error in Denisov's or Popov's results. The creation of such a tiny instability is quite normal at the initial stage prior to the established phase of the test run.

v) Dimensional reasoning --- Only one single value of d^2v/dt^2 was conducted for the establishment of Eq. 8.4. This is far from sufficient to justify d^2v/dt^2 as a relevant parameter.

4) Discussion of second derivative momentum theory

Since the laboratory work was established from a highly insensitive frictional zone, the development of an abnormally sensitive unsteady friction factor in Denisov's and Popov's model is highly unreliable.

The adoption of this second derivative friction theory is not recommended.

8.1.3.3 Unsteady friction in surge control system

1) Application in surge tank ---

Sarpkaya (1959) found that more than a 20 % error could be made in the computation of pressure oscillation in a surge tank installation, if the effect of unsteadiness on friction loss is disregarded. The nonlinear differential equation of motion in Sarpkaya's work was solved by means of step-wise integration.

Since its publication this numerical observation has been adopted as a criterion by some researchers to judge the potential effect of unsteady friction factor on other applications of common surge protection devices (Ruus 1981).

2) Illustration by example ---

The example of surge tank application in the article is numerically reproduced by the method of characteristics and the results are shown in Fig. 8.3.

The hydraulic condition of flow variation in the pipeline is equivalent to an increasing flow deceleration with:

$$S = 2, T_a = 175 \text{ and } H_e = 0.006$$

An identical pressure oscillation has been obtained by the present model simulation in the constant friction (factor) system, as computed by Sarpkaya. This validates the consistency of two analytical tools used for the surge tank simulation. Hence, deviations between Sarpkaya and the present research should thus be attributed to the selection of unsteady friction models.

As shown in Fig. 8.3, from the present numerical simulation with quasi-steady friction model, a decrease of 0.5 % and 1.7 % is observed for maximum surge and counter-surge pressure respectively. In contrast to the present finding, Sarpkaya (1959) found an increase of pressure of 30 and 55 % under the same system conditions. The first derivative momentum model was adopted by Sarpkaya. The reasons of these conflicting findings can be found from discussions below:

3) Problems associated with theory ---

The quasi-steady friction model itself is one of the unsteady friction models having a most simple form. The insensitiveness of the present quasi-steady friction simulation can be justified from the following four points:

- 1) Effect of friction --- Reference to the friction classification of Fig. A.5 shows that such a hydraulic condition falls deep inside the friction-free region where the transient behavior is not affected by the

existence of friction at all, and of course not by the unsteadiness of the friction factor.

ii) Flow regime --- The Reynolds Number of the steady state flow condition is in the order of 3×10^4 . Previous analysis of quasi-steady friction effect indicated that less than 10 % pressure reduction would result for even $H_z = 10$ in this highly turbulent condition (see Fig. 7.8).

iii) Common effect of unsteadiness in the transient --- The transient energy is reduced by the existence of an unsteady friction factor, as concluded previously in Ch. 7.0. The opposite effect on an unsteady friction factor (i.e. the increment of transient as derived by Sarpkaya) should never be found in an unsteady flow simulation.

iv) Unsteady friction model --- Dally's momentum friction model was adopted by Sarpkaya. The previous discussion suggests an error due to a scale effect exists in this friction model, in which the unsteadiness of friction factor becomes increasingly important for a large pipe system.

4) Discussion of friction theory application

The gross deficiencies of this surge tank application confirms the non-suitability of Dally's momentum friction theory in the application of general transient conditions. Consequently, the adoption of Sarpkaya's work as a guideline may have to be rethought.

With the completion of analysis in a uni-directional flow, investigation of current unsteady friction theories proceeds in the next subsection to the bi-directional flow condition.

8.1.4 Unsteady friction in bi-directional flow

The elastic behavior of the system has been considered in the unsteady friction theories currently adopted in bi-directional flows. All the anomalous energy losses observed in the physical experiments are attributed to the unsteadiness in the friction factor (f_w).

8.1.4.1 Linear transfer friction model

1) Theoretical background ---

This friction model simulates pressure and flow responses between two points in the system by a transfer function. This has been quite a popular method to investigate the unsteadiness of friction factor for laminar flow conditions, prior to the era of the computer. Names of researchers in this area have been listed in Table 2.1.

2) New theoretical development ---

With the aid of computer applications, Zielke (1966) skillfully integrated the most important property of this linear transfer model, namely the frequency dependent friction, into a function of past transient history. Consequently a practical equation was derived for the current application in the numerical modeling.

3) Illustration by example ---

Almost identical results of transients were obtained by Zielke (1966, Fig. 15) for the case of instantaneous flow closure, between adopting the time-dependent and the linear transfer friction models. The case was extracted from Holmboe and Rouleau's laboratory work (1967, page 178).

4) Discussion of transfer friction model

The further research potential of this linear transfer theory is seriously limited by its complexity and is

virtually superseded by Zielke's work. Its future pursuit is not suggested.

8.1.4.2 Viscous friction model

1) Theoretical background ---

This friction model predicts viscous losses for general transient turbulent flows by dividing the turbulent flow into two components, namely the laminar viscous layer and the inviscid core zone. The theory was proposed by Wood and Funk (1970) with the aid of extensive laboratory work. These results are also experimentally supported by Sauer (1969).

The authors observed the inadequacy of the quasi-steady friction factor in laboratory tests of instantaneous flow closure. Based on the viscous friction theory, these tests suggested that the error is entirely due to the distortion of flow profile across the viscous layer, while the main core flow remained in the same condition.

2) Illustration by example ---

One of Sauer's laboratory tests is numerically reproduced in this research and is shown in Fig. 8.4. The test is based on an instantaneous flow closure with relatively low friction head and a turbulent flow regime. The input data for the hydraulic conditions are equivalent to:

$$H_z = 0.054, T_a = 0 \text{ and } R = 4 \cdot 10^4.$$

As shown in Fig. 8.4d, results of their analyses indicate that the quasi-steady model indeed provides a much less effective transient decay than that observed from the experimental works. In this test, the adjusting factor (f_u/f_a) has been found to be in the order of 1.5.

The question remaining here is whether this observed energy loss can be explained by the proposed 'viscous model'.

3) Problems associated with theory ---

Two points related to the decay of transient energy in works from Wood, Funk and Sauer are discussed below:

1) Effect of surge wave travel --- The logic of increasing energy losses based on this theory is that the shear stress distribution is disturbed by the rapid surge wave travel. If this is so, the above modification factor (f_u/f_q) should become unsteady (T) and spatial (X), depending on the movement of surge wave.

An unsteady friction factor (f_u) up to 3 times the quasi-friction factor (f_q) has been derived by Wood and Funk (1970, Fig. 15) according to the average deviation observed during the whole test period. Then the adjusting factor (f_u/f_q) is applied homogeneously to the whole pipe (X) and the whole duration of the transient (T).

The homogeneous modification (f_u/f_q) of friction factor adopted in this 'viscous theory' suggests the effect of surge wave travel has not been well addressed.

1i) Effectiveness of viscous flow --- In the small piping system of Sauer's laboratory work (13.6 mm), the flow conveyed in the laminar layer was of the order of 1 % of the total flow during the steady state condition. In a normal water system with 300 mm pipe diameter, the laminar flow layer was reduced far below 1 % of the core flow conveyance.

A question remains as to whether or not it is reasonable to attribute all the significant unsteadiness of friction (f_u/f_q) to any imaginable velocity distribution in an insignificant laminar layer.

4) Discussion of viscous friction theory

As a result of the above discussions, it is suggested that the abnormal flow distribution inside the thin laminar layer may not appear to be the true cause for the decay of transients. The independence of time and location for the unsteadiness of friction (f_u/f_q) cannot be justified for highly unsteady and spatially bi-directional flow.

8.1.4.3 Time-dependent friction model

1) Theoretical background----

Zielke (1966) presented an equation for the estimation of the time-dependent friction factor (f_u) which can be incorporated into the method of characteristics to compute the hydraulic transient phenomena in frictional piping systems. The theory was originally developed for laminar flow and was later extended to turbulent flow conditions as claimed by Zielke.

The friction factor $f_u(T)$ at an instantaneous T can be derived from the following equation of Zielke:

$$f_u(T) = f_q(T) + f_w(T) \quad 8.5$$

$$\text{where } f_w(T) = 0.5 * f_q(T) * WZ(T) \quad 8.5a$$

$$\text{in which } WZ(T) = 1/V(T) * \int_0^T [dV(Z)/dT] W(T-Z) dZ \quad 8.5b$$

Basically the time-dependent friction factor $f_u(T)$ consists of two components:

- i) the component due to the quasi-steady flow motion ($f_q(T)$), which can be directly computed from the Darcy equation; and

ii) the component due to the unsteady flow motion ($f_u(T)$), which must be indirectly estimated from the combination of the quasi-steady flow motion ($f_q(T)$) with the weighted past velocity changes ($WZ(T)$).

The weighting function $W(Z)$ has been derived by Zielke as follows:

$$\begin{cases} W(Z) = e^{-26Z} + e^{-71Z} + e^{-135Z} + e^{-219Z} + \dots; & Z > 0.02 \\ W(Z) = 0.28Z^{-0.5} - 1.25 + 1.06Z^{0.5} + 0.94Z + \dots; & Z < 0.02 \end{cases} \quad 8.6$$

where $Z = 4 \nu * dt/d^2$.

This function (Eq. 8.6) can also be graphically expressed as shown in Fig. 8.5. As such, the overall relationship of Zielke's equation is governed by the Reynolds Number and the past history of the flow velocity change. This method of flow simulation is termed the time-dependent friction model because the time history of the transient is heavily weighted.

For a slow unsteady flow movement the weighting function $W(Z)$ approaches zero, resulting in an appropriate condition for application of the quasi-steady friction model. Zielke's model is therefore a convenient way of incorporating friction effects into this work.

2) Experimental Validation of time-dependent friction model

1) Holmboe's laboratory work --- During the development of the time-dependent friction theory, a physical test conducted by Holmboe (1964) was used to verify the model by Zielke (1966, Fig. 15). This experimental work is numerically reproduced here and is presented in Fig. 8.6. Basically, the test has been carried out for an instantaneous flow closure ($T_d = 0$) in a laminar flow system ($R = 640$) with a low friction head ($H_f = 0.054$).

The results of the physical test correlate quite well with those of the time-dependent friction model (Fig. 8.6d). During the whole period of the simulation, the pressure at the source (i.e. $H_m(T, X=0)$, for $T < 15$) has been found to be closely consistent for these two methods of analysis (numerical vs. laboratory).

ii) Sauer's laboratory work --- Zielke's friction model was also tested with the laboratory work carried out by Sauer (1969, page 49 - 52) in the present research. The data and the physical test results have been documented in Fig. 8.4. Basically, the hydraulic conditions of the test are similar to the above of Holmboe's work ($T_a = 0$ and $H_a = 0.054$), except it is in a turbulent flow regime ($R = 4 * 10^4$).

As shown in Fig. 8.4d, very comparable results between the numerical (Zielke's model) and physical (Sauer's observation) simulations have been found from their histories of peak decaying pressure at the source (i.e. $H_m(T, X=0)$, for $T = 1, 3, 5, \dots$).

3) Problems associated with theory ---

Two concerns are raised during the review and analyses of Zielke's work (1966). These are:

i) The weighting function $W(Z)$ was extended from the frequency component of U-tube mass oscillations, suggesting a strong tie between U-tube and pipe physical phenomena during the unsteady flow condition.

Analyses of U-tube unsteady flow movements (Appendix A.6) does not agree with this synonymous relation.

ii) All anomalous decays of transients were attributed to the existence of a time dependent friction factor (f_a), and this has only been calibrated by the transient pressure (H_m) vs. time (T). No attempt was made by Zielke to interpret the time history of the

unsteady friction factor (i.e. $f_u(T)$ vs. T), during the entire development of the theory.

The validity of this time dependent friction theory cannot be accepted if the logic of the time dependent friction factor (f_u) itself is not valid.

4) Discussion of time dependent friction model

It is concluded that the unsteady and spatial transient behavior of time-dependent friction model provide a better theoretical basis for unsteady friction losses, compared to other current unsteady friction models of bi-directional flows. The close correlation of transient pressure history between numerical and physical observations for both laminar and turbulent flow conditions further confirms its value for practical applications.

However, the absence of interpretation of the time dependent friction factor suggests incompleteness in this time dependent friction theory. A thorough analysis of the physical phenomena, including the unsteady and spatial properties of pressure, flow and friction factor, will be carried out later in this chapter during the development of an unsteady friction model for the general transient condition (consisting of both uni-directional and bi-directional flows).

8.1.4.4 Evaluation of other time-dependent friction models

The computation procedure of Zielke's time dependent friction is usually tedious and time-consuming. This raised some difficulty in execution of the computer models in early days when the capacity of computer hardware was limited. Considerable effort has been devoted by several researchers to find alternative solutions. Among these, two simplified methods of the time-dependent friction model are discussed below.

1) Trikha's simplifying function ---

Trikha (1975) reformulated the time-dependent friction model with reduced computation steps and simplified weighting function. Its application has been recommended and actually adopted by several researchers (e.g. Shur and Apelt, 1983). Trikha's weighting function can be approximated as shown in Fig. 8.5 and is expressed as:

$$W(Z) = 40 e^{-20000Z} + 8.1 e^{-200Z} + e^{-26.4Z} \quad 8.7$$

As shown in the figure Trikha's function is almost identical to Zielke's original function for $Z > 10^{-4}$. For $Z < 10^{-4}$ the curve of Eq. 8.7 becomes horizontal and a gross deviation from Zielke's work is expected. Unfortunately, a majority of hydraulic systems just fall into this flat part of the curve (see the applications of Shuy and Apelt, 1983, also marked in Fig. 8.5).

2) Hirose's weighting function ---

Hirose (1971) introduced an alternative weighting function for the application of Zielke's time-dependent friction model in turbulent flow, and its application has been recommended by some researchers (e.g. Chaudhry 1979, page 27). The new function is:

$$W(Z) = 1.414 W_1 * 2^{0.5Z} \quad 8.8$$

where W_1 was suggested by Hirose to be derived from future laboratory tests.

The weighting function (Eq. 8.8) is graphically presented in Fig. 8.5 with respect to W_1 varying from 0 to 7000. Hirose's numerical experiment is reproduced in this research as shown in Fig. 8.7. The test has been carried out for an instantaneous flow closure ($T_a = 0$) with low flow velocity ($v = 0.2$ m/s), low friction head ($H_f = 0.0065$) and turbulent flow ($R = 5.4 \times 10^4$) conditions.

Steady (f), quasi-steady (f_q) and Zielke's original time-dependent (f_d) friction models are also used for present transient simulations for the sake of comparison.

3) Problems associated with theory ---

Hirose's work (1971) is critically reviewed here and some comments are made as follows:

i) Arbitrary weighting function --- Hirose (1971) claimed that the simplified equation (Eq. 8.8) possesses characteristics similar to original Zielke's work (1966) as expressed by Eq. 8.6. As shown in Fig. 8.5, this is not true. The slope of Hirose's function is much steeper than that of Zielke's, and there is no relationship at all to link the two.

A close re-examination of the logic of Hirose's work suggests that the time-dependent friction theory may have been misinterpreted. The original $W(Z)$ vs. Z curve is plotted on normal scale graph paper (shown at the top-right corner of Fig. 8.5). A casual view of the graph could result in misreading it because for a highly decreasing function in laminar flow the turbulent flow should have the advantage of converging to zero much faster (as indicated by Hirose 1971, page 17). Comparing the same function replotted in the log-log scale chart, the facts are exactly opposite.

ii) Unrealistic transient simulation --- In a low frictional ($H_f = 0.0065$) and moderate turbulent ($R=5.4 \times 10^4$) system of Hirose's numerical example, the pressure vs. time curve should be a rectangular shape at the early decaying period. This can be verified by the numerical simulations with various frictional models (as shown in Fig. 8.7d), and by the physical simulations with similar hydraulic conditions (see Figs. 8.4d and 8.6d).

As shown in Fig. 8.7d, the pressure variations (e.g. $W_1=7000$) are not rectangular; instead they are sinusoidal in shape. Furthermore, the transient pressure with $W_1 = 0$, which is theoretically the same as the quasi-steady friction model, does not respond correctly. All these unreasonable transient behaviors can also be found in Figs. 6 - 9 of Hirose's work (1971).

iii) Non-effective decay --- Actually, the low flow velocity (0.2 m/s) in the system produces very small friction transfer, and possibly a tiny energy loss when flow reversal occurs. Consequently, the decaying process of transient energy is slow and inefficient (Fig. 8.7). The test conducted by Hirose (1971) is not an ideal one, even for simple demonstration purpose.

4) Discussion of other time friction models

It is concluded that the Trikha simplified time-dependent friction model needs to be applied with caution due to its constraint in certain hydraulic ranges. It is also concluded that Hirose's alternative weighting function should not be used without substantial improvement and modification.

Zielke's original model is adopted throughout this research, due to the requirement of a broad range of pipe hydraulics and the availability of a large computer system.

8.1.5 Theories of unsteady friction

Two unsteady friction concepts have traditionally been adopted for the analysis of unsteady friction phenomena, including the transient pipe flows.

8.1.5.1 Rate of change of uni-directional flow motion

1) Theoretical background ---

Many previous workers assume that the unsteadiness of the friction factor increases with the increment of flow acceleration or deceleration rates (Shuy and Apelt 1983 and many others). This can be written as:

$$f_u/f_a = b * d^2V/dT^2 \quad 8.9$$

where 'b' is the coefficient. The relation in the equation can be expressed more specifically as:

$$\{ d(f_u/f_a)/d|S| > 0; \text{ and} \quad 8.9a$$

$$\{ d(f_u/f_a)/dT_a < 0 \quad 8.9b$$

where the shape (S) and time (T_a) of the flow closure reflect the rate of deceleration (d^2V/dT^2).

2) Problems associated with theory ---

It can however be questioned whether this concept (Eq. 8.9) is valid for transient conditions. The following discussion views this issue from previous analysis of quasi-steady friction factor (f_a), which is an unsteady friction model with an elementary form. The characteristics of f_u/f_a are reflected here by ET_{qzs} or ER_{qzs} , representing the actual or relative effect of the unsteadiness of friction.

.i) Shape of flow closure (S) --- Elastic energy (E_m) in a pipe system increases rapidly with the shape of flow closure (S). As shown in Fig. 7.5c, the effect of unsteady friction (ET_{qzs}) on transients diminishes with the increasing shape ($|S|$) of flow closure. This contradicts the relation of Eq. 8.9a.

Actually, at the instant of flow closure, ($|S| = \infty$) which is the highest rate of deceleration, the netflux within the surge period is a constant ($E_w = 1$) and is independent of the friction in the system, and

therefore of the potential effect from the unsteady friction factor ($ET_{qss} = 0$ for $H_s < 1$, in Fig. 7.5a).

ii) Time of flow closure (T_q) --- Referring to Fig. 7.5b, the unsteadiness of the friction factor (ET_{qss}) increases with the decreasing time of flow closure until $T_q = 4$. This agrees with Eq. 8.9b.

This effect reverses with faster flow closures for $T_q < 4$, which is opposite to Eq. 8.9b.

iii) Relative energy increase --- Results summarized in Fig. 7.8 are used for the present discussion. For flow with increasing deceleration (i.e. $S > 1$) a two-staged opposite effect of unsteady friction (ER_{qss}) occurs, in a manner similar to that described in Fig. 7.5b.

However, for flows with a decreasing deceleration (i.e. $S < -1$), the effect induced by the unsteadiness (ER_{qss}) of friction decreases with the rate of deceleration (in both S and T_q), in contrast to those relations expressed in Eqs. 8.9a and 8.9b.

3) Discussion of rate of unsteady motion

It appears that the concept in which the unsteady friction effect (f_u/f_q) increases with the deceleration rate is not confirmed by the most elementary unsteady friction system (i.e. quasi-steady). Therefore, the use of this concept for other more sophisticated unsteady friction models needs to be rethought.

8.1.5.2 Effect of unsteady friction on general transients

1) Theoretical background ---

Another conventional concept is that the unsteadiness of the friction factor may affect the transient energy in two opposite ways, depending on the frequency and phase of unsteady motion (Kita et al 1980).

i) The unsteadiness of friction can result a positive damping effect as expressed below:

$$E_m(f_u) < E_m(f) \quad 8.10a$$

ii) In contrast to the above, a negative damping effect due to the unsteady friction factor has frequently been reported (Kita et al 1980):

$$E_m(f_u) > E_m(f) \quad 8.10b$$

2) Problems associated with theory ---

Previous analysis in the quasi-steady friction system concluded that only a positive effect on reducing transient energy exists for a quasi-friction factor. A critical review of current unsteady friction theories suggests the following potential causes for this confusion:

i) The omission of elastic energy in the interpretation of physical model test results may 'boost' the apparent unsteadiness of the friction factor on transients in either a positive (Eq. 8.10a) or a negative way (Eq. 8.10b).

For example, unreasonable distortion of shear stress in transient flow has been concluded by Daily (1956) and Carstens and Roller (1959).

ii) Conversely, the transient energy increases with friction, i.e. $E_m(H_z > 0) \gg E_m(H_z = 0)$. This exists for both uni-directional and bi-directional flows and can be explained with an understanding of periodic fluctuation characteristics (referring to Ch. 6.0).

Instead, in order to explain this property (Eq. 8.10b) observed from laboratory work, a reversed stress profile has commonly been proposed (e.g. Fig. 31, Sauer 1969).

3) Discussion of effect of unsteady friction

Present analysis clarifies that the existence of an unsteady friction factor definitely reduces the transient energy (i.e. $f_u/f_a > 1$). The presence of elastic energy and periodic fluctuations contributes to the confusion in current unsteady friction theories.

8.1.6 Discussion of current unsteady friction theories

Discussions from current unsteady friction applications have provided sufficient evidence to conclude that:

i) Uni-directional flow --- Current unsteady friction models adopted for the uni-directional systems are not justified, and their use as general guidelines for analysis should be discontinued.

ii) Bi-directional flow --- Zielke's time dependent friction is preferred to other current unsteady friction models for the analysis of bi-directional flows, for both laminar and turbulent flow conditions.

iii) Some unsteady friction concepts traditionally adopted for general transient conditions are actually misinterpreted. Among these,

a) one is the unsteadiness of friction increasing with the deceleration rate; and

b) the other is the unsteady friction factor affecting transient in opposite ways.

A thorough understanding of these current theories is necessary prior to the selection of a proper unsteady friction model, as well as to the borrowing (with courtesies) of their experimental data and results for use in present research.

8.2 Energy Relation in an Unsteady Friction System

8.2.1 Selected unsteady friction theory

Prior to the selection of an unsteady friction theory, the following remarks need to be addressed:

8.2.1.1 Properties of an unsteady friction model

An unsteady friction model should consist of these properties:

- i) It should be applicable throughout the whole transient period, covering periods during and after flow closure.
- ii) Besides predicting an accurate transient history, this model should also be capable reflecting the real transient properties, including the energy transfer and dissipation mechanisms.

8.2.1.2 Selection of an unsteady friction model

The time-dependent friction model (Zielke 1966) has certain advantages over other current unsteady friction models, for the application in general transient analysis. These include:

- i) Pioneer's's opinion --- Wylie and Streeter (1978, page 63) endorsed the adoption of Zielke's time-dependent friction model to the laminar flow condition.
- ii) Physical model verification --- A history of transient energy decay can be realistically predicted by this model for both laminar and turbulent conditions (8.1.4.3).
- iii) Numerical model incorporation --- This friction model has been incorporated into the method of characteristic since 1966. Most computational

difficulties or instability problems should have been removed.

iv) Current models --- The lack of current unsteady friction models reduces the number of available choices. Other current models have either errors in theory or in laboratory work, or are inflexible in their numerical modeling capability.

It is therefore quite natural to select Zielke's time-dependent friction model as a basis for the development of an unsteady friction model, to be applicable in general transient conditions. Unless specified otherwise, the term unsteady friction model ' f_u ' only refers to the time-dependent friction model for the remaining portion of this chapter.

8.2.1.3 Procedures for developing a proposed friction model

Analysis in this chapter is conducted in four parts for the development of the unsteady friction model. These include:

- i) Development of an energy relation for an unsteady friction system (Sec. 8.2);
- ii) Investigation of transient phenomena with the presence of time-dependent friction (Sec. 8.3);
- iii) Evaluation of the relevance of selected unsteady friction model on the maximum transient energy for hypothetical flows (Sec. 8.4); and
- iv) Application of the selected unsteady friction model to common surge control devices (Sec. 8.5).

8.2.1.4 Problems associated with unsteady friction theory

Some problems associated the current unsteady friction theories have been minimized, as discussed below:

1) Elastic energy --- With the adoption of the Equation of State, the numerical model simulation basically prevents the potential error of omitting the elastic behavior in a pipe transient flow, as was done in laboratory work associated with the development of two momentum friction theories (first and second derivative of flow change).

11) Unaccounted for decay in transient energy --- On the other hand, the availability of physical modeling data supplements the missing information of transient energy decayed but unaccounted for by the numerical model.

A question now arises as whether this unaccounted for decay of transient energy during the transient period is due to the mechanical transfer mechanism related to unsteadiness of friction, or due to the energy loss associated with the disturbance of the wall shear stress. This question is discussed below.

8.2.2 Energy in flow with periodic fluctuation

Due to the surge wave reflection, periodic fluctuations of flow appear during and after the flow closure period, throughout the entire pipe system (except at the source).

8.2.2.1 Characteristics of periodic fluctuation

In addition to the unsteady nature (as that appears in a mass oscillation) of flow fluctuations in general transient conditions also possess the following properties:

1) Time-specific property (T) ---

i) Pipe system --- Transient flow velocity changes primarily at the instant when the surge wave arrives. After this its change is not significant for the rest of the transient duration.

ii) U-tube system --- Conversely, flow in a U-tube oscillates in a continuous manner.

2) Location-specific property (X) ---

i) Pipe system --- Similarly, a high rate of transient flow velocity change can be found where the surge wave reflects, while other parts of the pipe system are basically uniform.

ii) U-tube system --- As discussed previously the mass in a U-tube moves as a rigid body, independent of location.

3) Abrupt change property ---

i) Pipe system --- The change of transient flow velocity is step-wise, but in the same direction as the source flow during the flow closing period. However, the periodic flow movement moves in an opposite direction after the flow closure.

ii) U-tube system --- By comparison, the change of flow in U-tube is gradual.

4) Rapid movement property ---

i) Pipe system --- The period of surge fluctuation is $t_s = 4 l/a$, which is in seconds in general practice.

ii) U-tube systems --- The period of flow fluctuation is $t_s' = 4 * l/v$, which is two orders of magnitude longer than that of the pipe system.

5) Discussion of fluctuation characteristics

The distinct properties possessed by the periodic flow fluctuations of the two systems suggest that the conventional treatment of unsteady friction phenomenon

developed from a mass oscillation may not be appropriate for a transient pipe flow.

8.2.2.2 Energy losses in a transient flow

It is well known that energy losses exist whenever and wherever a flow movement occurs. In a transient pipe flow, two components of energy losses exist:

1) Energy loss related to a friction factor (ED_f) ---

For a relative slow flow variation the parallel flow streamlines are not disturbed and the energy loss, commonly termed friction dissipation (ED_f), is primarily related to the friction factor (f_d).

Previous analyses (Ch. 6.0 and 7.0) already showed that the friction dissipation (ED_f) in transient flow (for $T < 100$) is insignificant. Instead, the existence of friction affects transient behavior in a form of (mechanical) friction transfer (ET_f).

2) Energy loss related to non-friction factor (ED_{nf}) ---

Upon a rapid flow velocity change in periodic fluctuations, the parallel flow streamlines would be disturbed. Several schools of thought have been brought up to tackle this loss. These include:

i) Flow separation --- Separation has been observed during the change of flow direction and magnitude (Zielke 1966, page 15). Energy losses much higher than the friction dissipation are expected.

ii) Boundary layer --- A large loss of energy in the boundary layer during the reversal of flow has been proposed, like a sock being pulled inside out (private correspondence from D. Stephenson, Dec. 5, 1984).

Consequently, in addition to the friction dissipation (ED_f), an energy loss unrelated to the friction factor (f_a) should be introduced. For clarity, this component of the energy loss related to the periodic flow fluctuation is here termed the non-friction dissipation (ED_n).

It should be noted that this loss may still be affected by the viscosity of the fluid. However, the name 'non-friction dissipation, (ED_n)' is kept in this thesis in order to keep it distinct from other energy forms, such as non-friction transfer (ET_n), friction transfer (ET_f) and friction dissipation (ED_f).

3) Discussion of energy loss in a transient flow

Previous analyses (Ch. 5.0 to 7.0) attribute all the effects of periodic fluctuations in a form of mechanical energy transfer (ET_n , ET_f and ET_{nf}). In this chapter, the effect of periodic fluctuations in the form of non-friction dissipation (ED_n) is the focus.

8.2.2.3 Relevance of non-friction dissipation

The relevance of non-friction dissipation are assessed by comparing two different transient flow systems, i.e. uni-directional vs. bi-directional flow. Figures of transient flow history previously plotted in early part of each chapter (Ch. 5.0 to 7.0) can be referred to the discussions below.

1) Direction of flow with periodic fluctuations ---

i) Flow closing period --- Generally, the flow changes in a gradual and step-wise alternation with the same direction of steady flow movement.

i) Decaying period --- The flow now abruptly reverses its direction at the instant the surge wave passes.

2) Magnitude of periodicity ---

i) Flow closing period --- The rate of flow change associated with periodic fluctuations is relatively small.

ii) Decaying period --- Periodic fluctuations continue after the flow closing period, with same period (T_a), but with much larger magnitude. This is apparent since the energy flux reversal is twice the sum of outflux accumulated during the whole flow closing period.

3) Time of transient (T) ---

i) Flow closing period --- During the flow closing period, both elastic energy (E_m) and non-friction dissipation (ED_m) are increasing with time. However, the former (due to the imbalance of flow energy) should be far greater than the latter (due to the heat exchange). This implies that:

$$E_m(T) \gg ED_m(T) \quad 8.11a$$

where $0 < T < T_a$.

Therefore, the importance of non-friction dissipation (ED_m) decreases during the flow closing period.

ii) Decaying period --- After the flow closure, the non-friction dissipation (ED_m) continues to increase, while the peak decaying energy (E_r) does the opposite.

$$\begin{cases} dE_r(T)/dT < 0 \\ \{ \\ dED_m(T)/dT > 0 \end{cases} \quad 8.11b$$

where $T > T_a$.

Hence, the importance of non-frictional dissipation (ED_m) affecting the transient decay increases with time.

4) Discussion of the relevance of non-frictional dissipation

From the above analysis it is reasonable to conclude that the non-frictional dissipation (ED_m) is insignificant in a uni-directional flow compared to the elastic energy (E_m).

Conversely, further analysis should be conducted on its effect in a bi-directional flow, where considerable non-frictional dissipation (ED_m) may be accumulated during the decaying period.

8.2.3 Transients in an unsteady friction system

8.2.3.1 Energy balance equation

In the neglect of friction dissipation (i.e. $ED_f = 0$), the transient energy can be expressed as:

$$\begin{aligned}
 E_m(T_1) &= E(T_1) - E_w(T_1) \\
 &= E(T_1) - [E_{wE}(T_1) - E_{wS}(T_1)] \\
 &= E(T_1) - \left[\int_0^{T_1} Q_E(T) dT - \int_0^{T_1} Q_S(T) dT \right]
 \end{aligned}
 \tag{8.12}$$

where the transient energy consists of two components:

- i) mechanical energy transfer, $E_w(T)$; and
- ii) internal energy decrease from energy loss, $E(T_1)$.

8.2.3.2 Unsteady friction transfer (ET_{Qf})

Current unsteady friction theories predict transient energy by adopting an unsteady friction factor (f_u). Under such

conditions, the friction dissipation $ED_z(f_u)$, though it may be increased several fold from $ED_z(f_a)$, are still far too small to be significant in a transient pipe flow. Instead, the unsteady friction factor (f_u) governs the transient phenomenon in the form of mechanical energy exchange, which is termed unsteady friction transfer (ET_{uq}).

The energy relation during the transient period would become:

$$E_m(T_1) = E_z - [E_{wz}(T_1) - E_{ws}(T_1)] \quad 8.13$$

$$= E_z - \left[\int_0^{T_1} Q_z(T) dT - \int_0^{T_1} Q_s(T) dT \right]$$

where the internal energy, $E(T) = E_z$, is constant during the transient period.

Here, the unsteady friction transfer is defined as:

$$ET_{uq}(T) = ET_z(f_q, T) - ET_z(f_u, T) \quad 8.14$$

where it does not appear directly in Eq. 8.13 because its existence has been reflected by a component of the mechanical energy process.

This unsteady friction transfer (ET_{uq}) can be further subdivided as:

i) unsteady surge transfer (ET_{uqs}), for uni-directional flow ($T < T_{mx1}$); and

ii) unsteady decay transfer (ET_{uqd}), for bi-directional flow ($T > T_{mx1}$).

8.2.3.3 Non-friction dissipation (ED_m)

Should the non-friction dissipation (ED_m) dominate the transient phenomena in an unsteady friction system the energy balance equation (Eq. 8.12) becomes:

$$E_m(T_1) = [E_z - ED_m(T)] - [E_{wm}(T_1) - E_{ws}(T_1)] \quad 8.15$$

$$= [E_z - ED_m(T)] - \left[\int_0^{T_1} Q_m(T) dT - \int_0^{T_1} Q_s(T) dT \right]$$

where the existence of non-frictional dissipation (ED_m) affects the pipe elastic energy (E_m) by reducing the internal energy $[E(T) = E_z - ED_m(T)]$ rather than by the mechanical transfer of energy.

8.2.3.4 Discussion of transient energy

Analysis of the energy relation for an unsteady friction system suggested that the transient energy (E_m) can be predicted by two methods:

- i) An indirect method by means of mechanical energy transfer (ET_{uq}); or
- ii) a direct method by the adoption of non-friction dissipation (ED_z).

Further analysis in the time-dependent friction model is undertaken in the next section, taking the unsteady friction transfer (ET_{uq}) into consideration.

8.3 Analysis of Time-dependent Friction Model

Previous analyses in quasi-steady friction systems indicated that the shorter the time of flow closure (T_q) the greater the effect of a quasi-steady friction factor on transient energy decay (ET_{qsd}). Furthermore, this effect cannot be easily observed with a short period of simulation time (T).

In the present analysis only an instantaneous flow closure ($T_d=0$) is simulated for various friction head and regime conditions, with the decaying period extended to $T = 101$.

8.3.1 Re-simulation with time-dependent model

Examples from Holmboe (1964) and Sauer (1969) are re-simulated in this research. The first case is for laminar flow (Fig. 8.6) and the second is for turbulent flows (Fig. 8.4).

Three transient properties have been analyzed and presented in Figs. 8.4 and 8.6. These include:

i) transient flow at mid-pipe $Q(T, X=0.5)$ as shown in part (b);

ii) friction factor at mid-pipe $f_w(T, X=0.5)$ as shown in part (c); and

iii) transient pressure at the source $H_m(T, X=0)$ as shown in part (d).

It should be noted that only point (iii) was investigated by Zielke (1966) during the development of his time-dependent friction theory.

In addition to the time-dependent friction theory two traditional friction models (constant and quasi-steady) are also simulated for comparison purposes.

As shown in Figs. 8.4 and 8.6, with the adoption of a time-dependent friction model (f_w), the decay of transient properties increases substantially in comparison to other friction models. Greater damping impacts in this time-dependent friction model can be explained from the hydraulic point of view as discussed below.

8.3.1.1 Transient flow during decaying period

Prior to an exploration into the relationship between the decay of transient energy and the various governing parameters, unsteady flow conditions will be discussed in terms of the time (T) and the location (X) along the system.

1) High flow --- As illustrated in part (b) of Figs. 8.4 and 8.6, the rate of transient flow at mid-pipe fluctuates with a period of $T_s=4$. Within this period, a positive outflow ($+Q$) and a negative reversed flow ($-Q$) appear intermittently. For clarity of discussion, these two flows are termed 'high flows' to distinguish them from other flow conditions.

At the mid-point of the pipeline the durations of both high flow events have the same one-wave travel time. For example, $1.5 < T < 2.5$ for the high outflow ($+Q$), and $3.5 < T < 4.5$ for the high reverse flow ($-Q$), as shown in Figs. 8.4b and 8.6b.

It should be noted that the duration of these high flows ($+Q$ or $-Q$) varies from a complete time duration (e.g. $0 < T < 2$, at the pipe-end) to just an instant moment (e.g. $1^- < T < 1^+$, at the source).

ii) Between the two high flow periods there are times when 'low flows' exist. For example, these low flows at the mid-point occur at $0.5 < T < 1.5$, $2.5 < T < 3.5$ and so on.

In both examples (Figs. 8.4b and 8.6b), the low flow rate is almost zero.

The peak rates of flow (both $+Q$ and $-Q$) gradually decrease with time, while the rate of low flow does the opposite. At the time when the transient energy is damped out the differences between the high and low flows have been virtually eliminated.

8.3.1.2 Comparison of friction model applications

The decay of transient energy is governed by the flow velocity and the friction factor. The inter-relationship between these two parameters is analyzed for the quasi-steady and time-dependent friction models. This is shown between part (b) and part (c) of the two figures.

1) Quasi-steady friction model ---

i) High flow period --- It can be observed that the quasi-steady friction factor is only slightly greater than its steady state value during the high flow conditions (e.g. $f_q/f = 1$, at $1.5 < T < 2.5$).

ii) Low flow period --- However, it reaches a very high order of magnitude during the low flow period (i.e. $f_q/f \gg 1$, at $0.5 < T < 1.5$).

2) Time-dependent friction model ---

i) High flow period. --- A significantly higher time-dependent friction factor can be observed (in which $f_w/f_q \gg 1$, at $1.5 < T < 2.5$) during the high flow period.

ii) Low flow period --- Conversely, the reverse situation occurs during the low flow period (in this case, $f_w/f_q \ll 1$, at $0.5 < T < 1.5$).

3) Comparison of friction models ---

i) Low flow period --- It should be noted that the effect of friction on transient energy is still insignificant during the low flow period. The difference of friction factor between the two models is negligible.

ii) High flow period --- On the other hand, an increase in the friction factor during the high flow period may become significant, affecting the transient energy.

4) Surge wave reflection during high flow period ---

i) Transient flow --- During the high but relatively constant flow duration [e.g. $1.5 < T < 2.5$ at mid-pipe, part (b) of Figs. 8.4, and 8.6], the surge wave travel has not crossed the mid-pipe during the whole period of $1.5 < T < 2.5$.

ii) Friction factor --- As shown in part (c) of Figs. 8.4 and 8.6, the unsteady friction factor has been significantly increased ($f_u/f_a \gg 1$) and it continues to vary during the whole period.

A rapid change in the friction factor ratio (f_u/f_a) also implies a rapid distortion of the shear stress profile, which contradicts the actual condition of non-wave reflection within each of the high flow period.

5) Discussion of friction model applications ---

The time-dependent friction model (f_u) predicts a much greater damping impact on transient energy during the high flow period than that predicted by the other two friction models (i.e. constant and quasi-steady) during the decay of transient energy.

However, the physical phenomenon of the unsteady friction factor (f_u) does not reflect the characteristics of wave reflection.

8.3.2 Parameters governing unsteady friction transfer

Relations between unsteady decay transfer (ET_{uqd}) and its governing parameters during the decaying period can be expressed as:

$$ET_{uqD}(T) = F(R, H_z, T) \quad 8.16$$

where $T_q = 0$ and $S = \infty$ for an instantaneous flow closure.

The unsteady friction transfer $ET_{uqD}(T)$ at $T = 101$ is shown in Fig. 8.8 summarized in Table 8.1. Results are discussed in this subsection.

8.3.2.1 Time of transient (T) as a relevant parameter

1) Transients in the model simulation ---

As shown in Fig. 8.8, the unsteady decay transfer (ET_{uqD}) increases with time (T).

i) Periodic fluctuation --- The magnitude of the periodic fluctuating flow decreases with time in a frictional system, as shown in part (b) of Figs. 8.4 and 8.6.

ii) Friction factor --- On the other hand, the increase of friction factor (f_u) with time is also observed in the time-dependent friction model, as shown in part (c) of Figs. 8.4 and 8.6.

2) Effect of flow velocity on transients ---

i) Non-friction dissipation --- In pipe flow hydraulics, the non-friction dissipation (ED_z) is somewhat related to the change of flow velocity at the moment of flow reversal [i.e. $dV/dT = O(V^0)$].

ii) Unsteady Decay transfer --- The mechanism of unsteady decay transfer (ET_{uqD}) is related to the existence of friction. It is related to $dV^2/dT = O(V^2)$ in a turbulent flow and to $dV^2/dT = O(V^1)$ in a laminar flow.

Therefore, upon the decay of flow velocity ($dV/dT < 0$), the unsteady decay transfer, $ET_{uqD}(T)$ would vary much more

slowly than that of the non-frictional dissipation, $ED_{\infty}(T)$, for the quasi-steady friction factor, $f_{\infty}(T)$.

3) Discussion of time parameter

Should the unsteady decay transfer (ET_{∞}) in the time-dependent friction model actually account for the friction dissipation (ED_{∞}), an increase in the hypothetical unsteady friction factor (f_u) with time is necessary.

This is exactly what is happening in the present model simulations.

8.3.2.2 Friction (H_f) as a relevant parameter

As shown in Table 8.1 and Fig. 8.8, the magnitude of unsteady decay transfer (ET_{∞}) is decreasing rapidly with increasing friction head (H_f).

1) Unsteady decay transfer in model simulations ---

i) Low friction system --- Unsteady decay transfer is very high in low friction head.

ii) High friction system --- Conversely, in a frictional system with $H_f > 1$, virtually no difference can be found among the various friction model applications.

2) Friction effect on flow momentum change ---

i) Low frictional system --- The existence of a high reversed flow rate (almost 100 % of the original outflow rate) in an instantaneous flow closure with low friction head suggests a strong potential for a change in the flow momentum (dV/dT) to be induced by the periodic fluctuations.

ii) High frictional system --- The existence of friction would smooth out the periodic fluctuations,

which in turn would reduce the change of flow momentum (dV/dT).

3) Discussion of friction parameter

The above relations between unsteady decay transfer (ET_{uqd}) and friction (H_f) strongly suggest the existence of a non-frictional component, governed by the change of flow momentum (dV/dT).

This component is part of the non-friction dissipation (ED_{Σ}) introduced in this research.

8.3.2.3 Flow regime (R) as a relevant parameter

As shown in Table 8.1 or Fig. 8.8, the unsteady decay transfer (ET_{uqd}) is increasing with the decrease in the flow regime (R).

1) Energy loss in laminar flow ---

The parallel flow streamlines of the viscous zone extend across the whole pipe section in laminar flow. Upon the rapid change of flow movement (dV/dT), the disturbance of flow is felt in the whole body of flow. Consequently, the energy loss greater than presented by the laminar friction theory occurs.

2) Energy loss in turbulent flow ---

The viscous layer is getting thinner with the increase in the Reynolds Number (R). Therefore, for the same magnitude of flow momentum change in terms of (dV/dT), an energy loss smaller than that of the laminar flow is expected.

It should be noted that the effect of a disturbance on the turbulent core flow is much less than the effect on a laminar flow.

3) Discussion of flow regime parameter

In addition to the non-frictional component (e.g. dV/dT), other factors related to the viscous properties of the fluid (e.g. Reynolds Number, R) may also be relevant.

The energy loss induced by the distortion of shear stress is an evidence that the decay of transient energy is due to the internal energy loss, represented by the non-friction dissipation (ED_{nf}).

Current unsteady friction theories, including the present time-dependent friction model, actually account for the mechanical transfer mechanism (ET_{unf}) by the importance and effect of the friction factor (f_u).

8.3.2.4 Location (X) as a relevant parameter

1) Transients during model simulation ---

The parameter (X) is not included in the Eq. 8.16, because the unsteady decay transfer (ET_{unf}) has integrated the effect of unsteady friction along the whole system. However, analysis of the time-dependent friction model simulation shows that upon an instantaneous flow closure ($T_{\text{cl}} = 0$) with a low friction head ($H_f < 0.1$) that:

- i) The duration of the time-dependent friction factor (f_u) with high magnitude is diminishing towards the pipe-end.
- ii) In the meantime, the duration of the high flow ($+Q$ or $-Q$) increases towards the pipe end.

The local effect of unsteady friction, which is the combination of both the unsteady friction factor, $f_u(X)$ and the flow $Q(X)$ therefore possesses the potential for homogeneity along the whole pipeline.

2) Characteristics of non-friction transfer ---

Under the above hydraulic conditions (i.e. $T_q=0$ and $H_z < 0.1$) the same order of flow momentum change (i.e. $dV(X)/dT = \text{constant}$) may be found at any location along the pipeline at the time the surge wave travels. Consequently the non-spatial characteristics also appear as a non-friction dissipation (i.e. $ED_z(X) = \text{constant}$).

3) Discussion of location parameter ---

The above potential uniformity suggests that the additional decay of transient energy is caused by the non-frictional dissipation (ED_z) upon reversal of the flow.

8.3.3 Theory from oscillating flow

The time dependent-friction model adopted for transient conditions was originally developed from the U-tube mass oscillation (Ch. 4, Zielke 1966). With the aid of the hypothetical pipe set-up (Fig. A.8), a time-dependent friction model is applied in this research to test mass oscillating flows. The test data is extracted from examples provided by various authors and researchers, including Safwat and Polder's laboratory work (1973), the previously mentioned example in Streeter and Wylie (1981, page 485) and Zielke's original illustrations (which came from Uchida 1956 and Fan 1958).

No difference in transient simulations can be found by the use of a time-dependent model when comparing with the quasi-steady model. This finding is reasonable because there is virtually no non-friction dissipation (ED_z) existing in the U-tube mass movement, in which the periodic surge fluctuation (with the period T_s) is non-existent and the flow momentum change during the mass oscillation is slow and gradual compared to the surge wave reflection.

Therefore, it can be concluded that the energy relation cannot be linked together for U-tube and pipe systems, as

claimed by current unsteady friction theories (Zielke 1966, Shur and Apelt 1983).

8.3.4. Unsteady friction in uni-directional flow

In addition to the period of transient decay, a general unsteady friction model should also be applicable to the period of flow closing. Further numerical tests with the aid of the time-dependent friction model were carried out for various normal flow closing conditions. Results of these tests are summarized in Fig. 8.9 for the flow closing period.

8.3.4.1 Variation of unsteady friction factor (f_u)

Two different characteristics of unsteady friction factor have been found during the flow closing period.

i) Early stage of flow closing period --- As shown in Fig. 8.9, during the early flow closing period the friction factor generated from the time-dependent friction model is only slightly smaller than that from the quasi-steady friction model. The deviation between these two models increases with the decrease in flow velocity.

ii) Final stage of flow closing period --- At times toward the end of flow closing period a steep drop in the time-dependent friction factor can be observed.

These characteristics contradict the principle of pipe flow hydraulics, related to the frictional effect.

8.3.4.2 Effect of unsteady friction during flow closure

As shown in Fig. 8.9, sensitivity tests in this research indicate that the deviations between the quasi-steady and time-dependent models do not significantly affect transient behavior during the flow closing period.

i) Early stage of flow closure --- The insignificant increase in the unsteady friction factor (i.e. $f_u - f_q = 0$) results in an insignificant transient effect (i.e. $E_m(f_u) - E_m(f_q) = 0$), even though the flow rate (i.e. $Q \gg 0$) is significant.

ii) Final stage of flow closure --- On the other hand, toward the end of flow closing period the flow velocity is very small during the occurrence of a large change in friction factor (see Fig. 8.9). Consequently, the effect of the unsteady friction is insignificant to the overall surge transients even though a large difference in friction factor exists.

It should be noted that the simulation of this hypothetical deviation of the friction factor (f_u) is necessary for the elimination of the numerical model instability in Zielke's model application.

It is concluded that The unsteady surge transfer (ET_{uqs}) can generally be considered zero. Further pursuit of this unsteady friction theory (f_u), other than the use of steady (f) or quasi-steady (f_q) friction model, is not needed for the analysis of uni-directional flow ($T < T_{mx1}$).

8.3.4.3 Effect of unsteady friction during whole transient

The effect of unsteady friction on general transients consists of three parts:

i) Magnitude of E_{mx1} --- The effect of the above friction factor change on transients is relatively minor. The change in the maximum surge energy (E_{mx1}) due to the existence of unsteady friction factor (f_u) is normally less than 5 %.

ii) Characteristics of periodicity --- Earlier analysis suggested that periodic surge fluctuations exist in uni-directional flow, but with a magnitude much smaller than those in a bi-directional flow. Hence, the effect

of the non-friction dissipation (ED_m) is not expected to be significant.

iii) Magnitude of E_T --- The influence of the diversified time-dependent friction factor (f_u) during the flow closing period terminates shortly after the end of flow closure, usually in less than the period of a surge wave fluctuation (i.e. $T_s = 4$). The peak decaying energy (E_T) is governed by the non-friction dissipation (ED_m) associated with other energy transfer mechanisms.

Though being diversified from the real mechanism, the time-dependent friction can still be used as an approximation of maximum transient energy without inducing significant error.

8.3.5 Discussion of time-dependent friction model

A fundamental difference in theoretical background is concluded from the present analysis of the time-dependent friction model. The physical phenomenon of the time-dependent friction factor (f_u) itself cannot be justified, and hence the model should be rejected in general.

On the other hand, by hypothetically increasing and altering the behavior of the unsteady friction factor (f_u), the time-dependent friction model can be used effectively and accurately to predict the transient energy in an unsteady friction system, if enough physical data is available for the necessary computations. For the time being, the model has been verified by two physical tests, one for laminar and one for turbulent flow conditions.

Furthermore, it has been demonstrated in this research that the general transient properties of flow, pressure and internal energy loss (except the unsteady friction factor) can be appropriately interpreted by the time-dependent friction model (directly or indirectly).

This time-dependent friction model is therefore conditionally accepted for the purposes of predicting transient energy in an unsteady friction system (for both uni-directional and bi-directional flows). This work is described in the next section.

The term 'unsteady friction (factor) system' is retained in this thesis for the sake of consistency with other systems, such as non-friction, steady friction and quasi-steady friction factor systems.

8.4 Maximum Transient Energy in Unsteady Friction System

8.4.1 Maximum surge and counter-surge energy

As previously discussed, the non-friction dissipation (ED_{nf}) is considered insignificant during the flow closure period. Hence, the application of a time-dependent friction model is neither recommended nor required. However, results almost identical to the quasi-steady friction model simulation can be found during the flow closing or early decaying period. The relationship below is therefore a valid assumption:

$$E_{\text{mx1}}(f_u) = E_{\text{mx1}}(f_q) \quad 8.17a$$

Furthermore, the effect of unsteadiness (f_u) in the friction factor should also not be significant for the counter-surge energy, since the time duration during this period is relatively short. Similarly, the above relation of equality can also be approximated without significant error induced. This means that:

$$E_{\text{mx2}}(f_u) = E_{\text{mx2}}(f_q) \quad 8.17b$$

8.4.2 Peak decaying energy (E_{101})

8.4.2.1 Parameters governing decaying energy

The non-friction energy (ED_{nf}) accumulated after the end of flow closure increases with time. This can be seen from the

peak decaying energy at $T = 101$, which is presented in Figs. 8.10 and 8.11. As directed by Eq. 4.7, the relationship between the peak decaying energy and its relevant parameters can be expressed as:

$$E_{101}(f_u) = F(H_z, S, T_d, R) \quad 8.18$$

where the effect of unsteady friction can be determined by comparing the difference in peak decaying energy between a quasi-friction $E_{101}(f_q)$ and a time-dependent $E_{101}(f_u)$ friction model.

8.4.2.2 Criteria for evaluating decaying energy

The peak decaying energy in a laminar flow is presented in Fig. 8.10 for three typical flow closing conditions. This relationship can be expressed:

$$E_{101}(f_u) = F(H_z, T_d) \quad 8.18a$$

where $R < 2000$ and $S = -2, 1$ and 2 .

The effect of unsteady friction on transient decay can be evaluated from the following unsteady decay reduction (ER_{uqd}):

$$ER_{uqd} = 1 - E_{101}(f_u)/E_{101}(f_q) \quad 8.19$$

where ER_{uqd} is based on a relative comparison basis.

Fig. 8.11 locates the hydraulic conditions for achieving the same energy reduction as ER_{uqd} . This can be expressed as:

$$ER_{uqd} = F(H_z, S, T_d, R) \quad 8.20$$

Three isoquants of $ER_{uqd} = 0.25, 0.50$ and 0.75 are plotted in Fig. 8.11.

8.4.2.3 Results of model simulation

As shown in Fig. 8.10 the patterns of energy reduction obtained from the time-dependent friction model are similar to those obtained from the quasi-steady friction model (Fig. 7.9). However, the magnitude of the peak decaying energy $E_{101}(f_d)$ is greatly reduced. Fig. 8.11 provides a broader representation of the analytical results.

1) Friction head (H_f) as a relevant parameter (H_f) ---

As shown in Fig. 8.10 the unsteady decay reduction (ER_{uqd}) due to unsteady friction is increasing with friction. It should be noted that this transient property of ER_{uqd} is due to the rapid decrease of the denominator $E_{101}(f_d)$. The unsteady decay transfer (ET_{uqd}) is actually decreasing with friction.

2) Flow regime as a relevant parameter (R) ---

The significant effect of Reynolds Number on the unsteady friction transient behavior can be seen in Fig. 8.11. For example,

i) Laminar flow --- More than 50% of the surge energy can be damped out at $H_f < 0.1$ for laminar flow conditions.

ii) Turbulent flow --- In a typical turbulent flow ($R=10^6$), less than 50 % energy reduction can be achieved even for $H_f = 5$.

3) Time of flow closure as a relevant parameter (T_d) ---

As shown in Fig. 8.11, the effect of unsteady friction is increasing with the decrease in time of flow closure, with an instantaneous flow closure maximizing the effect. The analysis in Fig. 8.11 only covers up to $T_d < 10$.

Normally, for $T_a > 10$, the non-friction dissipation (ED_n) associated with the periodic fluctuations becomes very small.

4) Shape of flow closure as a relevant parameter (S) ---

The effect of unsteady friction on transients is increasing with the shape of flow closure $|S|$. For $|S| > 5$, the transient in normal flow closure behaves in a manner similar to that of an instantaneous flow closure.

8.4.3 Discussion of maximum transient energy

The existence of unsteady friction factor does not affect significantly the maximum surge (E_{mx1}) and counter-surge energy (E_{mx2}):

However, a time-dependent friction model must actually predict transient behavior for both laminar and turbulent flow conditions if the time (T_a) of flow closure is short and the time (T) of decay is long, particularly when the Reynolds Number (R) is small.

8.5 Friction Models for Surge Control Devices

Application of unsteady friction models in pipe systems with surge control devices is discussed in this section.

8.5.1 Current application of unsteady friction model

Application of a current unsteady friction model to a surge tank system has been demonstrated in this research to be unsuccessful (8.1.3.3). In this subsection, two more cases of model applications are extracted from the literature and are discussed below.

8.5.1.1 General conclusion from current practice

Shuy and Apelt (1983) adapted and incorporated various frictional models into the method of characteristics to

analyze hydraulic transients in turbulent piping systems caused by valve closure and pump-trip-out.

No significant deviation is found from the comparison of results simulated by different friction models. As a result of the non-effective unsteadiness of friction on transients, Shuy and Apelt (1983) concluded that:

"A new and better unsteady friction model of general transient problems is lacking and urgently needed."

8.5.1.2 Re-simulations of current work in control devices

In order to have a better understanding of these two authors' view of analysis, the following two cases are numerically reproduced in Fig. 8.12 and 8.13, for a valve and a pump system, respectively.

The equivalent hydraulic conditions for the numerical model simulation are:

Valve: $H_z = 0.17$, $H_o = 0.09$, $S = 1$, $T_a = 2.5$ and $R = 5 \cdot 10^5$

Pump: $H_z = 0.18$, $H_o = 0.40$, $T_m = 0.10$ and $R = 8 \cdot 10^5$

Various friction models used by Shuy and Apelt are:

- i) Steady friction model;
- ii) quasi-steady friction model;
- iii) Time-dependent model with Trikha's version (1975);
- iv) Unsteady mass oscillation model (Hino, 1977); and
- v) Momentum friction model (Carstens and Roller, 1959)

8.5.1.3 Results of analysis related to current works ---

Some of deficiencies in Shuy and Apelt's (1983) work have been found in this research and are summarized below:

1) Improper selection of friction models ---

The unsteady friction theories of Carstens and Roller (1959) and of Hino (1977) were developed under the assumption that the unsteady flow in a pipe acts as a rigid water column while the wall shear stress will vary according to the unsteady flow condition. Investigations of the transient history in these systems show that both valve and pump cases are governed by the elasticity of the system during the flow closing period. Naturally, the rigid water column assumption should never be applied during the decay of transients. In applying these two friction models in the method of characteristics by Shuy and Apelt (1983), the basic assumption made for these two friction theories is violated.

The frequency period of $T_a = 4$ adopted for Hino's theory is not a correct assumption for the transient flow movement in a pipe prior to the complete closure of the valve (or check-valve at the pump). The original work by Hino et al was based on the concept of a U-tube mass oscillation.

As discussed throughout Sec. 8.1, serious constraints exist in the application of current unsteady friction models. Shuy and Apelt's work related to time-dependent friction model is the focus of the discussion below.

2) Limits of Trikha's simplified model ---

Trikha's equation can only be used for $Z > 10^{-4}$ (Trikha 1975, page 102). As shown in Fig. 8.5, both numerical examples (valve and pump) are far below this lower bound (where $Z < 10^{-5}$ in both cases).

Consequently, the malfunction of this time-dependent friction model would be expected in the corresponding transient simulations.

3) Insufficient time of simulation (T) ---

It should be noted that the effect of unsteadiness of friction factor in a turbulent flow system can only be observed for a sufficiently long decay time. For example, at $T = 101$, a clear and significant impact can be seen (Figs. 8.12 and 8.13) for the application of both quasi-steady and time-dependent friction models in both numerical examples.

As indicated in Figs. 8.12 and 8.13, no significant difference is obtained in the present analysis of time-dependent friction, for times of simulation less than 20.

8.5.1.4 Discussion of current control devices

In contrast to the conclusion made by Shuy and Apelt (1983) that a new and accurate unsteady friction theory is needed for transient analysis, the present research suggests that a new and better understanding of transient flow phenomena is required.

The available time-dependent friction models can also be used effectively in common surge control devices if the physical phenomena are properly addressed.

8.5.2 Selected unsteady friction model in practices

As previously discussed, the existence of unsteady friction does not materially affect the transient behavior during the surge (E_{mx1}) or early decaying (e.g. E_{mx2}) period. The present analysis of unsteady friction in a pipe system with a surge control device is focused on the transient energy decay at $T = 101$.

Four common surge control devices are considered, and their general responses to the existence of unsteadiness of friction are summarized in Table 8.2.

8.5.2.1 Unsteady friction model in air chamber or surge tank

1) Two-way operation ---

Transient flows in a two-way air chamber or a two-way surge tank functions in a manner similar to a one-armed U-tube mass oscillation. In these cases, the unsteady friction model is not required because the periodic fluctuations are virtually non-existing.

2) One-way operation ---

If a check valve is installed, a one-way air chamber or one-way surge tank is set-up. In these cases, the time of flow closure is long ($T_a \gg 10$) and consequently the flow velocity of the periodic fluctuations in the pipe line is relatively low. The application of a time-dependent friction model does not appear to be necessary.

8.5.2.2 Unsteady friction model in Pump trip-out

1) Pump with check valve ---

The time for the pump flow to diminish to zero is usually short, of the order of $T_a \ll 10$. The step-wise pump flow closure induces much higher magnitudes in the periodic fluctuations during the decay of transients.

Should a check valve be installed at the pump, the analysis of transient decay can be better approached with a time-dependent friction model for both laminar and turbulent flow conditions.

2) Pump with check valve ---

However, if the flow is allowed to reverse back to the pump, the transient condition will be dominated by the static head (H_a), which is the source of energy to drive the flow reversal. The adoption of a time-dependent friction model becomes unnecessary for this condition.

8.5.2.3 Unsteady friction model in valve closure

For a typical low head valve (e.g. $H_0 = 0.01$), the closure of the valve produces a high shape ($S \gg 1$) of categorized flow closure. In this case, the flow momentum change in periodic fluctuations would be relatively high, in which the decay of transient energy becomes relevant.

Unless the time of valve closure is very long, such as $T_c \gg 10$, the application of time-dependent friction to the analysis of transient decay is desired for both laminar and turbulent flows.

8.5.2.4 Discussion of model in control device systems

The application of time-dependent friction model in pump and valve systems may be desired if the decay of transient energy becomes an issue.

8.6 Conclusions

Analysis in this chapter has resulted in a number of conclusions, related to current unsteady friction models. An unsteady friction model has been adopted which is suitable for general practice.

1) Current unsteady friction model ---

Further to the literature review conducted in Ch. 2, current unsteady friction theories and their applications have been critically examined in this chapter. This is made possible with the presently established:

- i) interpretation tool by means of the Energy Balance Concept; and
- ii) knowledge of transient phenomena, including energy dissipation and mechanical transfer.

The current unsteady friction theories and applications in a pipe transient flow consist of:

i) First and second derivative momentum models, used for uni-directional flow systems;

ii) linear transfer, viscous layer, and alternative time-dependent friction models in bi-directional flow systems; and

iii) surge tank, pump, and valve control devices applied in various unsteady friction models.

It is concluded in this research that the current unsteady friction models and their applications may not be well addressed until now, and may need major revisions. The problems and constraints related to these models include:

ii) Inheritance of unsteady friction theories developed from mass oscillation;

i) adoption of Newton's basic equation of motion for non-rigid flow motion;

iii) neglect of inaccuracy measurements induced by physical modeling instability;

iv) inadequate dimensional analyses and improper scale effects; and

v) constraints in boundary conditions and time of model simulations.

One among these models, Zielke's time-dependent friction model is however suggested for further analysis because of its potential in practical application.

2) Selection of a general unsteady friction model ---

In addition to Zielke's experimental work, transients have been numerically modeled with a longer time duration and wider range of various hydraulic conditions for the evaluation of the time-dependent friction model. The following two conclusions can be made.

1) Theoretical background --- The important role of unsteadiness in the friction factor (f_u) has been vastly exaggerated in time-dependent friction models. This can be explained as follows:

a) By adopting this model, the decay of the transient is hypothetically induced by a mechanism of unsteady decay transfer (ET_{uqd}).

b) The additional decay of transient energy is actually caused by the energy losses (ED_u) due to the existence of periodic fluctuations, but is unrelated to the unsteady friction factor (as expressed in ET_{uqd}).

In current practice, the assumption of $ED_u = ET_{uqd}$ is generally assumed.

11) Application of unsteady friction model --- The time-dependent friction model is selected in this thesis as an unsteady friction model for predicting transient behavior provided that the physical phenomenon of unsteady friction factor (f_u) is treated with caution.

Two conditions for model application are classified:

a) The time-dependent friction model (f_u) is not required during the flow closing and early decaying periods.

b) On the other hand, the model can predict transient behavior with better accuracy at sufficiently long times after the end of flow closure.

In general, the effectiveness of model application increases with the decrease of Reynolds Number (R) and the time (T_d) of flow closure.

3) Application of friction models ---

Based on the analysis of typical events of flow closure and of the maximum transient energy in a wide-range of hydraulic conditions, the following conclusions can be made.

1) Air chamber or surge tank --- The time-dependent friction model is not required for analysis of air chamber or surge tank installation since there is insignificant periodic fluctuations induced during and after the flow closing period.

11) Pump and valve --- Application of a time-dependent friction model for valve closures with short time of closure, and for pump trip-out with check valve, is effective for sufficiently long times of decay.

It should be noted that the decay of transient energy is an important design criteria for the resumption of a normal system operation.

Table 8.1 Summary of Unsteady Decay Transfer

	<u>Unsteady Decay transfer, $ET_{uqd}(T=101)$</u>		
	$R = 2000$	$R = 10^4$	$R = 10^6$
$H_x = 0.01$	0.42	0.28	0.06
$H_x = 0.1$	0.15	0.12	0.03
$H_x = 1$	<0.001	<0.001	<0.001

Table 8.2 Friction Model in Surge Control Device

<u>Control Devices</u>	<u>Effectiveness on energy decay</u>	<u>Reference</u>	<u>Decay at $T=101$</u>	
			<u>ER_{uqd}</u>	<u>ER_{qs}</u>
Surge Tank-two way (turbulent)	Nil	Fig. 8.3	0	1 %
-one way	small	Fig. 8.3	5 %	1 %
Air Chamber-two way (turbulent)	Nil	Any	0	1 %
-one way	small	Any	5 %	1 %
Pump trip -turbulent (with c.v.)	vary	Fig. 8.13	35 %	15 %
-laminar	high	blood vessel	75 %	50 %
(no c.v.) Any	Nil	Fig. 8.13	0	0
Valve -turbulent	vary	Fig. 8.12	25 %	10 %
-laminar	high	machine fuel	50 %	30 %

Note: ER_{uqd} for assessing time-dependent friction model;
 ER_{qs} for assessing quasi-steady friction model.

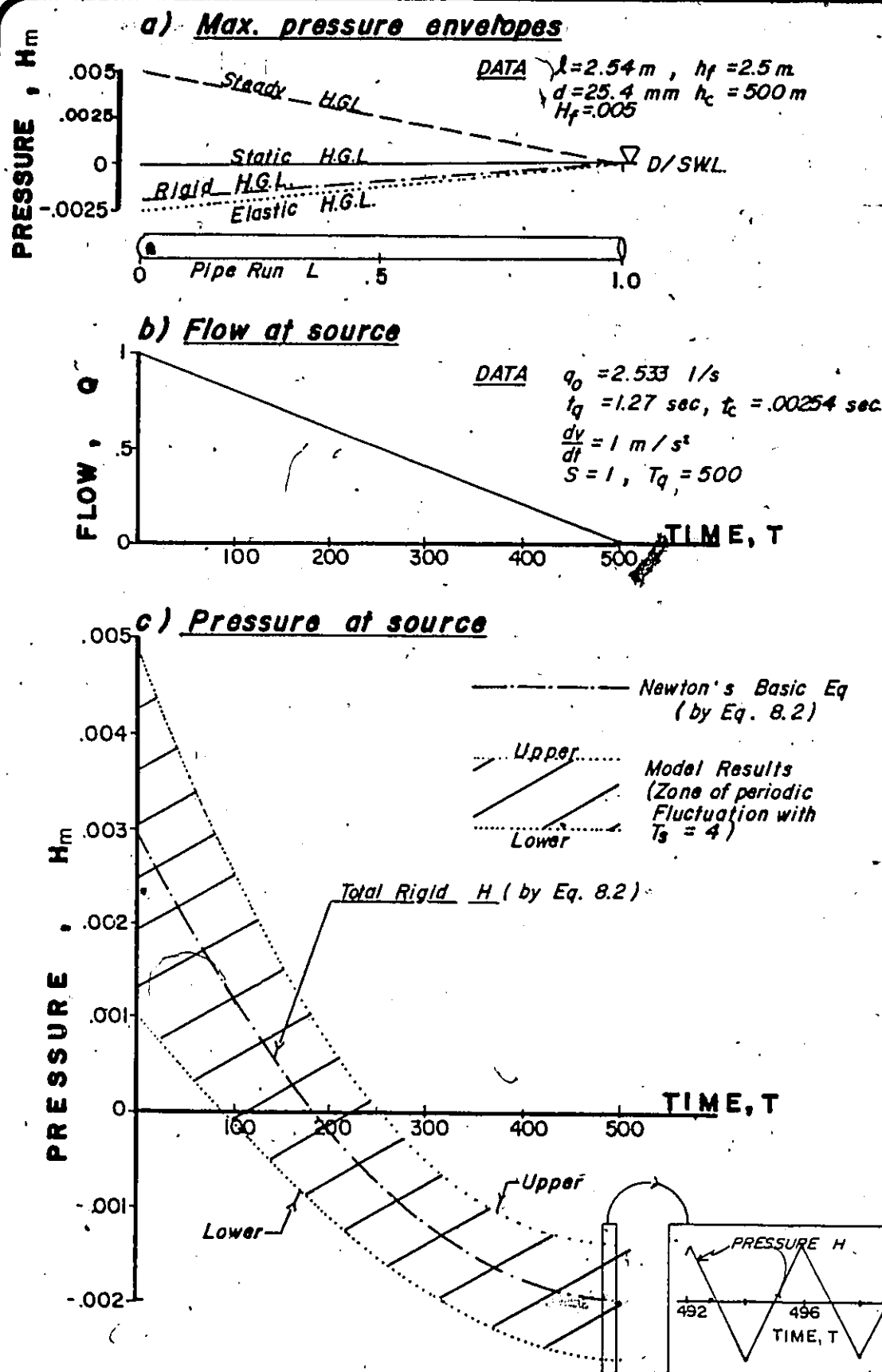


Fig. 8.1 — 1ST ($\frac{dy}{dt}$) Momentum Friction Theory.
 (a example from Daily, 1956)

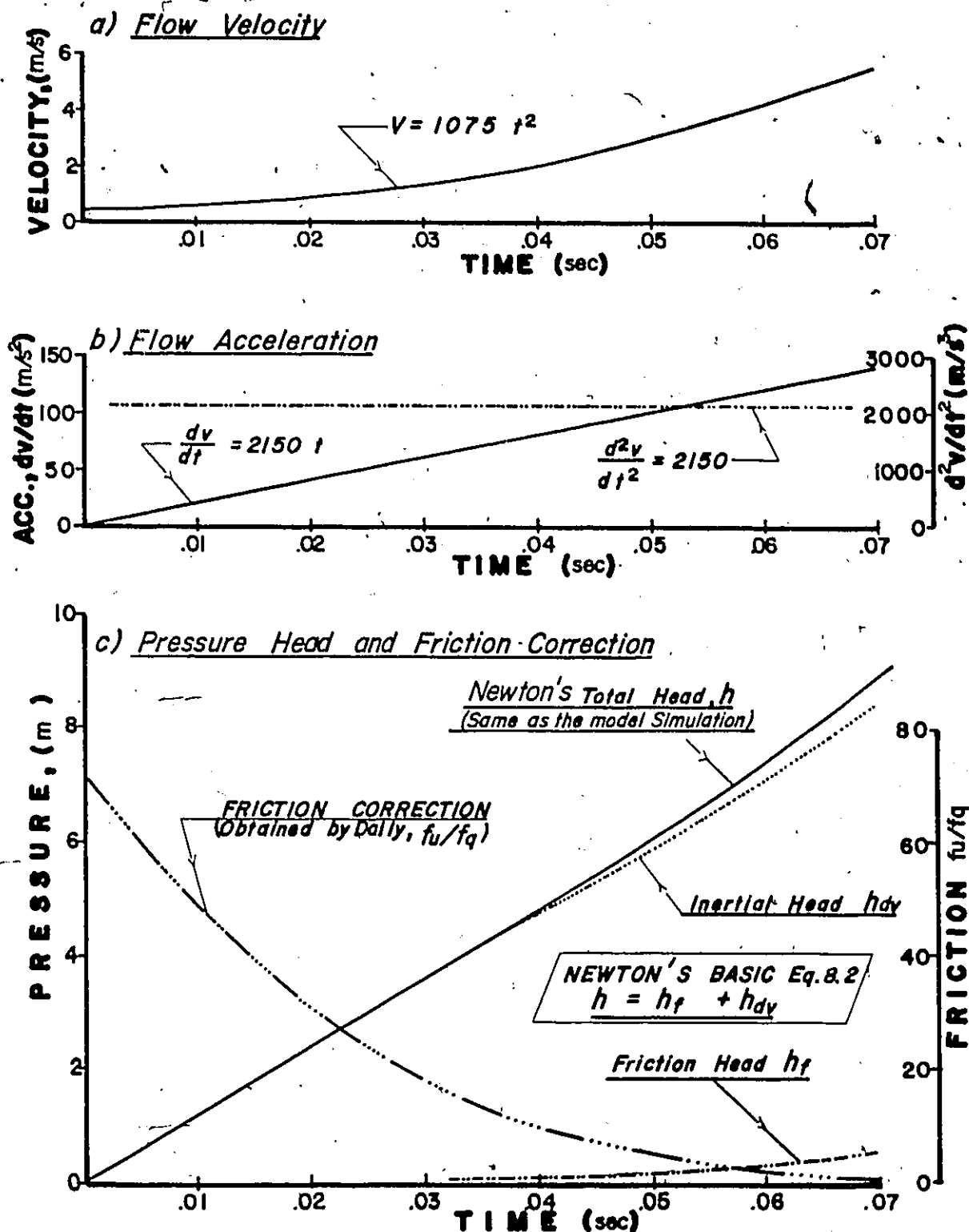
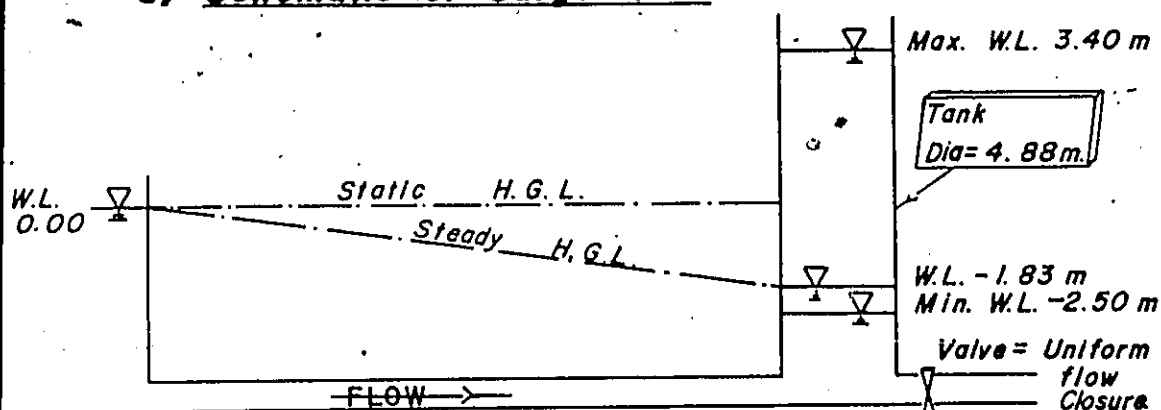


Fig. 8.2 — 2ND (d^2v/dt^2) Momentum Friction Theory.
(a example from Popov, 1974)

a) Schematic of Surge Tank



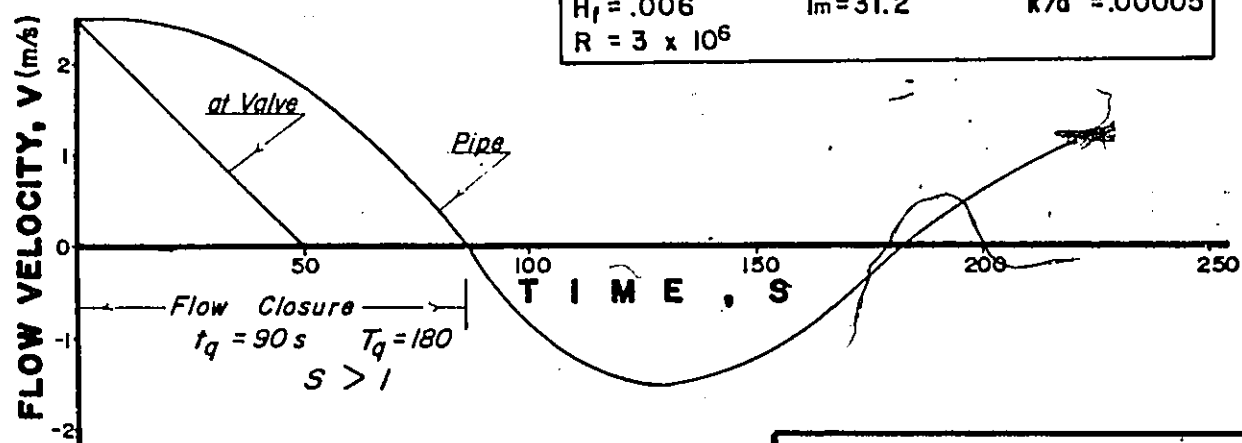
PIPE SYSTEM

$d = 1.22 \text{ m}$	$l = 610 \text{ m}$	$f = .012$
$q_0 = 2.9 \text{ m}^3/\text{s}$	$v_0 = 2.48 \text{ m/s}$	$a = 1220 \text{ m/s}$
$h_f = 1.83 \text{ m}$	$h_c = 303 \text{ m}$	$t_c = .5 \text{ s}$

DIMENSIONLESS

$H_f = .006$	$T_m = 31.2$	$k/d = .00005$
$R = 3 \times 10^6$		

b) Flow in pipe



c) W. L. in Tank

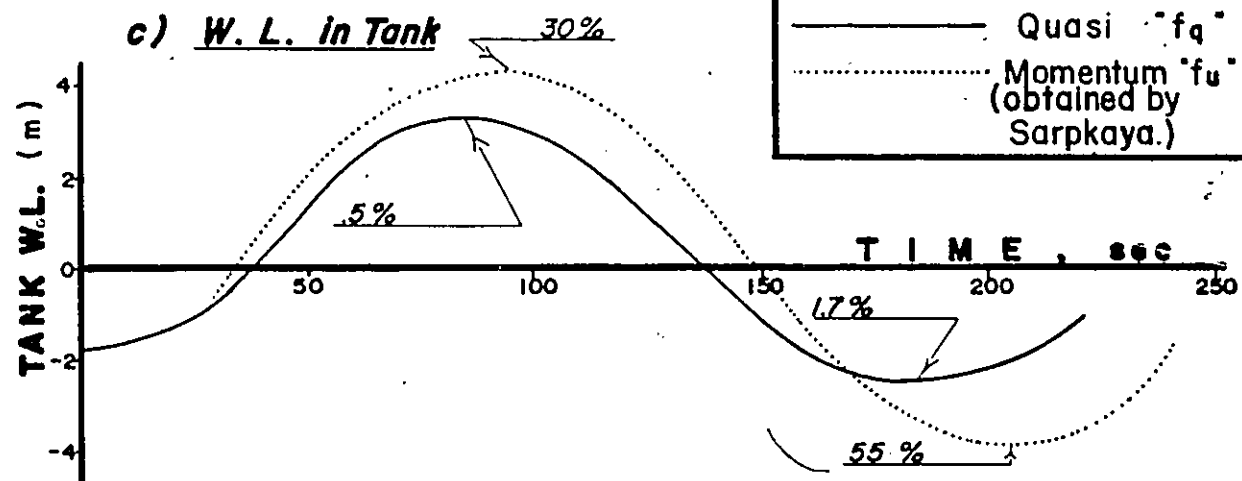
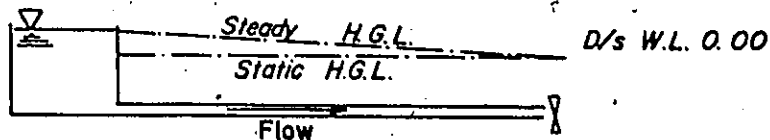


Fig. 8.3 — Application of Unsteady Friction Factor in Surge Tank (an example from Sarpkaya, 1959.).

a) Hydraulic System.



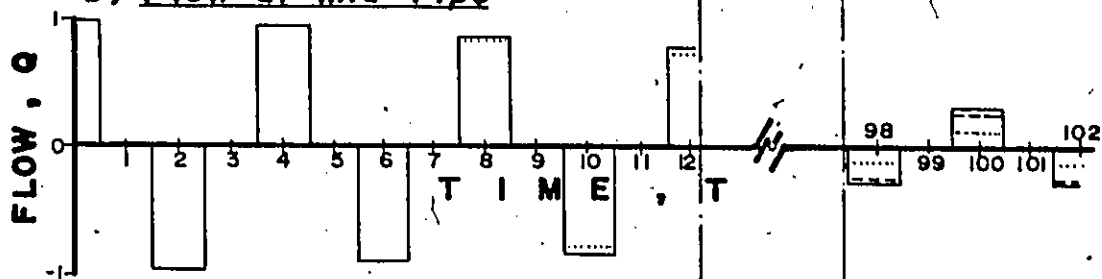
DATA

$l = 32.3 \text{ m.}$ $d = 13.6 \text{ mm.}$ $a = 1357 \text{ m/s.}$
 $q_0 = .43 \text{ l/s.}$ $v_0 = 2.93 \text{ m/s.}$ $f = .0212$
 $h_f = 21.6 \text{ m.}$ $h_c = 398 \text{ m.}$ $\dot{V} = 1 \times 10^{-6} \text{ m}^3/\text{s.}$
 $H_f = .054$ $T_q = 0$ $R = 4 \times 10^4$

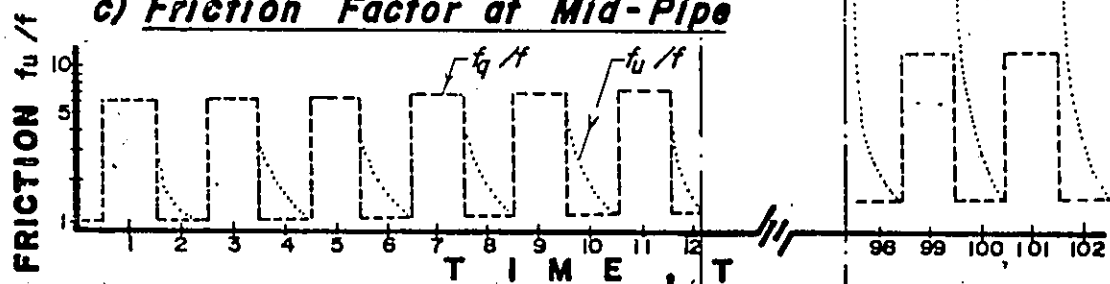
LEGEND

———— Const. " f "
 - - - - - Quasi. " f_q "
 Time-Dep " f_u "

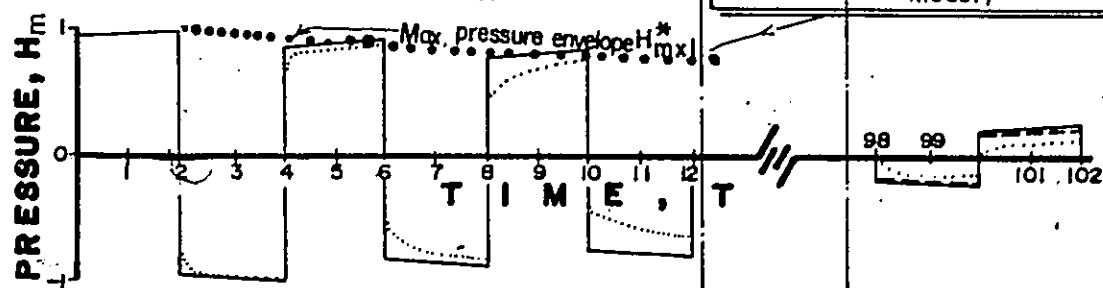
b) Flow at Mid-Pipe



c) Friction Factor at Mid-Pipe



d) Pressure at Valve



Observed maximum Pressure, H_{mxl}^*
 $H_{mxl}^* = H_{mxl} (1.5 f_q, \text{ for viscous model})$
 $H_{mxl}^* = H_{mxl} (\text{time-dependent model})$

Fig. 8.4 — Viscous of Friction Theory (an example from Sauer , 1969.).

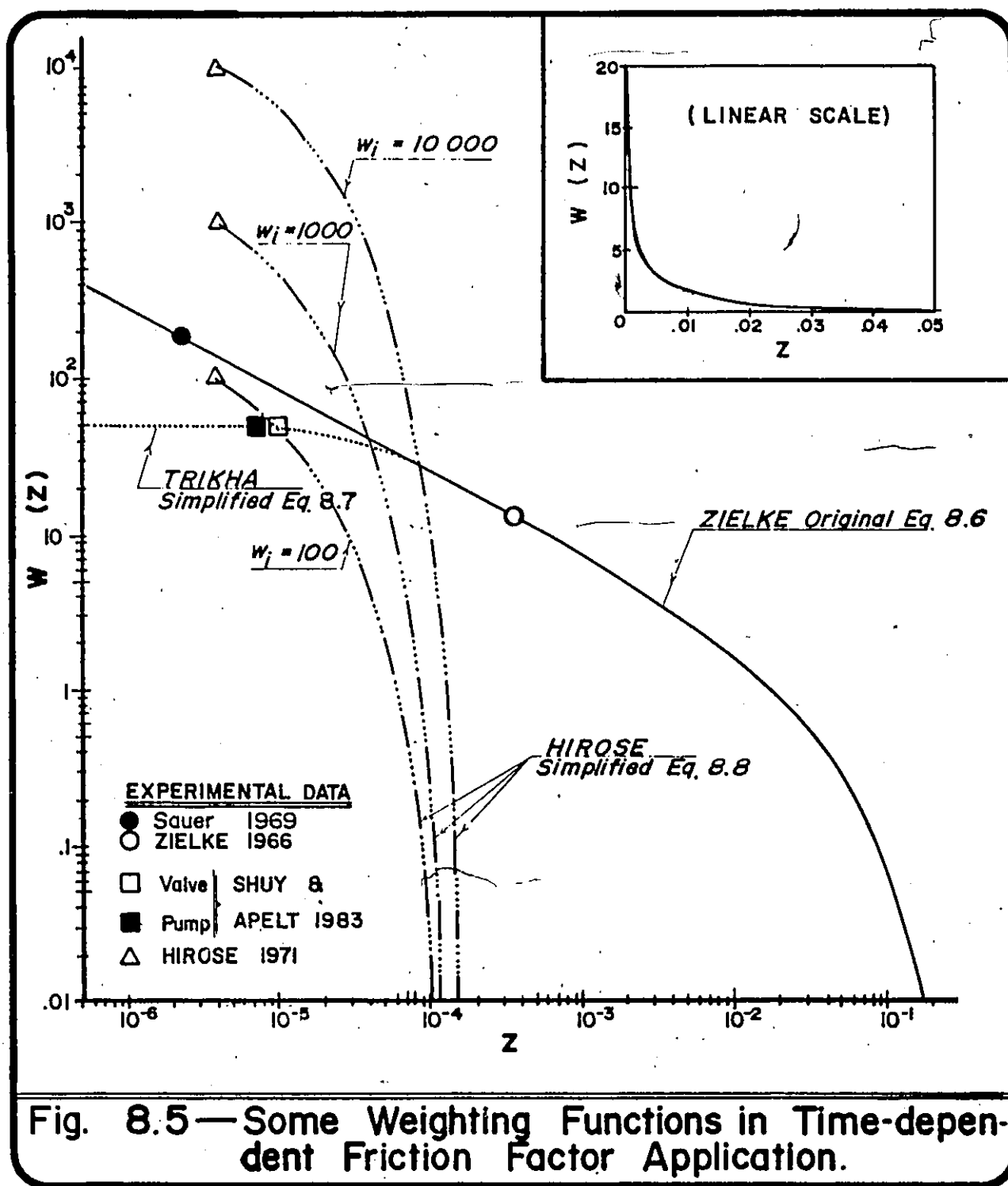
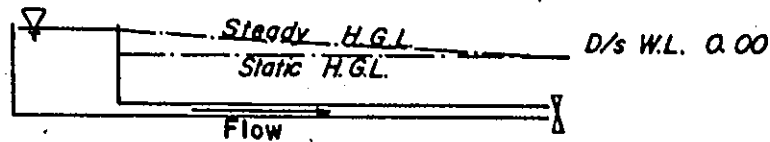


Fig. 8.5—Some Weighting Functions in Time-dependent Friction Factor Application.

a) Hydraulic System.



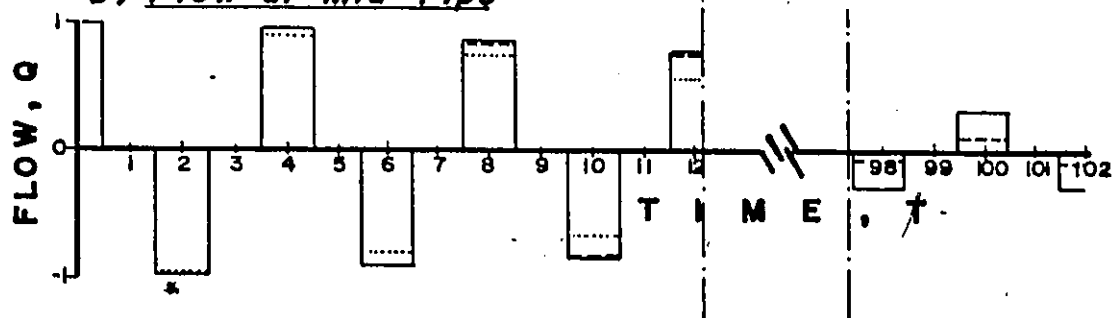
DATA

$l = 36 \text{ m.}$ $d = 25.4 \text{ mm.}$ $a = 1325 \text{ m/s.}$
 $q = .51 \text{ l/s.}$ $v = 1 \text{ m/s.}$ $f = .1$
 $h_f = 7.1 \text{ m.}$ $h_c = 132 \text{ m.}$ $\delta = 39.7 \times 10^{-6} \text{ m}^2/\text{s.}$
 $H_f = .054$ $T_q = 0$ $R = 640$

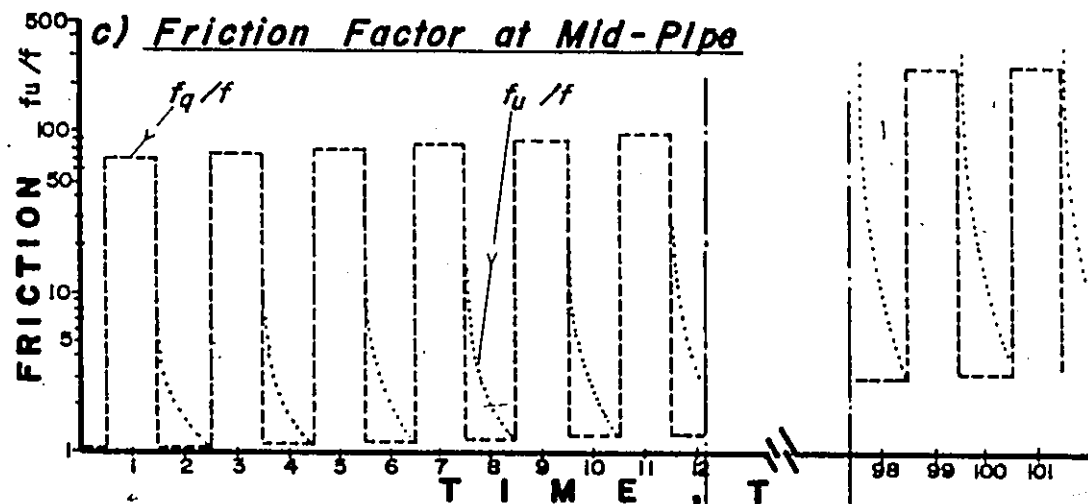
LEGEND

— Const. " f "
 - - - Quasi " f_q "
 Time-Dep " f_u "

b) Flow at Mid-Pipe



c) Friction Factor at Mid-Pipe



d) Pressure at Valve

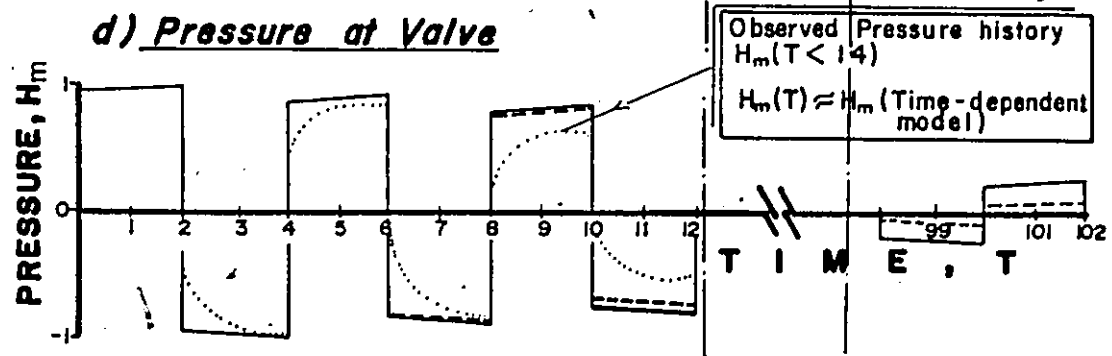
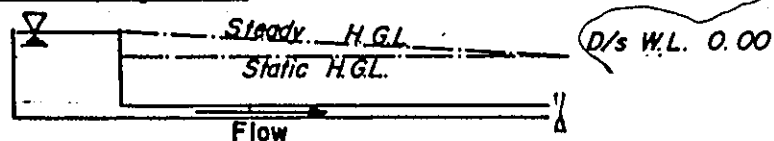


Fig. 8.6—Effects of Time-dependent Friction Factor in Holmboe Works (1964)

a) Hydraulic System.



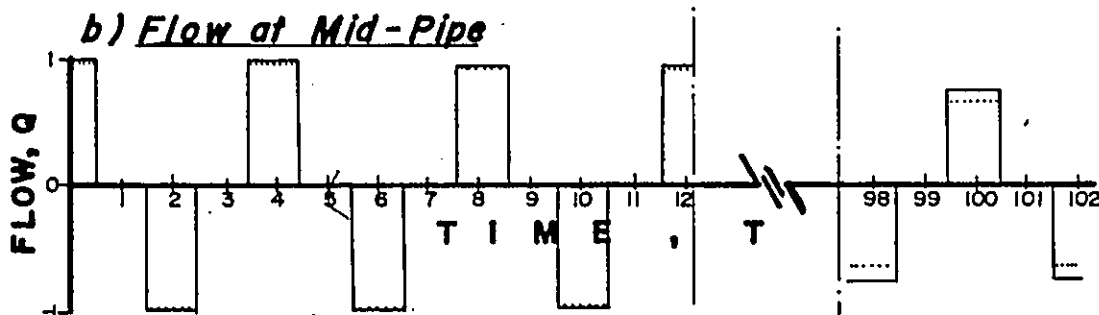
DATA

$l = 915 \text{ m.}$ $d = 305 \text{ mm.}$ $a = 915 \text{ m/s.}$
 $q_0 = 14.6 \text{ l/s.}$ $v_0 = .2 \text{ m/s}$ $f = .02$
 $h_r = .12 \text{ m.}$ $h_c = 18.3 \text{ m.}$ $\nu = 1.13 \times 10^{-6} \text{ m}^2/\text{s.}$
 $H_f = .0065$ $T_q = 0$ $R = 5.4 \times 10^4$

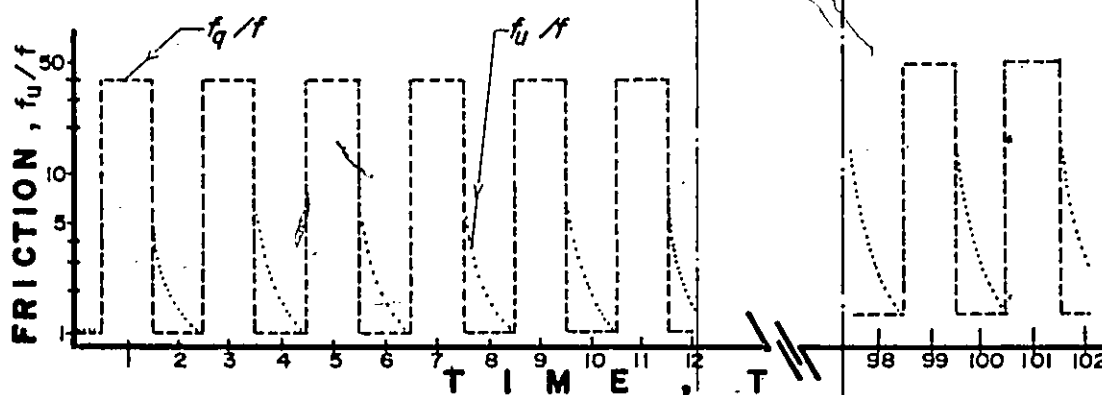
LEGEND

———— Const. " f "
 - - - - - Quasi. " f_q "
 Time-Dep " f_u "
 HIROSE $W_i = 0$ HIROSE $W_i = 7000$

b) Flow at Mid-Pipe



c) Friction Factor at Mid-Pipe



d) Pressure at Valve

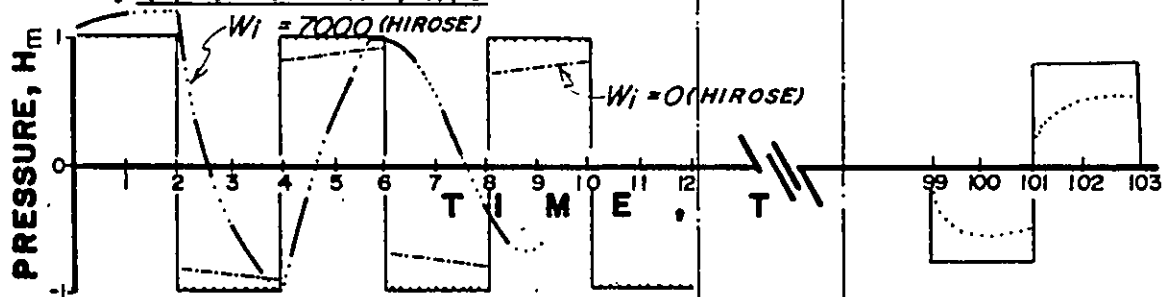


Fig. 8.7 — Evaluation of Alternative Weighting Function (an example from Hirose, 1971)

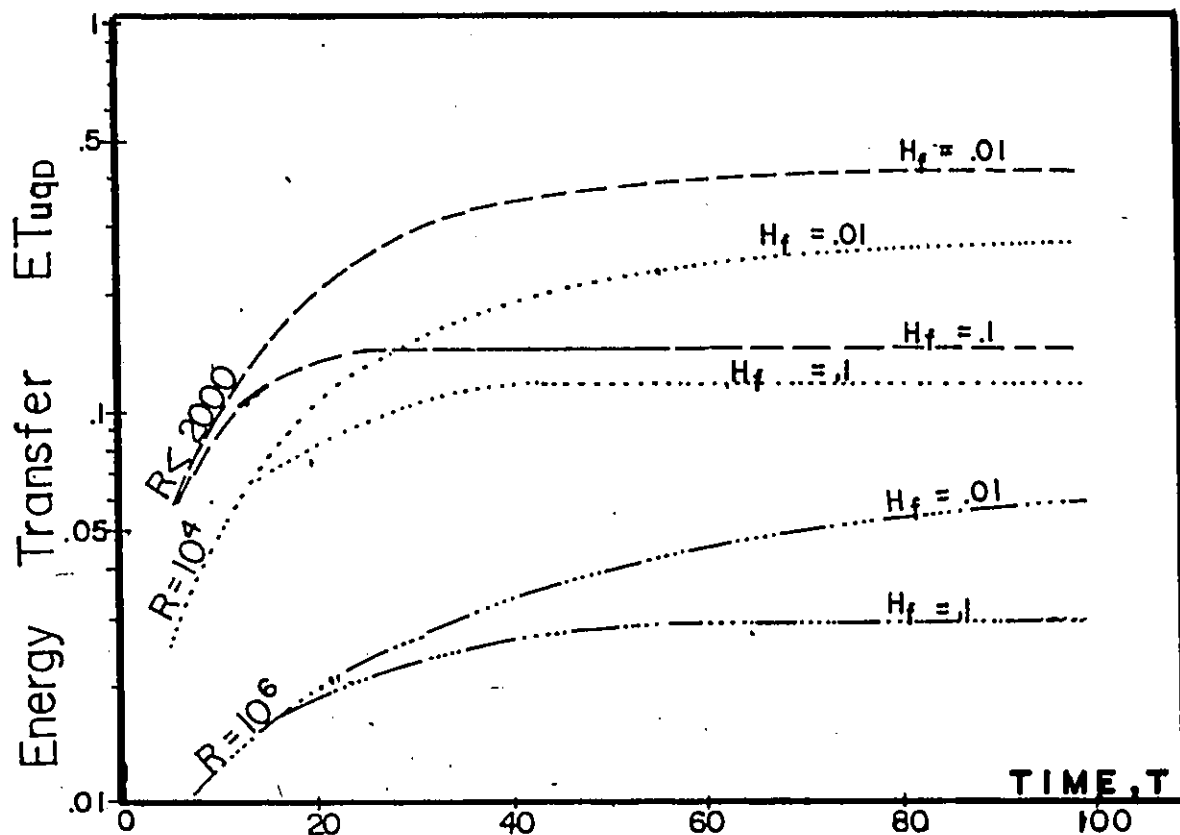


Fig. 8.8 — Effects of Time-dependent Friction Factor on Transients during the Time of Decay.

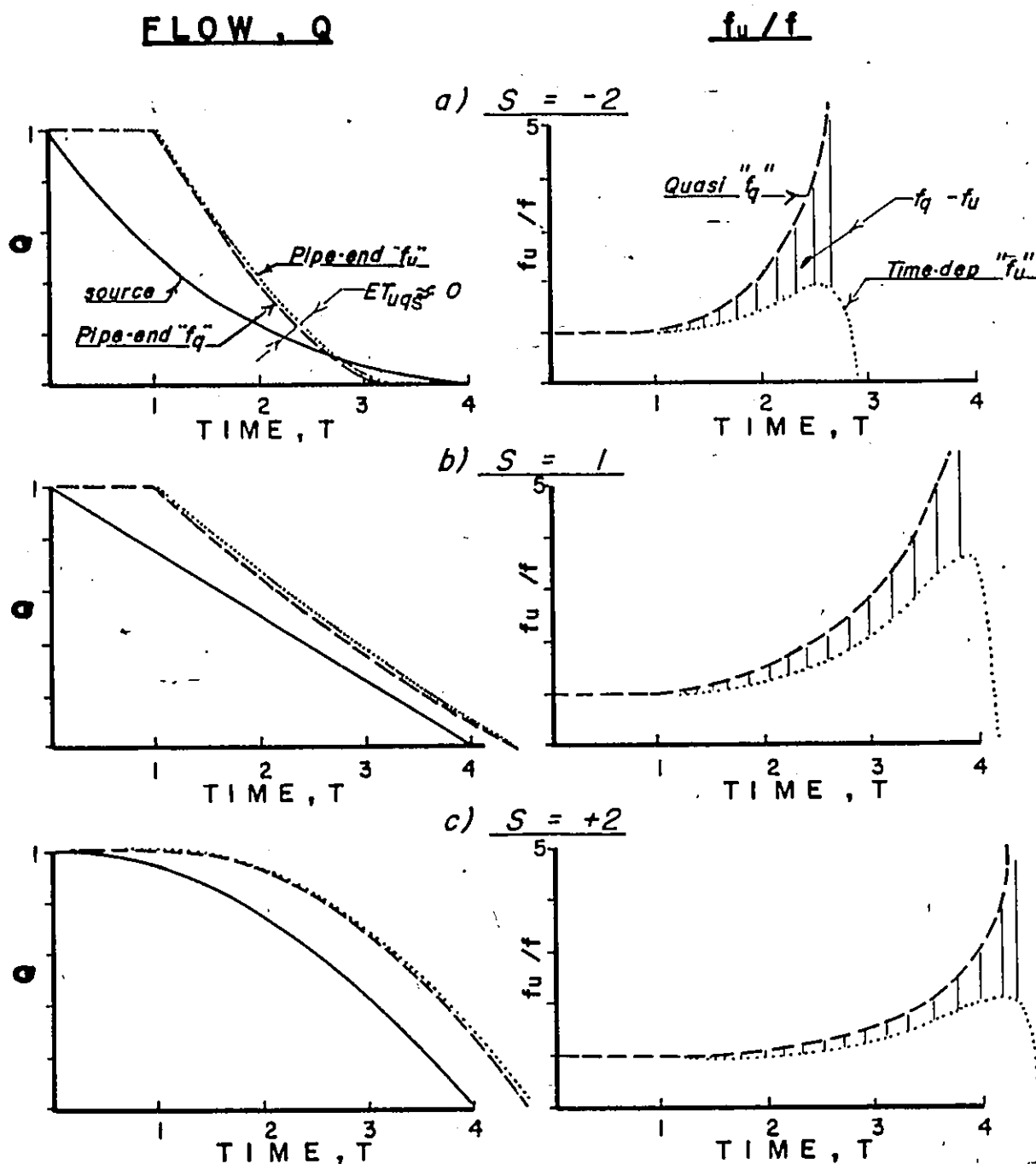
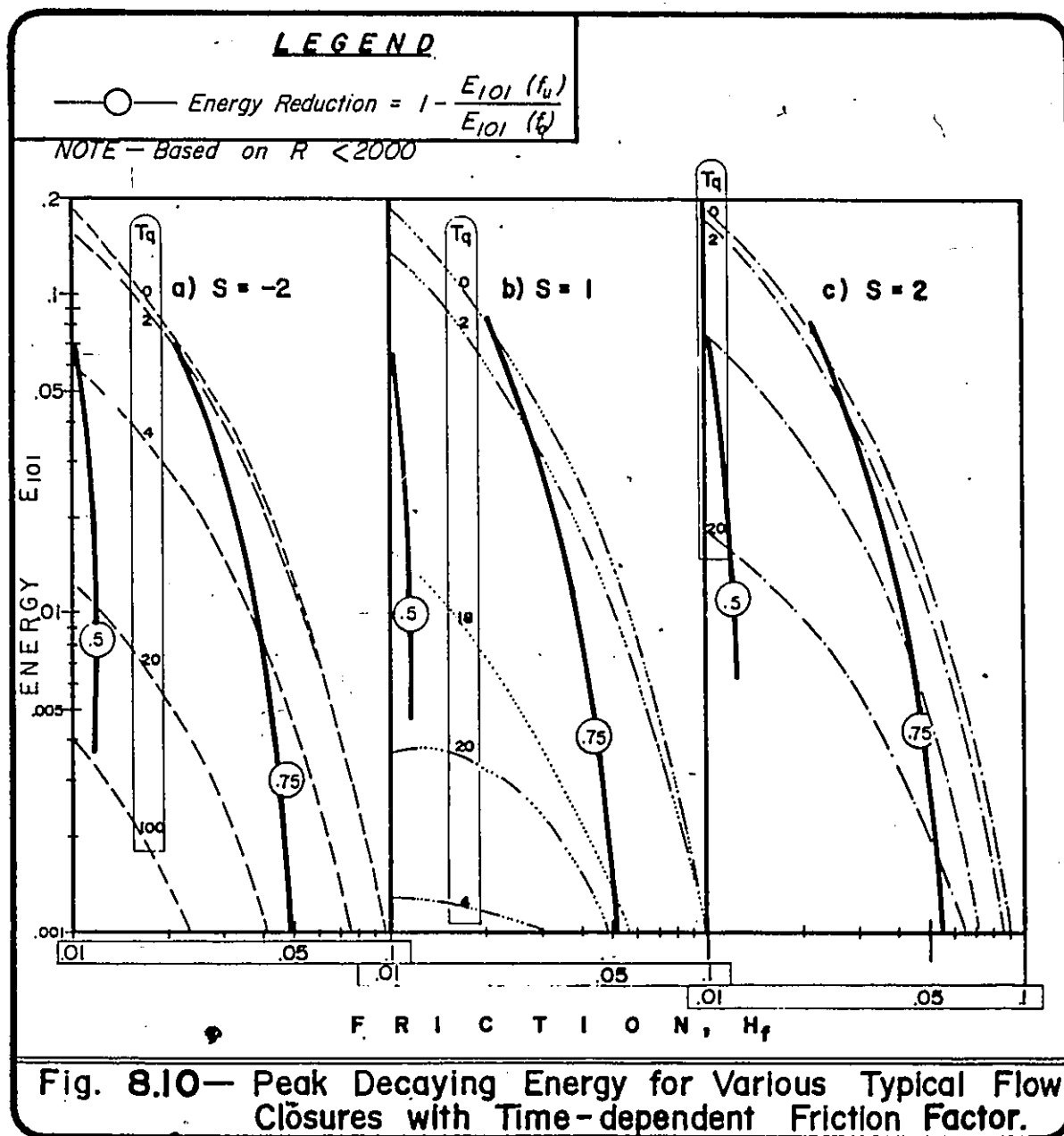
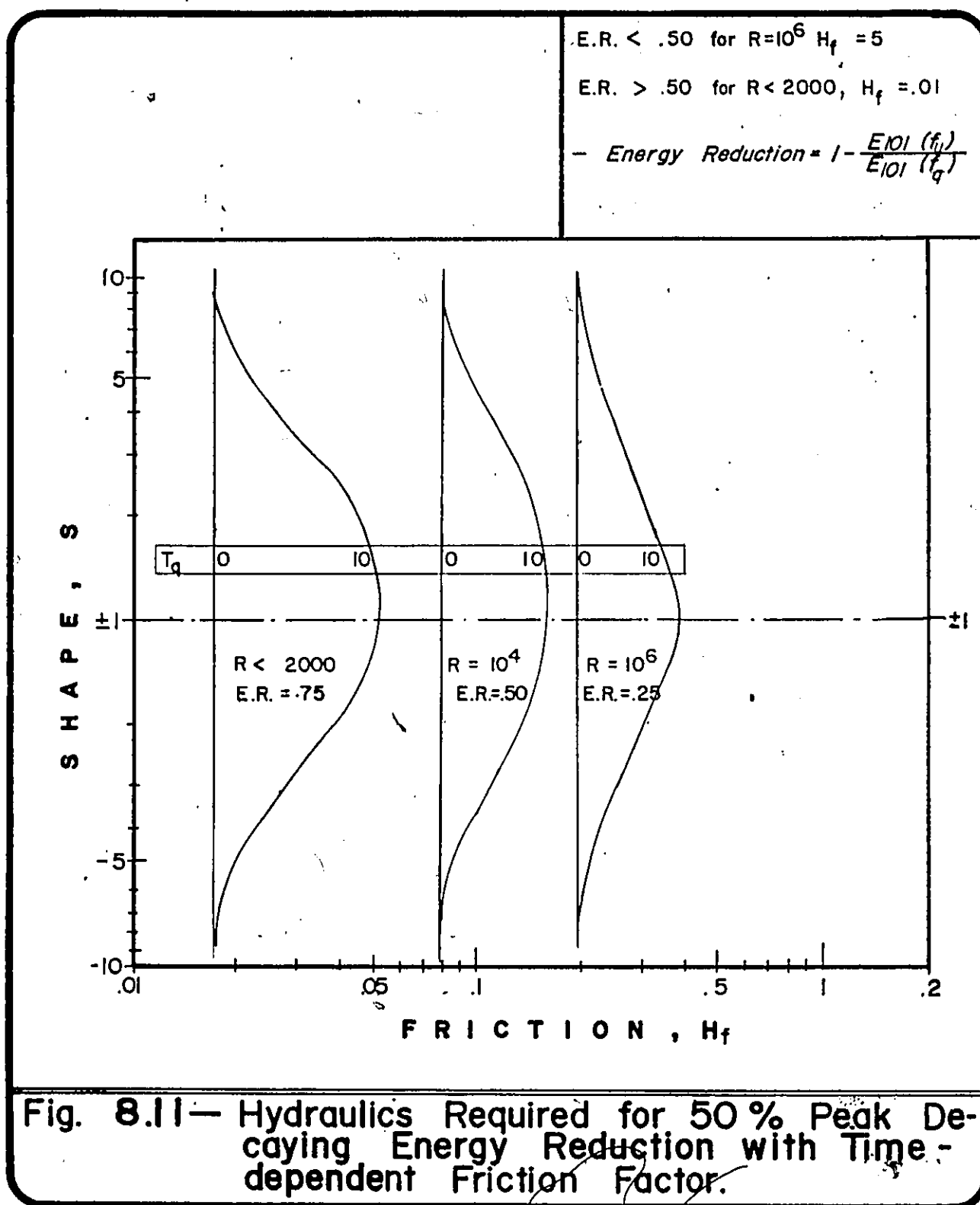
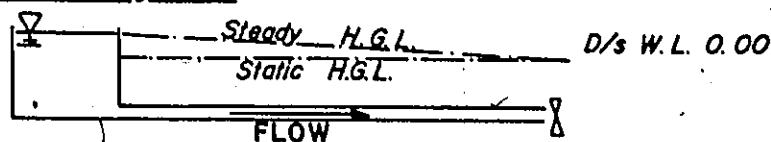


Fig. 8.9 — Effects of Time-dependent Friction Factor on Transients in Uni-directional Flows.





a) Hydraulic System.



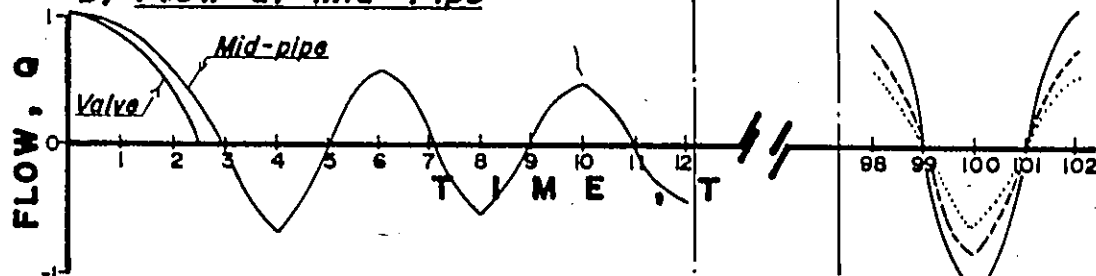
DATA

$l = 2300 \text{ m.}$ $d = 200 \text{ mm.}$ $a = 1150 \text{ m/s.}$
 $q_0 = 82.3 \text{ l/s.}$ $v_0 = 2.62 \text{ m/s.}$ $f = .013$
 $h_f = 27.5 \text{ m.}$ $h_c = 301 \text{ m.}$ $\sqrt{s} = 1.05 \times 10^{-6} \text{ m}^{3/2} \text{ s}$
 $H_f = .17$ $T_d = 2.5$ $R = 5 \times 10^5$
 $S = 1$ $H_0 = .09$ $t_c = 2 \text{ s.}$

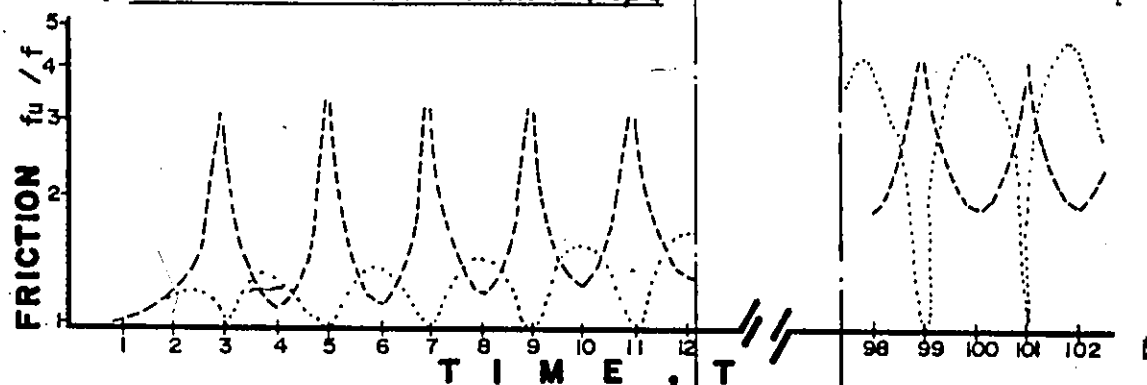
LEGEND

———— Const. " f "
 - - - - - Quasi " f_q "
 Time-Dep " f_u "

b) Flow at Mid-Pipe



c) Friction Factor at Mid-Pipe



d) Pressure at Valve

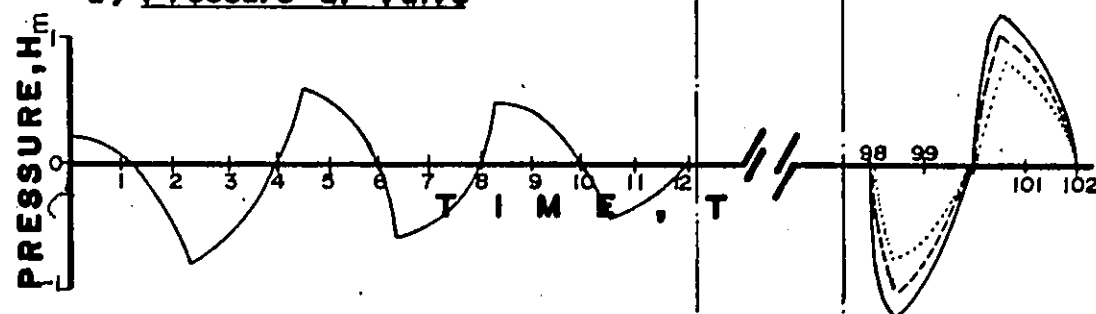


Fig. 8.12—Application of Unsteady Friction Factors in Valve Closure (an example from Apelt, 1983.).

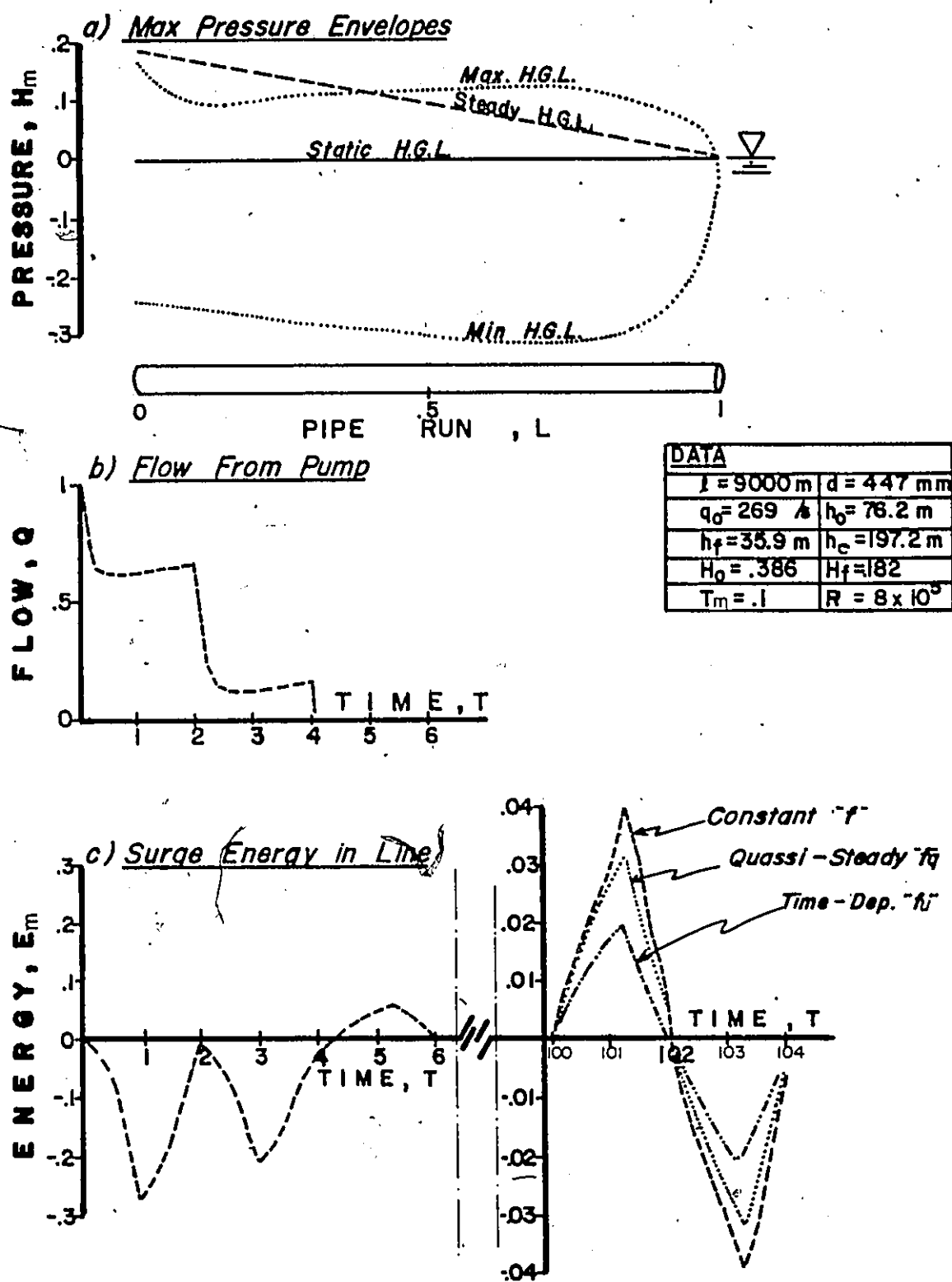


Fig. 8.13 — Application of Unsteady Friction Factors in Pumping System (an example from Apelt, 1983.).

9.0 PRINCIPAL CONCLUSIONS AND RECOMMENDATIONS

9.1 Principal Conclusions

A number of principal conclusions are listed below, and detailed conclusions appear in the text where they are substantiated.

9.1.1 General analytical approach

In order to carry out a high quality comprehensive analysis of hydraulic transients, a new analytical approach has been adopted in this work. This approach is based on:

1) dimensional analysis --- The functional relationships established by dimensional analysis, reflects the transient characteristics under various conditions and thus serves as a guide to the research direction.

2) transient modeling --- A computer model (Otthammer, 1987) based on previously well-established principles; i.e. method of characteristics (Wylie and Streeter, 1978), has been developed for the scientific research and engineering applications. This model is used to quantify the relevance of the governing parameters.

In addition to this numerical technique, physical modeling data obtained from available literature has been used to identify transient phenomena which may be unaccounted for numerically.

3) transient interpretation --- Physical pipe systems are grouped under four surge control devices (i.e. valve, pump, air chamber and surge tank). For ease of analysis, these are represented numerically by the shape and time of flow closure, where the shape of closure reflects the rate of flow deceleration and the time of closure reflects the time span of uni-directional flow.

Whereas traditional methods rely on at-a-point pressure measurements in pipe flow analyses, in this work, the state of the entire pipe system is described by adopting the use of a pressure envelope.

When this technique is used in the framework of a mass balance equation, a wholistic perspective of interpretation on transient (elastic) phenomena is provided. This is formally referred to as the energy balance concept.

9.1.2 Transient phenomena

1) Energy relation --- The term used to denote the pressure envelope in this work is called the transient (elastic) energy. Using the energy balance concept, it is shown that this energy is a function of two components: i) the energy flux into the system which is quantified by the mass balance over the pipe control volume (mechanical energy transfer), and ii) the decay of internal energy which happens due to a) pipe wall friction dissipation; and b) the energy loss in the water column itself due to periodic flow fluctuations (non-friction dissipation).

During the flow closing period, the governing parameters describing the transient phenomena is the mechanical energy transfer [case (i) above]. With the passage of time, the non-friction dissipation [case (iib) above] may also become important, while the friction dissipation [case (iia) above] is always insignificant during the transient period.

2) Elasticity --- Strictly speaking, pipe flow movement is always non-rigid (elastic) during the period of transience. However, for the purpose of performing a simplified analysis, under certain conditions, the flow may be assumed to act as a rigid body (and hence Newton's Second Law is sufficient to describe pipe flow phenomena). These conditions have been precisely identified in this

work. In general, conditions are satisfied during part of the time prior to flow closure.

Also, contrary to the traditional view, it has also been shown in this work, that though the maximum pressure envelope is linear, the flow movement may still be highly elastic and hence Newton's basic motion equation alone cannot be applied.

3) Friction classification --- Limitations inherent in traditional analytical techniques have led to conflicting results of the role of friction in transient phenomena. In this work, its effects have been classified according to: i) whether it causes an increase in the maximum surge energy or ii) a decrease; and, whether the transient phenomena are iii) dominated by friction or, iv) independent of friction.

4) Optimal surge energy --- It is commonly considered in transient flow hydraulics that a uniform flow closure produces the least transient energy in the system. However, it has been demonstrated in this research that the optimal shape of categorized flow closure varies with the friction head and the time of flow closure. Based on this finding, a procedure has been developed for the determination of the region of optimal conditions for common surge control devices.

9.1.3 Development of unsteady friction theory

1) Common understanding of friction theories --- Several well acknowledged concepts related to friction have been reviewed, and the following conclusions can be made.

- i) The general belief that friction will always reduce the transient energy is in reality only conditionally true. For example, exceptions exist in both surge and counter-surge conditions.

ii) The well-known assumption that the unsteady friction factor increases rapidly with increasing flow deceleration is not justified. Instead, this theory should in fact be stated that the unsteadiness of friction increases with decreasing flow deceleration.

iii) It is commonly held belief that the unsteady friction factor for laminar flow conditions has been thoroughly analyzed, but significant research is still needed for turbulent flow regimes. However, much more work has been done on turbulent flow conditions than on laminar ones in the case of unidirectional flows and the related surge control applications.

iv) It is currently held that the existence of an unsteady friction factor may increase transient energy or may decrease it, depending on the flow regime and hydraulic conditions. However, the systematic analysis done in this research suggests that the unsteadiness of friction can only have a one-sided effect in which the transient energy will always be reduced.

v) Considerable effort has been devoted by many researchers to the mathematical treatment of friction terms, for achieving better accuracy. However, a sensitivity analysis shows that a basic numerical treatment, i.e. linear first order iteration, has served equally well for a complete range of frictional conditions.

2) Fundamental concepts --- Many unsteady friction concepts developed from mass oscillation have been applied to general transient conditions. This assumption may have misled many previous research works tackling the transient problems in pipe systems. It is shown in this work that additional concepts need to be applied for the analysis of water hammer:

i) Elasticity of the pipe systems --- In contrast to previous studies which adopt Newton's basic motion equation to interpret the physical phenomena of uni-directional flows, it is found herein that the effect of periodic surge fluctuations and the imbalance of net flow flux dominate the characteristics of water column movement during the whole flow closing period.

ii) Friction dissipation --- Contrary to interpretation of previous research, the friction dissipation term is found to be an insignificant part of the total energy exchange in general transient conditions. Instead, it is found that the mechanical process is the primary cause of decay of transient energy.

iii) Non-friction dissipation --- It is well known that the unaccounted for decay of transient energy observed in physical modeling is caused by the distortion of the shear stress profile, due to periodic flow fluctuation. While this concept is true, the discrepancy in transient energy has traditionally been accounted for through the unsteadiness of the friction factor. However, in this research it is concluded that this anomalous energy loss is due to internal energy dissipation unrelated to the wall-water interface.

3) Current unsteady friction models --- It is concluded from this research that the theoretical background and application of current unsteady friction models has to be rethought. These traditional analyses can be grouped as: i) first and second derivative momentum friction models; and ii) linear transfer, viscous layer and time-dependent friction models for periodic transient flows.

Although all of these models are unsatisfactory from a theoretical point of view, the time dependent friction



model can, on a practical level, be applied. This is possible because the actual energy loss due to the periodic surge fluctuation may be indirectly determined by the manipulation of the unsteady friction factor of the model.

4) Application of friction models --- Conditions have been defined for the application of steady, quasi-steady and unsteady (time-dependent) friction models in general transient analysis:

i) The time-dependent (unsteady) friction model must be used if an investigation is required over a long decay period and the time of flow closure is short.

This is applied in valve (with relatively rapid closure) and pump (with check valve) systems associated with long decay periods. This is important from a design point-of-view in the consideration of the resumption of normal system operation.

ii) The sophistication of the quasi-steady friction model is necessary if the friction in the system is high, the regime is laminar and the closure is of the decreasing deceleration type.

This condition is sometimes found in valve or pump installation in machinery, oil pipelines or biological circulatory systems.

iii) Otherwise the steady friction model is adequate to describe the transient phenomena of all four surge control devices as long as the extended decaying period is not the time of interest.

9.2 Principal Recommendations

A list of recommendations is proposed to extend the present research (in the appropriate application of

friction models, realistic interpretation of transient phenomena), and, to indicate potential areas in future research...

9.2.1 Recommendations for present research

1) Development of an unsteady friction model --- Further physical tests are recommended, to provide necessary data of anomalous losses for the verification of the weighting function in the time-dependent friction theory. At present, only two sets of physical test data are available.

In addition to the three existing equations [These are: (i) Continuity; (ii) Motion and (iii) State], the fourth one [i.e. (iv) Non-friction dissipation] may be required in the method of characteristics, for the analysis of transient decay. Further analytical work in this subject may however be pursued by the incorporation of energy losses in flow separation or boundary layer movement. In this case, the disturbance of flow streamlines related to the periodic flow fluctuation should be investigated based on the actual internal energy loss, which is in a form of heat dissipation (not the hypothetical unsteady friction transfer, as assumed in current unsteady friction models).

In the meantime, the time-dependent friction model (f_u) should remain in use for predicting transient flow behavior until a better unsteady friction model is developed (or the method of characteristics is updated).

2) Analysis of common surge control devices --- By adopting a proper friction model, transient characteristics can be accurately simulated by an numerical model. Results of such an analysis should then be summarized in a form which can serve a two-fold purpose:

1) Scientific meaning --- This analysis should be appropriately formulated to reflect and quantify the

relevance of parameters governing transient phenomenon.

ii) General practice --- It should also facilitate practical application of the results obtained in this research. Comprehensive design nomographs should be completed for common surge control devices.

9.2.2 Recommendations for future research

Recommendations for further research can be put in the following four categories: i) non-elastic pipe properties; ii) minor losses; iii) vapor pressure impact; and iv) network looping.

Although the magnitude of energy losses induced by the above four areas is such that the basic conclusions rendered in this research are not affected, future research may be undertaken to quantify this potential effect and refine the assumptions made above.

1) Non-elastic behavior --- Pipe fixation, material plasticity and soil compaction are affected by the periodic shocks during the transient period. As a result for example, an unsteady surge wave speed may be adopted for the transient simulation.

2) Minor losses --- In current practice, minor losses (at junctions or bends) are usually either approximated as uniformly distributed losses along the whole pipeline, or they are considered as concentrated at the point of initiation. Although the former case may somewhat affect the analysis during the transient period (Wylie and Streeter, 1978, page 51) the assessment of the overall transient condition is not adversely altered and the present approach to analysis remains appropriate.

The latter case may in some situations be of consequence in general practice. Examples include orifice head losses

at surge tanks and air chambers; and system losses inside a pumping station.

3) Vapor pressure --- The impact of vapor pressure has not been directly incorporated in the present research and represents a limitation in scope of this work. Although vapor pressure is an important aspect of transient phenomena, difficulties associated with this subject include: i) the existence of a two-phase flow condition; ii) spuriously high pressure due to mathematical errors; and iii) data accuracy in physical modeling.

4) Network looping --- A pipe network is commonly found in municipal water supply systems. In some situations, a single pipeline can be simulated, while the network hydraulics can be taken into consideration through the method of equivalent kinetic energy (A. Fok, 1978; J. Parmakian, pers. comm., 1977).

On the other hand, there are conditions where a detailed representation of a network system is required (M. Chaudhry, 1979; R. Ruus pers. comm., 1984).

It is recommended that these four areas of future work should be assessed by physical experiments in conjunction with mathematical model studies.

APPENDICES

Appendices enclosed in this thesis consist of the following subjects related to transient characteristics in a pipe system.

<u>Section</u>	<u>Title</u>	<u>Reference in Main text</u>
A.0	ANALYSIS RELATED TO THE PRESENT RESEARCH	
- A.1	Practical Range of Friction Head	Ch. 4
- A.2	Selection of Maximum Transient Energy	Ch. 4
- A.3	Boundary of Alternate Fluctuating Flow	Ch. 5
- A.4	Optimal Energy in Uni-directional Flow	Ch. 6
- A.5	Classification of Friction	Ch. 6
- A.6	Investigation of Oscillating Flow	Ch. 8
B.0	RIGID WATER COLUMN THEORY IN PIPE FLOW	Ch. 6
C.0	TRANSIENT ANALYSIS OF CONTROL DEVICES	
- C.1	Transient Analyses of Valve Closure in a Pipeline	
- C.2	Transient Analyses of Pump trip-out in a pipeline	
- C.3	Transient analyses of Air Chamber in a pipeline	
- C.4	Transient analyses of Surge Tank in a pipeline	
D.0	SOME CONTRIBUTIONS FROM THE THESIS	

A.0 ANALYSES RELATED TO THE PRESENT RESEARCH

A.1 Practical Range of Friction Head (H_f)

A.1.1 Parameters governing friction head

The dimensionless friction head (H_f) adopted in this research ranges from 0 to 10. In order to assess this range of practical applicability, Fig. A.1 has been constructed by relating the friction head to only two parameters (i.e. pipe diameter and flow velocity) for a 1000 m long smooth watermain with concrete or steel material (i.e. surge wave speed, $a = 1000$ m/s). Then the functional relation expressed in Fig. A.1 is:

$$H_f = F(d, v) \quad \text{A.1}$$

where the friction ' H_f ' is non-dimensional, while system diameter ' d ' and velocity ' v ' are dimensional terms.

A.1.2 Practical frictional ranges

Results obtained from the figure suggest that under a normal operational condition, the friction head will not exceed the value of:

- i) unity, even for an abnormally long (say, 10 km) concrete or steel watermain;
- ii) ten, for an abnormally low wave speed, 100 m/s (e.g. the pipe is replaced by plastic material); and
- iii) ten, for the 500 km steel pipeline carrying oil (say, 10 times the water viscosity) with normal oil pipeline diameter ($d > 0.5$ m.) and oil flow velocity ($v < 2$ m/s.).

A.1.3 Practical pipe hydraulics related to friction

In addition to these, several points related to the practical range of the friction head should be noted:

- i) A plastic pipe would rarely be used for a steady pressure greater than 100 m. This is due to its relatively low design pressure (H_a) in the plastic piping design (i.e. $H_a = H_s + H_f \ll 10$, where H_s is the static head);
- ii) A Hazen-William friction factor of pipe is seldom below 130 in new long water or oil mains. This frictional condition of new pipe corresponds closely to the vicinity of the hydraulically-smooth zone;
- iii) The main line between the supply source (pump or reservoir) and the receiving end (reservoir or standpipe) is frequently cleaned or relined in order to minimize the energy loss in an operational water or oil supply network. It has been found that the Hazen-William friction factor in the main line is usually above 120 in which a smooth pipe can be considered. It should be noted that hydraulic conditions (for both steady and unsteady flows) in the main line normally dominate the rest of the network system;
- iv) The pipe roughness of an old sewerage forcemain is much higher and cannot be considered as a smooth pipe. However, a sewerage forcemain is usually associated with a relatively low heads (e.g. $h_a = h_s + h_f < 50$ m.) resulting in $H_f \ll 10$; and
- v) The pipe friction factor would be high in a system with very small diameter, highly elastic material and large fluid viscosity (such as blood vessels). Possibly it becomes a laminar flow. On the other hand, the flow velocity in this system would be very low (say, $v < 0.1$ m/s.). Consequently, the friction head would not likely exceed the value of 10.

A.1.4 Conclusions of frictional head ranges

From above discussions, it appears that the range of the dimensionless friction head adopted in the present study ($0 < H_f < 10$), covers all the normal hydraulic conditions in a closed conduit system. Furthermore, flow closure with higher friction head ($H_f > 10$) does not likely induce transient pressure high enough to require surge protection.

Although this work focuses on water systems, it is noted that the range of conditions tested far exceeds this scope, and implies the ready application of the results contained here to non-hydraulic systems.

A.2 Selection of Maximum Transient Energy

A.2.1 Definition of maximum energy

The maximum value of transient energy (E_{max}) was previously defined in the main text as:

$$E_{max} = \int_0^1 H_m(X, T_{max}) dX \quad (4.10)$$

It records the maximum value of transient energy encountered during the time the transient existed in the system, and hence can be considered as the time-maximum transient energy (E_{max}).

Another term used in this research to evaluate the critical transient events is the envelope-maximum transient energy (E_{max}'), which is defined as the integration of $H_{max}(X)$ along the pipeline. Here $H_{max}(X)$ is the maximum transient pressure encountered at a point X over the entire time history of the transient period. The envelope-maximum transient energy can be shown as below:

$$E_{\max}' = \int_0^1 H_{\max}(X) dX \quad \text{A.2}$$

$$= \int_0^1 \int_0^{T_{\max}} H_m(X, T) dT dX$$

where the energy is developed from a double integration, in which the time-maximum value, $E_{\max}$ (defined in Eq. 4.10) is part of it.

Since the maximum pressure envelope contains points from different times, the envelope-maximum transient energy is an imaginary term which has not existed in the transient history. However, the adoption of this conservative envelope-maximum transient energy ($E_{\max}'$) is preferred in practical engineering analysis and design.

A.2.2 Deviation of results

If the difference between these two energy terms is relatively small, they can be merged into one term for the purpose of simplicity. With this in mind, a comparison between the envelope and the time-maximum transient energy has been conducted for various hydraulic conditions, including both non-frictional and frictional systems. The results of this comparison are summarized in Fig. A.2. For simplicity, the analysis related to the non-frictional system would be interpreted at $H_z = 0.01$ in the figure. The main findings of this comparison are that:

- i) In a non-frictional pipe system, the envelope-maximum surge energy seldom exceeds 5 % of the time-maximum for an increasing deceleration flow, and 10 % for the flow with decreasing deceleration;
- ii) For an instantaneous flow closure, difference of greater than 5 % between the envelope and time-maximum

surge energy occurs at the friction head (H_f) between 0.2 to 2 with the maximum error 15% at about $H_f = 1$;

iii) For a rapid flow closure ($T_q > 2$), the difference between these two terms (i.e. E_{mx1} vs. E_{mx1}') diminishes with the increase of friction (H_f); time (T_q) and shape factor ($|S|$) as for both deceleration flows;

iv) At very high friction head (say, $H_f > 2.5$), no difference can be found as friction plays the major role during the flow closing period; and

v) The deviation between the envelope and time-maximum surge energy decreases with the progress of time. Hence, the present comparison results of maximum surge energy (Fig. A.2) can be used as a conservative consideration for the maximum counter-surge energy, and other subsequent peak decaying transient conditions.

A.2.3 Applications of energy terms

i) Due to the relative small difference between these two terms in the majority of practical applications (i.e. less than 5 - 10 % error in $T_q > 0$, as shown in Fig. A.2), the envelope-maximum surge (E_{mx1}) and counter-surge (E_{mx2}) energy may be used (on the conservative side) as the time-maximum when depicting the transient history by the use of design nomographs available in the literature; and

ii) Conversely, only the time-maximum surge energy is used for the analysis of the peak decaying energy (E_r) due to the following reasons:

a) it is extremely difficult to update the decaying maximum surge pressure envelope due to the small change of maximum pressure (at each point along the line) in each surge fluctuation period;

b) the difference between the time- and envelope-maximum surge energy diminishes as the time progresses; and

c) it is more convenient to plot the transient history by the given magnitude and the time of occurrence of the peak decaying events.

A.2.4 Conclusions for selecting energy terms

For the purpose of simplicity and consistency, unless otherwise specified, the present thesis adopts the symbol E_{mx} (i.e. the term E_{mx} has been eliminated) to represent:

i) the envelope-maximum energy during the investigation of maximum surge (E_{mx1}) and maximum counter-surge energy (E_{mx2}); and

ii) the time-maximum energy for the historical event of transient energy associated with the energy balance application, and the peak decaying energy which occur after the first counter-surge period (i.e. E_T , for $T > T_{mx2}$).

A.3 Boundary of Flow with Alternate Additive Fluctuation

A.3.1 Criteria governing the flow interception

Recall the equation of source flow closure with increasing deceleration, and rewritten as below:

$$Q_s(T) = 1 - (T/T_q)^2 \quad (4.21)$$

In order to achieve an alternative additive flow, the pipe-end flow must drop below the source flow during the period of first wave reflection at pipe-end ($1 < T_{rs} < 3$). Hence, applying the constant rate of wave reflection ($RQ_{rs} = 2$), the pipe-end flow becomes:

$$Q_m(T) = 1 - 5Q_{ms}(T) * [(T-1)/T_d]^2 \quad A.3$$

$$= 1 - 2 * [(T-1)/T_d]^2$$

where $1 < T < 3$.

A.3.2 Shape of flow closure for flow interception

The time of the first intercepting point of two flows (T_{ms}) can be obtained for the shape of flow closure (S), by solving the following equation which combined Eq. 4.21 and A.3. This is $Q_s = Q_m$, which implies that:

$$1 - (T_{ms}/T_d)^2 = 1 - 2 * ((T_{ms} - 1)/T_d)^2 \quad A.4$$

It can be restructured and simplified as:

$$(T_{ms}/(T_{ms} - 1))^2 = 2 \quad A.5$$

Solving Eq. A.5, the region boundaries of alternate additive fluctuations can be derived as:

- i) for $1 < T_{ms} < 2$, $S < 0$;
- ii) for $T_{ms} = 2$, $S = 1$; and
- iii) for $T_{ms} = 3$, $S = 1.70$

Case i) should be omitted as Eq. A.5 is not developed for flows with decreasing deceleration.

From the results of Cases ii) and iii), it appears that the range of shape factor should fall somewhere:

$$1 < S < 1.7 \quad A.6$$

A.3.3 Conclusion of alternate fluctuation

The upper boundary of alternate additive fluctuations, which has been adopted in the main text, is valid for the transient analysis in a non-frictional pipe system.

Relevant transient properties in uni-directional flows

In general, relevant properties related to uni-directional flows have been introduced during the course of the analysis. Three most important among these are:

- i) Rigid water column theory;
- ii) Optimal surge energy; and
- iii) Friction classification.

The first one, Rigid water column theory, due to its broad scope of analysis and lengthy discussions, will be presented separately in Appendix B. The other two are however discussed in A.4 and A.5.

A.4 Optimal Surge Energy in Uni-directional Flow

A.4.1 Functional relationship of optimal shape factor

An analysis of the optimal flow closure on maximum surge energy has been conducted and summarized in Fig. A.3. The relationship in the figure can be expressed as:

$$S^* = F(H_z, T_d) \quad A.7$$

A.4.1.1 Non-existence of periodic fluctuations

As shown in the figure, for $T_d < 2$ the optimal shape factor (S^*) of flow closure is:

$$S^* = 1 \quad A.8a$$

where H_z is any value.

This finding is reasonable since the periodic alternate fluctuation has not been created for $T_d < 2$.

A.4.1.2 Effect of periodic fluctuations

On the other hand, the optimal shape of flow closure decreases with the increasing friction for $T_a > 4$. For example, the optimal shape factor is:

$$S^* = 1.5$$

A.8b

where $H_x = 0$.

Then at $H_x = 1$, the optimal shape factor varies as:

$$S^* = -1.25 \text{ for } T_a = 4; \text{ and}$$

A.8c

$$S^* = -3.00 \text{ for } T_a = 100.$$

A.8d

A.4.1.3 Friction dominant zone

The plots of the optimal shape curves in Fig. A.3 terminate at the boundary of the Friction dominant zone where the effect of flow closing characteristics on maximum surge energy disappears.

A.4.2 Practical applications in optimal operation

Techniques for the determination of optimal surge energy are not well developed. This is demonstrated through examples provided in the following subsections.

A.4.2.1 Optimal flow closure for Physical tests

Uniform flow closure has most often been selected for experimental tests (Daily 1956, Carstens and Roller 1959 etc). One of the main reasons for this choice is that based on Newton's second law of motion below:

$$h_{max1} = (l/g) * (dv/dt) + f * (l/d) * (v^2/2g) \quad A.9$$

uniform deceleration ($dv/dt = 1$) should produce the optimal transient condition.

However, this finding contradicts what has been presented in the Figure A.3.

A.4.2.2 Optimal orifice in Air chamber installation

In their publications (Ruus 1977 and Chaudhry 1979), both authors suggested that optimal surge pressure in air chambers occurs without the installation of an orifice. That is:

$$H_{ox} = 0 \quad \text{A.10a}$$

Actually, a low orifice loss (H_{ox}) may frequently induce a more ideal flow pattern (e.g. $S = S^* = 1.5$ in low friction). For example:

$$H_{ox} = 0.1 * H_o^* \quad \text{A.10b}$$

where H_o^* is the chamber head,

This low orifice head (Eq. A.10b) may result in an optimal transient condition (Fok 1978). Without the orifice installation, the high pressure inside the chamber often discharges flow into the line when $S > S^* = 1.5$.

A.4.2.3 Optimal valve operation

1) Non-frictional system ---

A similar condition can be found in publications of valve operation (Wood and Jones 1973, and Ruus et al 1980). The authors suggested that the optimal surge pressure occurs at the highest valve head, in the absence of friction. For an example of butterfly valve (an equivalent shape factor of valve closure), the optimal surge energy occurs at [see Fig. 4(f) of the paper]:

$$H_0 > 0.5$$

A.11a

Analyses from the present model simulations shows that the optimal surge energy of the butterfly valve is located at:

$$H_0 = 0.1$$

A.11b

In reality, for the shape (S) of valve closure (less than 1.5, the optimal surge energy occurs at:

$$0.1 < H_0 < 0.5$$

A.11c

These consist of at least half of the common types of valves presented in their publications, including uniform, ball, square gate and butterfly valves.

2) Frictional system---

Conditions required for achieving optimal valve closure have been developed. Fig. A.4 covers seven common surge control devices (equivalent to $-1.85 < S < 2.75$).

As shown in the figure, in the case of low friction and low valve head, the optimal valve shape factor approaches minus infinity, such as: ---

$$S^* \ll -1.85$$

A.12a

where $H_z < 0.01$ and $H_0 < 0.01$.

This is consistent with the findings from Wood and Jones (1973) and Ruus et al (1980).

On the other hand, at high valve head condition, the optimal shape factor of valve closure approaches:

$$S^* = 1.35$$

A.12b

where $H_0 > 10$.

A.4.3 Conclusions of optimal surge energy

1) It has been found that the optimal shape factor (S^*) is 1.5 in a non-frictional system. This decreases with the increasing friction head (H_f) and time of flow closure (T_a), until it reaches the friction dominant condition.

2) Traditional analysis conceived that the uniform flow closure produces the least surge energy. This misconception has been felt in engineering practices in,

- i) the selection of physical modeling test;
- ii) the analysis of low-loss orifices in air chambers, and
- iii) the application of low shape factors for valve design.

A.5 Classification of Friction

A.5.1 Traditional theoretical concepts

There has not been a good understanding of the role of friction affecting the maximum surge energy. Inconsistent conclusions have been made as discussed below.

A.5.1.1 Maximum transient energy decreases with friction

It has long been a general belief that the existence of friction would only result in less transient energy in the piping system (e.g. Stephenson 1966). In reality, the introduction of friction in the system can also increase the maximum transient energy as previously discussed in Ch. 6.0.

The incompleteness of the traditional concept is partly due to the misconception regarding the importance of friction dissipation (ED_f). It may also be partially attributed to the current use of the rigid flow motion in interpreting the transient phenomena. Based on the basic motion equation

(Eq. A.9), it is held that the higher the friction term, the smaller the maximum surge energy that could be obtained.

The analyses of both non-frictional (ET_w) and frictional transfer (ET_z) processes affecting the maximum surge energy (E_{mx1}) and maximum counter-surge energy (E_{mx2}) have not previously been established.

A:5.1.2 Surge energy decreases with friction

Conclusions opposite to the above have been made quite frequently by some researchers. For example, Ruus concluded that maximum surge pressure increases with friction in his publications on air chamber and pump design charts (1977, 1981 and 1985).

In his analysis, the dimensional maximum surge pressure head (h_{mx1}'') at pump was used, such as:

$$\begin{aligned} |h_{mx1}''| &= h_a + |h_{mx1}| \\ &= h_z + h_s + |h_{mx1}| \end{aligned} \quad A.13$$

where h_s is the static head at the source (Refer to Fig. 1.1 for the schematic of a pump system).

As refer by Eq. A.13, in the event of the increase of friction head (h_z), the maximum surge pressure $|h_{mx1}''|$ will definitely increase, if the decrease of real maximum surge energy $|h_{mx1}|$ is relatively small (i.e. $|h_{mx1}| < h_z$). Therefore, it could be concluded that the adoption of this traditional dimensional parameter (h_{mx1}'') fails to reflect the real relationship between the maximum surge pressure (h_{mx1}) and friction (h_z) parameters.

Further, conclusions drawn from the application of this traditional dimensional parameter can not be justified from dimensional reasoning.

A.5.1.3 Numerical experimental observation

During the analysis of a uniform valve closure, Stephenson (1966) observed that the maximum surge pressure (H_{mx1}) increased with friction (H_f), such as:

$$E_{mx1}(H_f > 0) > E_{mx1}(H_f = 0) \quad A.14$$

The author of the paper explained the relation of Eq. A.14 as an abnormal event and attributed its possible cause as an excessive incoming flow due to the malfunction of valve operation.

In reality, the hydraulic conditions used in his numerical experiment consisted of a relatively wide range in friction head ($0 < H_f < 1$), a long time of flow closure ($T_q > 4$) and a small valve head ($H_o < 0.1$) in a uniform valve closure ($S = 1$). It should be noted that the combination of small valve head ($H_o < 1$) and uniform valve closure ($S = 1$) results in an equivalent high shape factor of flow closure (i.e. $S \gg 1.7$). This specific event falls into the region where the maximum surge energy should actually increase with friction. Perhaps, this misinterpretation came from Stephenson's assumption that the friction will naturally function as an energy damper during the entire flow closing period.

A.5.2 Friction classification in hypothetical flows

A comprehensive analysis of friction effects on maximum surge energy has systematically been conducted in this research, covering the wide range of frictional (H_f) and flow closing conditions (S and T_q).

A.5.2.1 Definition of zoning on friction classification

Basically, the effects of friction on maximum surge energy can be categorized into four zones, namely:

i) Friction - free --- in which the change of friction does not significantly effect the magnitude of maximum surge energy (i.e. $dE_{mx1}(H_f) / dH_f = 0$);

ii) Intermediate - I --- in which the maximum surge energy decreases with friction (i.e. $dE_{mx1}(H_f)/dH_f < 0$);

iii) Intermediate - II --- in which the maximum surge energy increases with friction (i.e. $dE_{mx1}(H_f)/dH_f > 0$); and

iv) Friction - dominant --- in which the same maximum surge energy would be obtained independent of the individual characteristics of flow closure (i.e. $E_{mx1} = F(H_f)$, where S and T_a are eliminated). In this condition, maximum surge energy also decreases with the increasing friction head (i.e. $dE_{mx1}(H_f) / dH_f < 0$).

A.5.2.2 Characteristics of zoning

The characteristics of the friction classification are summarized in Fig. A.5 for various hydraulic conditions.

1) Increasing deceleration ($S > 1$) ---

i) Friction - free zone --- For flow with an increasing deceleration, the friction is insensitive in the low friction head area, i.e. $H_f < 0.2$.

ii) Intermediate - I zone --- For the case of rapid flow closure in a moderate-high friction system ($T_a < 2$ and $H_f > 0.2$), maximum surge energy decreases with increasing friction. However, this condition diminishes with increasing time of flow closure.

iii) Intermediate - II zone --- The energy would increase with friction when flow closes slowly with moderate to high frictional systems, i.e. $T_a > 4$ and $0.2 < H_f < 2$.

iv) Friction - dominant zone --- The friction will entirely dominate the surge conditions when it is sufficiently high, e.g. $H_f \gg 2$.

2) Decreasing deceleration ($S < -1$) ---

i) Friction - free zone --- For flow with decreasing deceleration, the insensitive friction condition is reduced to a very small area where the friction head is small, but the time and shape factor of flow closure are high.

ii) Intermediate - I zone --- The maximum surge energy decreases with friction within low to moderate friction heads.

iii) Intermediate II zone --- The maximum surge energy will increase with friction for the slow flow movement in a high friction system, e.g. $T_a > 10$ and $H_f > 1$.

iv) Friction - dominant zone --- Similarly, friction will dominate the transient conditions when it is sufficiently high, e.g. $H_f \gg 2$.

A.5.3 Friction classification for practical applications

The effect of friction on common surge control devices has been classified:

A.5.3.1 Valve operation in friction classification

Results for $0.01 < H_f < 10$, $0 < T_a < 100$ and $0.01 < H_o < 1$ are summarized in Fig. A.6.

1) High valve head ($H_o = 1$) ---

In a high valve head condition ($H_o = 1$), a similar condition to that in categorized flows (Fig. A.5) is found, because the effect of the valve head (H_o) is insignificant.

Consequently the flow through the orifice follows the pattern of the valve shape factor.

2) Low valve head ($H_0 = 0.01$) ---

However, towards the low valve head ($H_0 = 0.01$), E_{mx1} increases with friction for most of the hydraulic conditions. In this case of low head valve, the flow decelerates with $S \gg 1.7$.

A.5.3.2 Other devices in friction classification

Fig. A.7 summarizes the friction classification for three other control devices, i.e.

- i) Pump;
- ii) air chamber; and
- iii) surge tank (with orifice).

The mechanical time scale of the device can not be predetermined, and $T_m \gg T_q$. With the adoption of (T_m), two friction zones are established:

- i) Friction - free, where the friction is insensitive to the transients during the surge period; and
- ii) Intermediate - I, where the maximum surge energy decreases with friction.

A.5.4 Conclusions of friction classification

1) Theoretical background --- The effect of friction on maximum surge energy (E_{mx1}) has not been well interpreted. Some research workers concluded that friction should reduce the maximum transient energy, while others did the opposite. The cause of this conflict is due to:

1) Error in dimensional reasoning for the formation of grouped parameters;

ii) misconception of friction dissipation concept for the analyses of transients; and

iii) lack of a systematic and comprehensive analysis of friction effects on transients (with steady factor).

2) New friction classification --- The role of friction has been re-classified into four zones for categorized flows. These are:

i) Friction - free --- where the existence of friction does not affect the transient;

ii) Intermediate - I --- where the maximum surge energy decreases with the increasing friction;

iii) Intermediate - II --- where the energy increases with friction; and

iv) Friction-dominant --- where friction dominates the transient condition, independent of flow closing characteristics.

3) Practical application --- Friction classification has been made on four common surge control devices, namely valves, pumps, air chambers and surge tanks.

A.6 Investigation of Oscillating Flows

An investigation of unsteady flow phenomena in mass oscillations is necessary for a complete understanding of the history of current unsteady friction models developed for general transients.

A.6.1 Theoretical Background

A U-tube configuration is commonly used in the investigation of oscillating flow movement in a gravitational field. Newton's second law of motion forms the basis of the investigation, where the pressure head (h) in a non-frictional system is:

$$h_m = l/g * dv/dt \quad A.15$$

and the equation in a (steady) frictional system is updated:

$$h_m = l/g * dv/dt + f * v^2/2g \quad A.16$$

As shown in the L.H.S. of Fig. A.8a, the U-tube mass flow oscillates as a rigid water column and its motion has traditionally been derived as:

$$d^2h/dt^2 + \tau_w * (dh/dt)^n + 2g/l * h = 0 \quad A.17$$

where n is an exponent = 1 and 1.75 for the laminar and turbulent flow respectively, and τ_w reflects the properties of wall shear stress which can theoretically be determined as:

$$\tau_w = 0.5 \rho * v^2 * f_a \quad A.17a$$

The relationship of Eq. A.17 is basically the same as the well known Newton's Second Law of Motion (A.16), except that the friction term has been modified to account for the effect of a quasi-steady friction factor. This quasi-steady friction term (Eq. A.17a) has long been of concern. Numerous laboratory experiments have been conducted in order to refine this equation. The general conclusion of these investigations is that:

In addition to the quasi-steady friction factor, the wall shear stress also depends on the frequency (ω) or acceleration (dv/dt) of flow oscillation.

For example, Vardy (1980) and Safwat et al (1973) suggested a relationship (equivalent to that of Eq. 2.1 and 2.2), in which the unsteady shear stress thus proposed is in a form as:

$$\tau_w = 0.5 \rho * v^2 * f_a + b * \rho * (dv/dt) \quad A.17b$$

where the coefficient, b , is determined from physical model tests.

A.6.2 Numerical model set-up and verification

In this research a hypothetical piping system is set up for the use of the present numerical model (by means of method of characteristics) for the U-tube simulation as shown in R.H.S. of Fig. A.8a. This contrasts with prior work where an analytic solution is applied, and incidently serves as a secondary validation of the computer model. Furthermore, the adoption of this hypothetical system provides a means to bridge work between "experiment in Physics" and "application in Engineering".

The hypothetical U-tube system possesses the following features:

- i) Same diameter as the U-tube system;
- ii) Same length of the frictional pipe being placed horizontally, which results the same frictional head in the system; and
- iii) Same height of vertical (frictionless) standpipes at two ends of the system, which provides the same effect of gravitational force but does not induce any additional friction effect.

An example is selected from Streeter and Wylie (1981, page.485) for the model verification and for the later discussion of unsteady flow phenomena in the next subsection. The system is under a laminar flow condition. Another two examples have been established from the same

system configuration but with some modification of the friction factor, one for frictionless and the other for constant friction factor systems.

The relevant data for these three examples is summarized in Fig. A.8a. The transient flow and pressure history have been plotted in Fig. A.8b and A.8c. This schematic model should be considered to reliably represent real U-tube hydraulic conditions because:

Exact unsteady flow movement and pressure variations have been simulated in comparison to that derived from Eq. A.17, in both non-frictional and quasi-steady friction (factor) systems.

Results derived from these numerical simulations are used for the investigation of energy relations in U-tube mass oscillation.

A.6.3 Unsteady flow phenomena

A.6.3.1 Elasticity in the system

The case of the non-frictional U-tube system is proposed for the investigation of its rigidity in unsteady flow movement. The relevant data consists of:

$$a = 1000 \text{ m/s}, v_0 = 0.5 \text{ m/s}, h_c = 50 \text{ m}, t_c = 0.003 \text{ sec.}$$

1) Elasticity in a U-tube system ---

The time duration from the instant at maximum flow velocity $v = 0.5 \text{ m/s}$ to the moment at $v = 0$ is considered as the time period of flow closing. In this example, $t_a = 0.6 \text{ seconds}$ which is one-quarter of the oscillating period ($t_s = 2.4 \text{ s}$). Consequently, the dimensionless time of flow closure is $T_a = t_a / t_c = 200$. As shown in Fig. A.8b, the flow variation in the U-tube oscillation can be approximated to an equivalent shape of flow closure $S = 2$.

The control volume of the U-tube system (R.H.S. of Fig. A.8a) is defined in a manner similar to that of the pipe system. The maximum drop of pressure head ($h_{mx1} = 0.4\text{m}$) is equivalent to:

$$\begin{aligned} H_{mx1} &= h_{mx1} / h_c \\ &= 0.4 / 50 \\ &= 0.008 \end{aligned}$$

In this case, the incoming and outgoing flow rate are identical, resulting in zero netflux during the flow oscillations as expressed below:

$$Q_s(T_1) = Q_x(T_1)$$

and

$$E_w(T_1) = E_{wx}(T_1) - E_{ws}(T_1)$$

$$\begin{aligned} &= \int_0^{T_1} Q_x(T) dT - \int_0^{T_1} Q_s(T) dT \\ &= 0 \end{aligned}$$

where T_1 is any time during the unsteady flow motion.

This non-elastic behavior can be shown in the H.G.L. of Fig. A.8a. The elastic energy in the system remains at zero (referred to the datum), independent of the water level fluctuations inside the tube.

2) Elasticity in a pipe system ---

Assume a flow closure with the above mentioned flow closing properties of $T_a = 200$ and $S = 2$ in a pipe system. Since the water level at both source and pipe-end does not vary, the change of pressure along the system has to be transformed into a form of elastic energy stored in the system, and consequently the net energy flux accumulated during the flow closing period is no longer zero.

In this example, the inflow at the source would vary exactly the same as shown in Fig. A.8b. However, the outflow at the pipe-end would have a secondary component of periodic fluctuations with a period $T_s = 4$, to be superimposed on the source inflow. These periodic surge fluctuations are similar to the flow conditions shown in Fig. 5.4a, but with a much smaller scale. The maximum energy outflux accumulated during the flow closing period (assume $T = 0$ at the instant of maximum flow velocity) would be:

$$\begin{aligned}
 E_w(T_s) &= E_{max} * v/a * l \\
 &= (0.5 * 0.008) * 0.5/1000 * l \\
 &= 2 * 10^{-6} * l \\
 &= 6 * 10^{-6} \text{ m. of pipe} \\
 &= 3 * 10^{-3} \text{ cc of water}
 \end{aligned}$$

Instead of a drop of 0.4 m water level between both ends of the pipe as a U-tube, the kinetic energy of the pipe system has been converted to the compression or expansion of the water and pipe wall, which is equivalent to $2 * 10^6$ times the pipe capacity.

This demonstrates that even though the flow movement in a pipe system may be accurately predicted by the rigid water column theory, the real phenomenon governing the transient flow conditions is the elastic energy induced by unbalanced flow motions along the system (no matter how small it is).

A.6.3.2 Friction dissipation in a steady friction system

An example of a constant friction (factor) system is used for the investigation on the oscillation damping. The mechanical process of friction transfer (ET_x) is certainly not the governing phenomenon in a U-tube system, since the essential condition of the elastic energy does not exist. Instead, the decay of unsteady flow movement depends entirely on the friction dissipation (ED_x), in a form of heat exchange.

The relevant data in the present example consists of:

Dimensional --- $f = 0.051$ and $h_z = 0.062$ m at $v_o = 0.45$ m/s;
 $h_c = 45$ m and $t_c = 0.003$ s.

Non-dimensional --- $H_z = 0.0014$, $T_a = 200$ and $T_z = 800$.

Within a complete period of oscillation, the energy loss induced by friction dissipation would be approximated as:

$$\begin{aligned} ED_z(T_a) &= \int_0^{T_a} Q(T) dT = v/a * T_a * E_z \\ &= 0.45/1000 * 800 * (0.5 * H_z) = 0.18 * H_z \\ &= 0.011 \text{ m. (written in a dimensional unit)} \end{aligned}$$

The friction dissipation of 0.011 m is comparable to that observed in the drop of water level within a period of oscillation (Fig. A.8c).

The significant effect of friction dissipation on damping the oscillating movement in a U-tube can be observed in Fig. A.8b and A.8c.

A.6.3.3 Effect of Quasi-steady friction (factor)

The case of the original example (Streeter and Wylie 1981, page 485), was actually based on the quasi-steady friction model. The laminar flow with $R = 1250$ occurs at the instant of maximum flow velocity (i.e. $t = 0.6$ sec. in Fig. A.8b). The significance of the quasi-steady friction factor effect can be clearly observed in Fig. A.8b and A.8c. This finding is reasonable as this specific U-tube system possesses two important properties related to friction:

- 1) The magnitude of friction energy in this system is significant in relation to the energy loss induced by friction dissipation during each oscillating period (i.e. $ED_z > 0.18 * H_z$ for $0 < T < T_a$); and

11) The laminar flow ($R = 1250$ in this example) is the most effective flow regime in terms of the impact of quasi-steady friction factor on the damping of unsteady-flow movement.

A.6.3.4 Energy loss due to flow reversal

Analyses conducted in this thesis (Ch. 8) show that Zielke's time-dependent friction model realistically predicts the energy loss induced by the flow reversal process in the bi-directional flow. This time-dependent friction model is now adopted for the U-tube unsteady flow simulation.

The effect of this unsteady friction model application is zero in the present example of U-tube oscillation. This finding should be expected as the process of flow reversal in the U-tube system is slow and gradual (in terms of surge wave travelling). This is in contrast with the abrupt change of periodic transient flow direction and magnitude in the pipe system with the period of fluctuation ($T_s = 4$). Consequently, this U-tube oscillation results in only large scaled periodic oscillation (the period of flow is much greater than that of surge wave, T_s) and/or flow separation upon flow reversal (if there is any).

A.6.3.5 Distortion of shear stress distribution

It is well acknowledged that the shear stress profile may be disturbed with the increasing frequency (ω) or acceleration (dv/dt) of oscillating flow movement (Richardson and Tyler, 1929). Consequently the unsteadiness of the friction factor is increased. This finding is perfectly possible and reasonable based on the following evidence:

1) The essentially rigid flow movement in a U-tube system eliminates elastic behavior otherwise so important in determining the transient characteristics in pipe systems. These include the rapid periodic fluctuations resulting from the wave reflection.

ii) The absence of the friction transfer eliminates energy exchange phenomena unrelated to the traditional friction dissipation concept. For example, the cases where potential transient energy increases due to the existence of friction, i.e. $E_{max}(H_z > 0) > E_{max}(H_z = 0)$, will never occur in a U-tube system.

iii) The oscillating flow reversal is relatively slow and gradual, thus virtually eliminates the occurrence of other potential energy losses unrelated to the friction factor.

The above three events reduce the effect of other potential energy losses regardless of the friction dissipation, thus simplifying the analysis of unsteady flow phenomena in a U-tube system. This also implies the validity of Newton's basic motion equation for application in U-tube systems.

iv) The relatively slow flow movement (in terms of surge wave travel time, in which $T_a \gg 100$) provides a sufficiently long duration for the friction to dissipate a large quantity of heat energy during the oscillating period.

v) The continuous whole body water movement increases substantially the potential effect of friction dissipation, compared to a partial and non-uniform flow movement present in general transient conditions.

The above two events greatly increase the importance of friction energy affecting the unsteady flow phenomena. Consequently, the unsteadiness of the friction factor due to the change of flow regime (v) or due to the distortion of shear stress distribution (dv/dt) becomes much more relevant.

A.6.4 Conclusions of U-tube oscillating flow

Analysis has been carried out on several cases of U-tube systems, and conclusions made as follows:

i) The discrete expression of Newton's Second Law of Motion is directly valid in the mass oscillating system, as the whole water mass moves as a rigid body. Elastic properties, which govern the transient phenomenon in a pipe system do not exist here.

ii) The decay of U-tube flow movement is entirely governed by the friction dissipation (ED_z), which is in contrast to the mechanical transfer phenomenon as observed in a pipe system.

iii) Energy losses other than friction dissipation have not been detected during oscillating flow reversal. This is not true in a pipe system as the incidence of flow separation has been reported in laboratory works (Zielke, 1966).

iv) The real form of shear stress distortion in a pipe flow has not yet been observed in physical tests. On the other hand, the distortion of shear stress distribution was actually measured from a U-tube oscillation test more than half a century ago (Richardson and Tyler 1929).

Therefore, it can be concluded that rapid changes of flow regime (v) and flow acceleration ($+dv/dt$ or $-dv/dt$) may have a large friction dissipation in a mass oscillation, and consequently affect significantly the decay of oscillating flow movement.

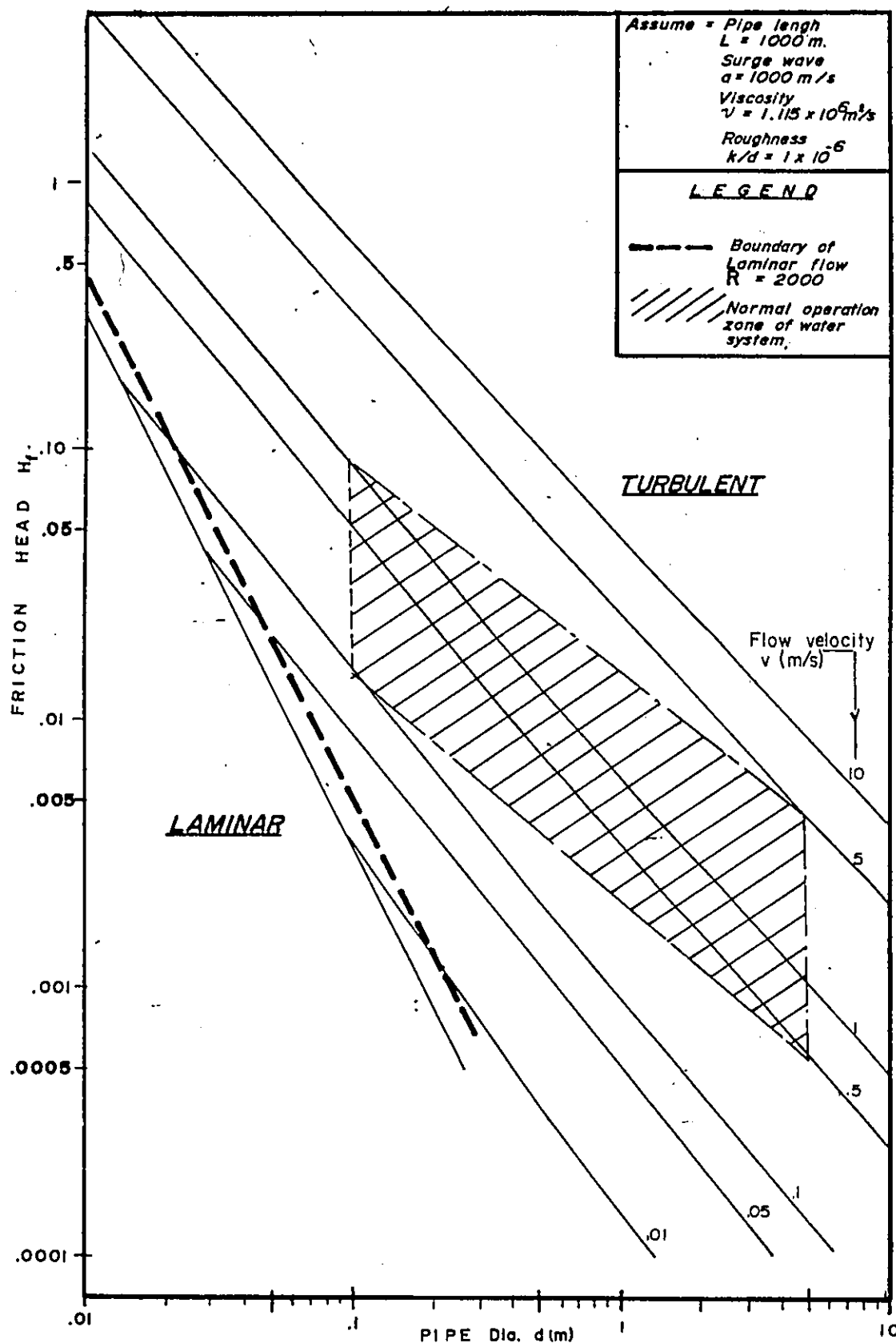
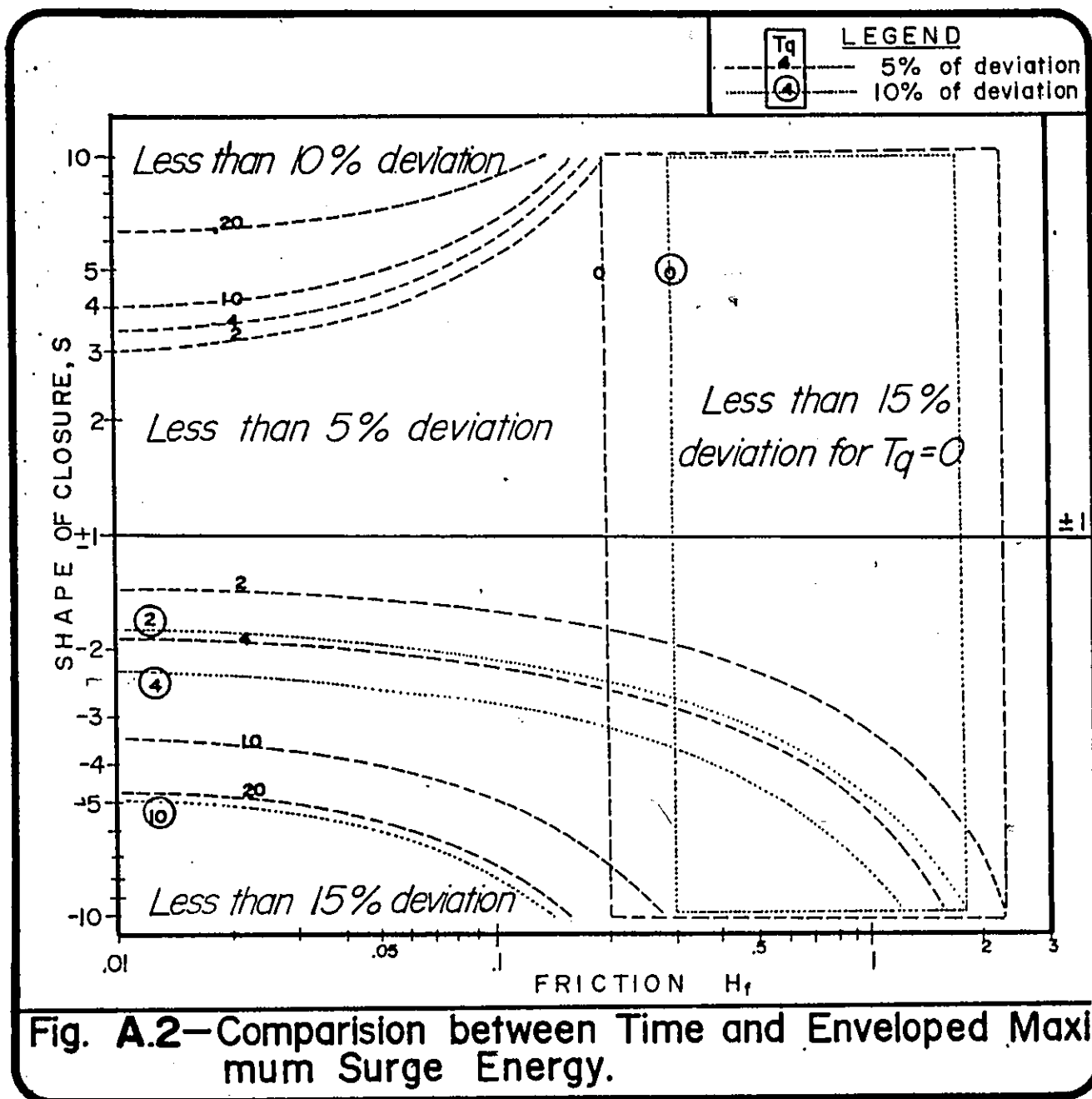
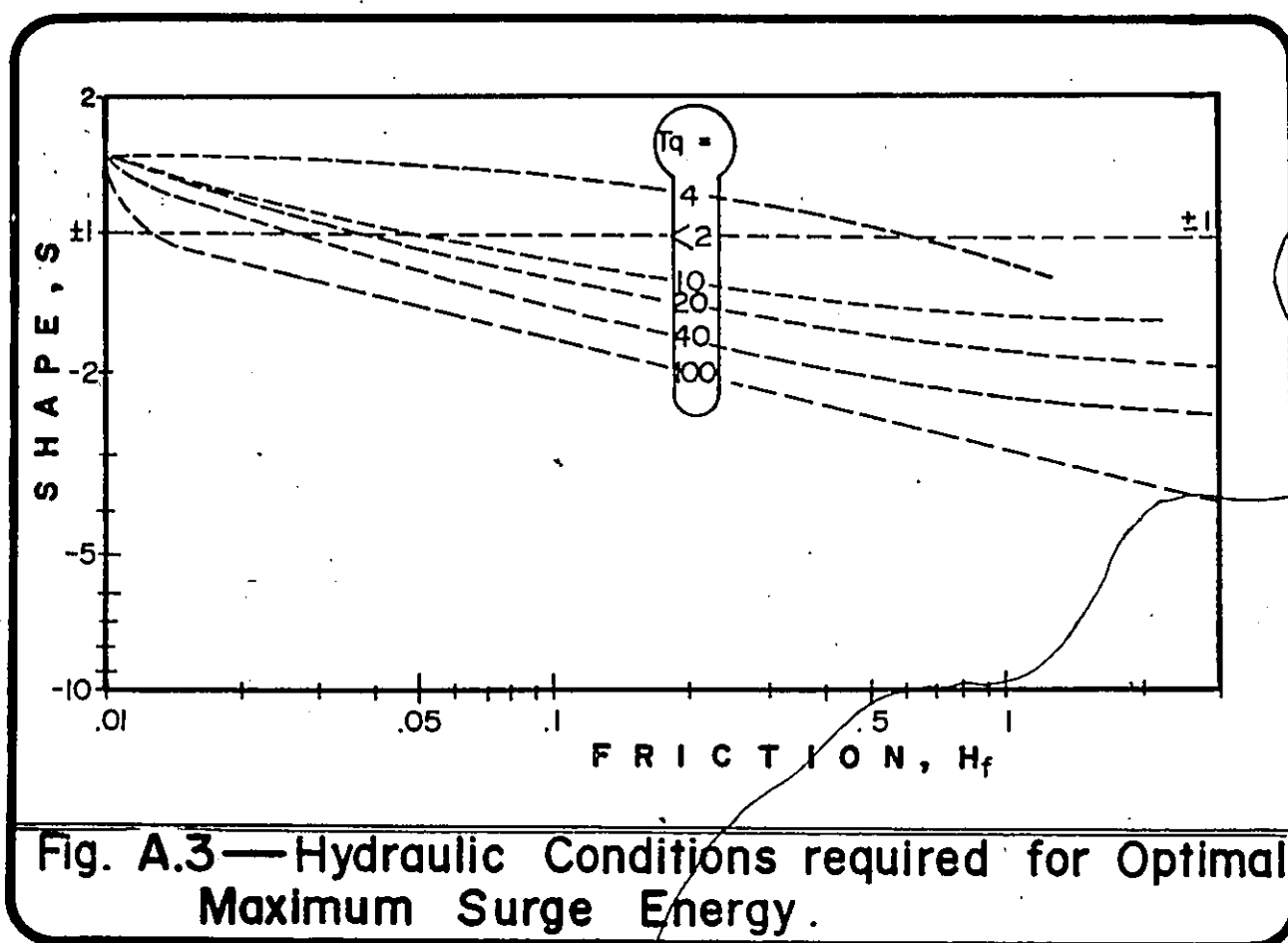


FIG. A.1 — FRICTION HEAD IN A SMOOTH PIPE.





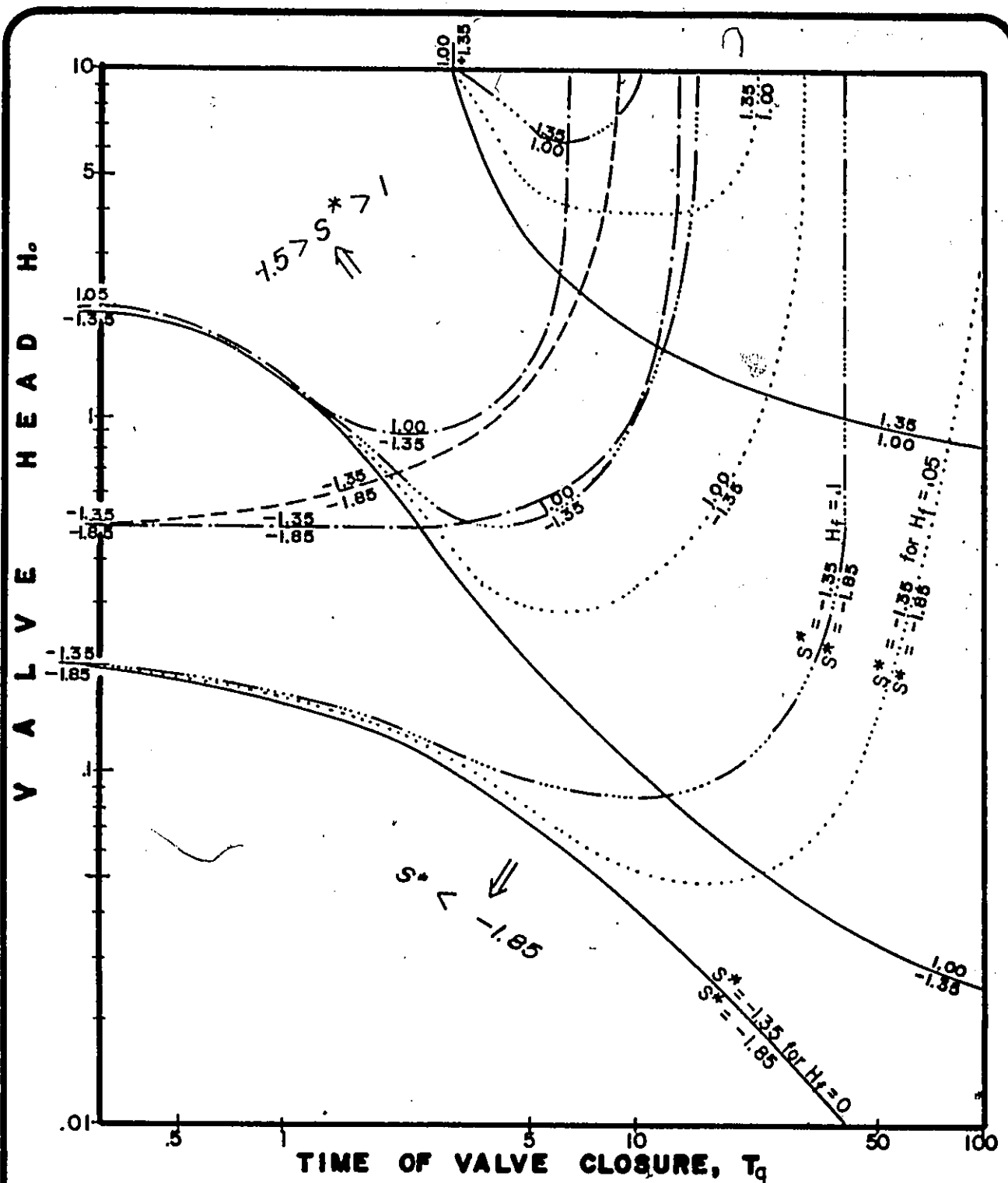
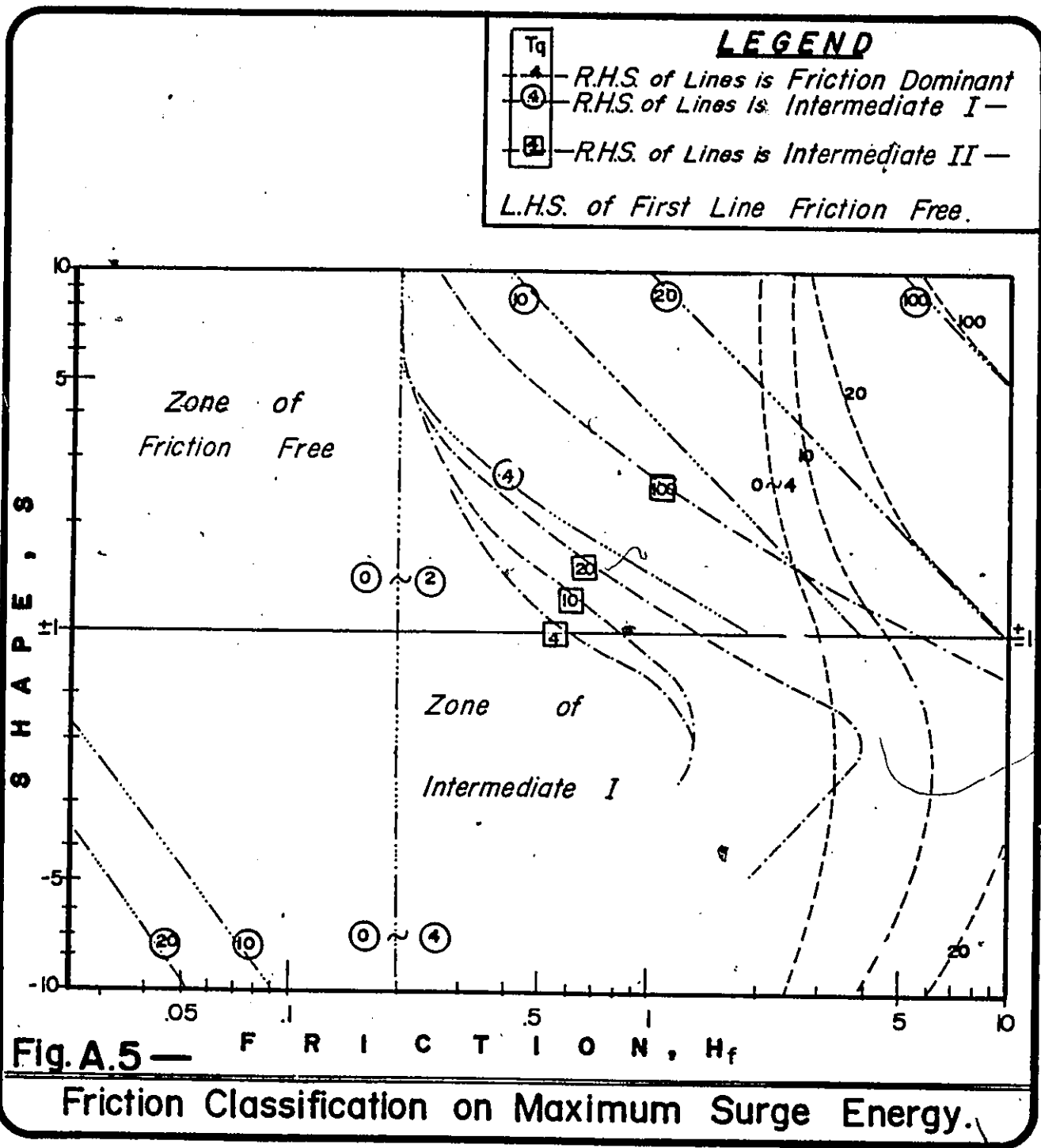


Fig. A4—Conditions Required for Optimal Valve Closure. (S^*)

LEGEND

H_f	1
	.5
	.4
	.05
	0



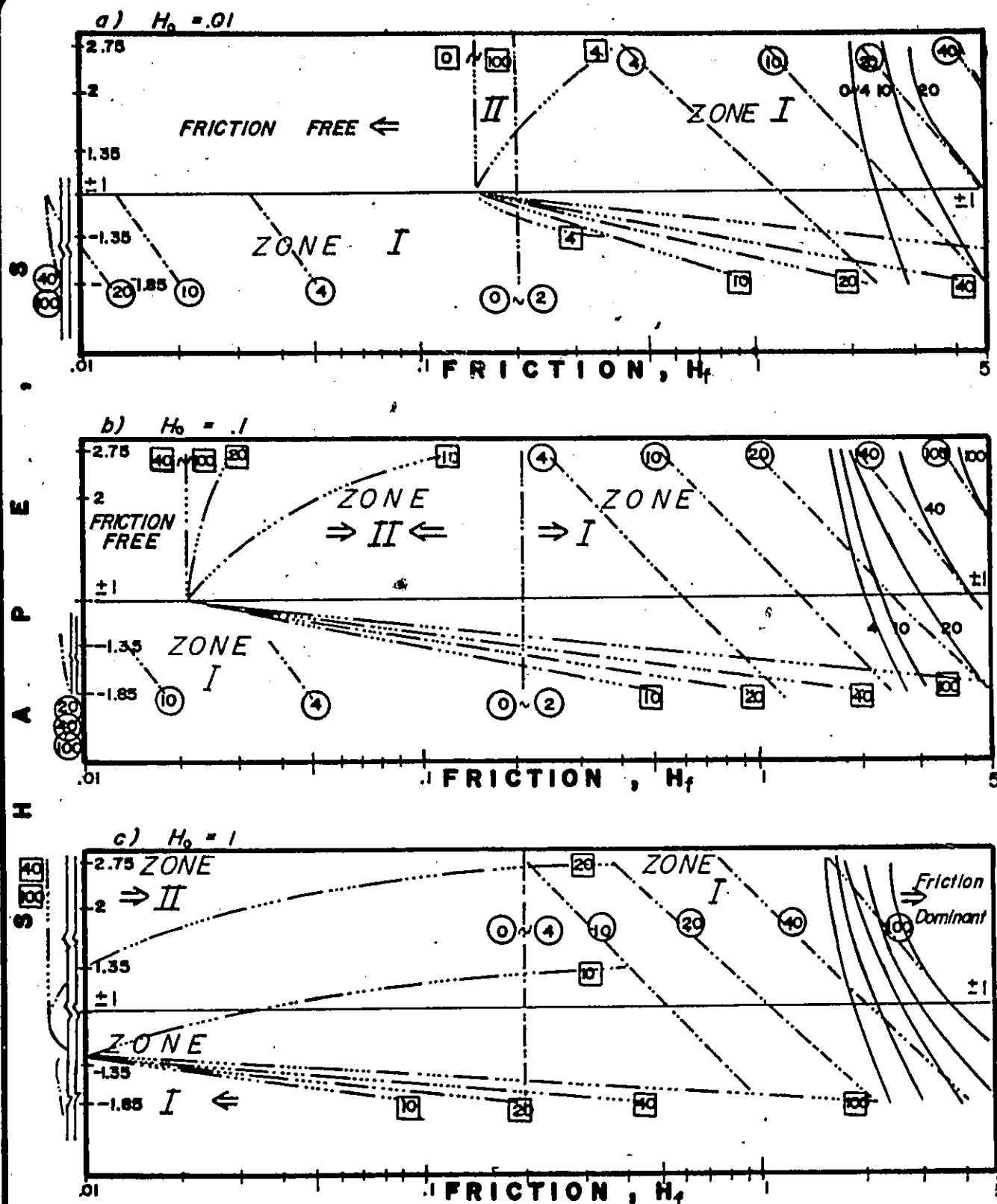


Fig. A.6—Friction Classification Max. Surge Energy (Valve).

LEGEND
 R.H.S. of Line is Friction Dominant
 R.H.S. of Line is Intermediate I
 R.H.S. of Line is Intermediate II
 L.H.S. of First Line is Friction Free.

T_2
 2
 4
 10
 20
 40
 100

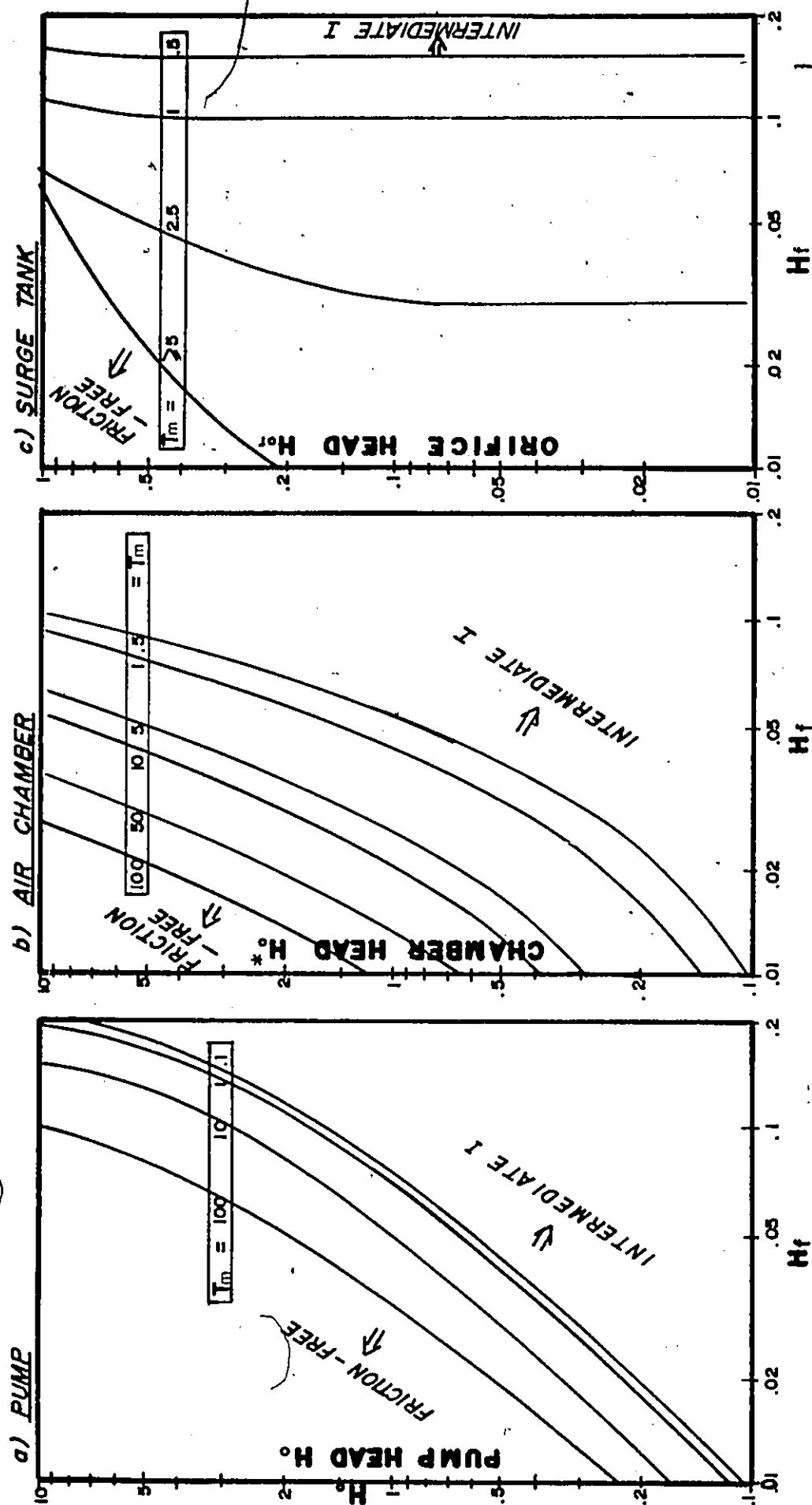


Fig. A.7 — Friction Classification on Max. Surge Energy. (PUMP, CHAMBER & TANK)

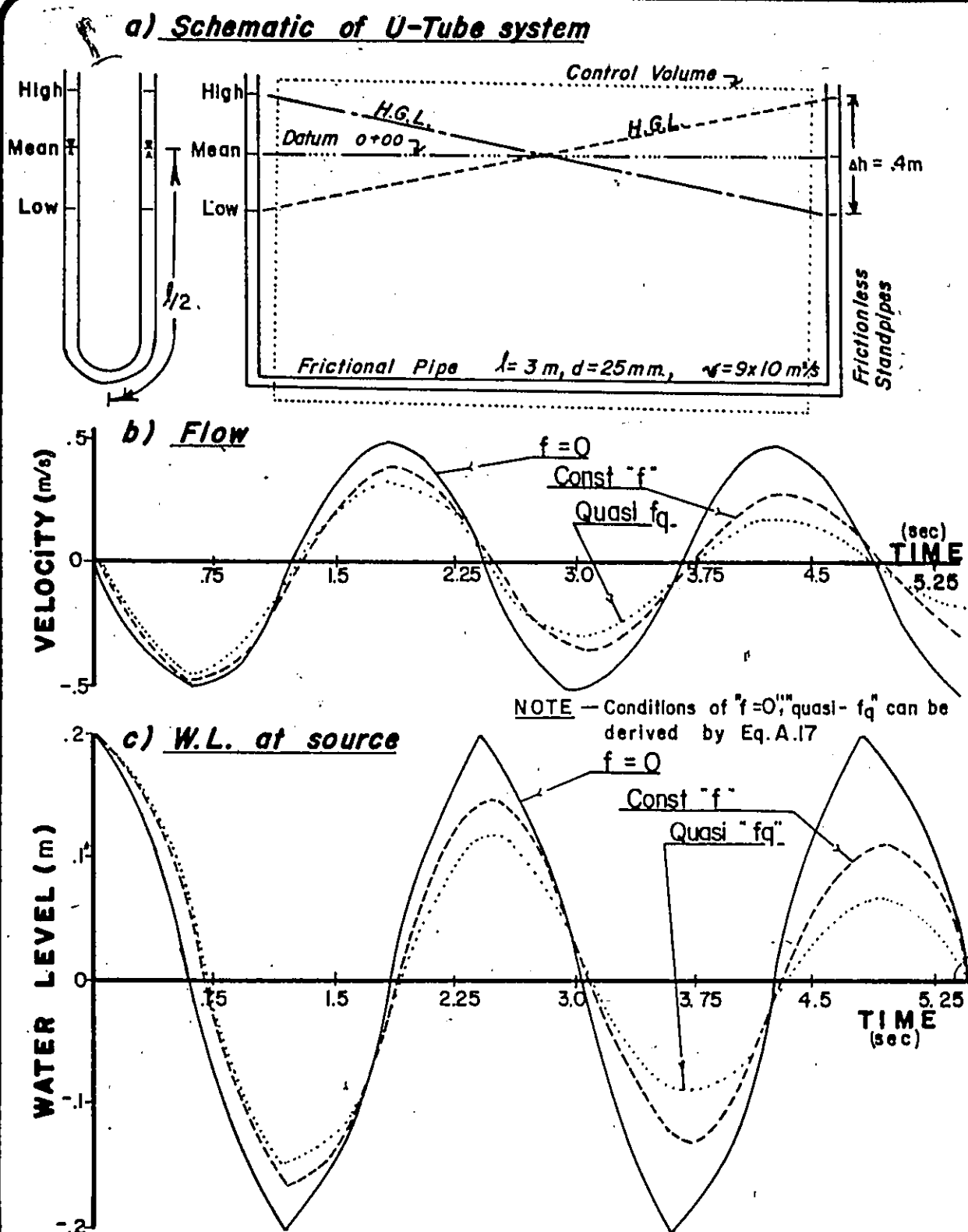


Fig. A.8 — Oscillation in U-tube (an example from Streeter & Wylie, 1981.)

B.0 RIGID WATER COLUMN THEORY IN PIPE FLOW

B.1 Introduction

Analysis in the main text of this thesis concluded that the elastic properties of the pipe wall and water column govern the physical phenomena for a transient pipe flow. However, there has been substantial effort devoted to define the region where the rigid-water column theory (RWCT) can be assumed for general practice. In the literature it appears that a limited success with RWCT had been achieved for application to valve systems with uniform closure, and to simple surge tanks without orifices.

This appendix reviews the historical development of RWCT, identifies the constraints of the current theory, utilizes the concept of energy balance for analysis, introduces a new methodology for RWCT classification, and finally classifies the regions of validity of RWCT for both hypothetical flow and surge control device systems.

The transient properties of uni-directional flow is the main theme of the present research into RWCT in pipe flow motion. Some of the theoretical background related to the rigid flow movement was previously introduced during the development of energy transfer concepts. Several notations and figures from the main text are adopted in this appendix.

B.2 Historical Development of Rigid Water Column Theories

This section describes the theoretical background of rigid and non-rigid water column theories, and summarizes the results of current literature related to this area.

B.2.1 Classical rigid water column theory

At the turn of the century (1915), Johnson derived a method for estimating maximum surge pressure (h_{mx1}) at a valve in a non-frictional system ($H_f=0$) with uniform valve closure

(S=1). Two equations were used by Johnson for the development. The first one is Newton's basic law of motion:

$$h = l/g * dv/dt \quad B.1$$

where, h = pressure head at valve,
 l = length of pipe,
 g = gravitational acceleration, and
 dv/dt is the flow deceleration.

The second equation is the orifice flow loss relationship, which is expressed as:

$$Q = C_d * (2g)^{0.5} * A_d * (h)^{0.5} \quad B.2$$

where, Q = orifice flow,
 C_d = discharge coefficient, assumed to be constant during the valve closure, and
 A_d = valve area opening..

By the combination of these two equations, the well-known classical Rigid Water column Theory (RWCT) was formulated, expressed as:

$$h_{max1}/h_0 = K_1/2 + (K_1 + K_1^2)^{0.5} \quad B.3$$

where, h_{max1} is the maximum surge pressure at valve, occurring at the end of valve closing period;
 $K_1 = (l * v_0 / g * h_0 * t_c)^2$; and
 h_0 = valve head.

B.2.2 Elastic water column theory

For the sake of completeness in assessing the current RWCT, two elastic water column theories are described below.

B.2.2.1 Joukowski theory

In 1898, Joukowski established a theoretical relationship between pressure and velocity change during transient flow

conditions. The 'elasticity' of both the water and the pipe walls is then reflected by the surge wave speed, 'a'. This relation can be expressed as:

$$dh = a/g * dv$$

B.4

where, dh = change in pressure head; and
dv = change of flow velocity.

With an instantaneous flow closure, the pressure head (h) becomes the well-known Joukowski's head ($h_c = av/g$).

B.2.2.2 Allievi's theory of elastic flow system

In 1925, Allievi's 'Elastic Theory' (1902) was translated by Halmos and introduced to the North American engineering community. In addition, two dimensionless quantities were introduced, equivalent to valve head (H_o) and time of valve closure (T_a). These can be written as:

$$\begin{aligned} H_o &= h_o / h_m = g * h_o / a * v_o \\ \text{and} \\ T_a &= t_a / t_c = a * t_a / l \end{aligned}$$

Assuming a uniform valve closure, the maximum pressure rise was derived by Allievi as:

$$H_{max1} = 0.5 (H_o * T_a)^{-2} * (1 + (4 (H_o * T_a)^2 + 1)^{0.5}) \quad B.5$$

These two elastic theories (Eqs. B.4 and B.5) are fundamental to most modern developments in hydraulic transients. They form the background for the subsequent development of the 'graphical method' and 'characteristic method' for the transient simulation in pipe hydraulics adopted in this research.

B.2.3 Current literature related to classical RWCT

Since the publication of Johnson's classical paper on the rigid water column approximation (Eq. B.3), the subject of

rigid water column theory has been and still is popular in transient analysis. Substantial effort has been devoted by numerous researchers and engineers to improve the accuracy of the classical RWCT, to determine the range of its application, and polish the form of its presentation. Some of these works are discussed below. Virtually all the analyses focused on the non-frictional system. Only a scattered and random investigations were undertaken for frictional systems.

i) Quick (1933) --- Allievi's elastic water column theory (Eq. B.5) was confirmed by a series of experimental tests. At the same time, Quick also found that the rigid water column theory (B.3) shows a close comparison with that of the elastic theory for $H_0 < 0.5$. On the other hand, the RWCT may provide only half of what obtained from Allievi's equation.

ii) Parmakian (1963, page 7) --- Based on results of his semi-empirical analysis, Parmakian suggested that the dimensional time (t_a) of valve closure should be greater than $0.003 * \text{pipe length } (l)$, for the use of classical RWCT.

In a normal concrete or steel pipe, the pipe length $l = 1000 \text{ m}$ and the surge wave speed $a = 1000 \text{ m/s}$. Accordingly, the dimensional (t_a) or non-dimensional (T_a) valve closing time required for the adoption of Johnson's RWCT is 3 seconds or 3 units respectively (where $l/a = 1 \text{ sec.}$).

iii) Valentine (1965) --- Based on numerical analysis by graphical methods, Vallentine concluded that the classical RWCT agrees with the Allievi's solution only for values of $h_0/v_0 < 50$ (which is equivalent to a non-dimensional valve head, $H_0 < 0.5$, in a concrete or steel pipe) and only for valve closing times $T_a > 16$. Substantial error would be induced for cases outside this range.

iv) Wood (1973 and 1974) --- With the aid of computer modeling, Wood devoted tremendous effort to establish corrections for the use of the rigid water column theory. In general, the relative error of the RWCT method decreases with the increase in the time of valve closure (T_q).

v) Wylie (1981) --- Wylie suggested that a valve closing time of $T_q = 20$ may be considered a slow stopping time, and therefore the rigid water column theory might be applied.

Conclusions made by these authors are summarized in Fig. B.5, and will be discussed later in light of the new RWCT.

In addition to the above work, considerable analysis and discussions related to this classical RWCT also appeared in many other papers (e.g. Stephenson 1966, Wood and Jones 1973) and text books (e.g. Stephenson 1984 page 106, Watters 1984, page 63).

B.2.4 Discussion of current treatment in RWCT

The general findings and conclusions of these publications can be summarized as follows:

i) Classical RWCT simplifies the procedure of estimating the maximum valve surge pressure (H_{max1}) and has proven to be quite accurate in very low head valve systems.

ii) However the degree of inaccuracy in this treatment increases rapidly with increasing valve head (H_0) and decreasing valve closing time (T_q).

iii) Past research on this subject has been neither complete nor satisfactory.

This last statement can be supported in a number of ways, as will be shown in the next section.

B.3 Constraints Associated with Classical RWCT

This section reviews various constraints related to the classical RWCT which has been adopted since 1915.

B.3.1 Constraints in the scope of application

The classical RWCT can only be applied under very limited conditions as described below:

i) Valve operation in control surge --- This represents only one of the four common surge control devices and presently, the classical RWCT cannot be applied to pumps, surge tanks and air chambers.

ii) Uniform closure ($S=1$) in a valve operation --- Again this represents only one of many common types of valves in general practice. No consideration of RWCT application has ever been given to the others; six among these are: butterfly ($S=-1.85$); ball ($S=-1.35$); square gate (combination of $S=1.35$ and $S=-1.35$); circular gate ($S=1.35$); needle ($S=2$) and accelerated circular gate valves ($S=2.75$).

iii) Low head ($H_0 \ll 0.5$) in a uniform closing valve -- Similarly, this represents only part of the broad range of potential valve heads. Hence, the classical RWCT fails to be applicable in many pipe systems. In particular, while low head conditions were generally applicable the advent of the wide-spread use of plastic pipes has made common high non-dimensional heads (H_0). This substantially widens its practical range (e.g. $0.5 < H_0$) of valve head conditions.

iv) Non-frictional system ($H_f = 0$) --- This condition is rarely approached in real engineering applications. Rather, applications of RWCT in situations of high

friction head are frequently encountered in general practice (e.g. oil pipeline or plastic pipe).

B.3.2 Constraints to correction methods

There are procedures recommended in the current literature for the modification of results estimated from Johnson's (1915) theory, but these tend to be tedious and sometimes confusing. It appears to be much simpler and more straightforward to directly use a nomograph of uniform valve closure in a non-frictional system, instead of applying the simple classical RWCT equation (Eq. B.3) plus complicated corrections, recommended by various authors.

In order to appreciate the complexity of these modifications, works can be referred from the following publications: i) Figs. 6 and 7 in Quick, 1933; ii) Figs. 1-3 in Valentine, 1966; iii) Fig. 1 in Wood 1973 and iv) Fig. 5 in Wood 1974.

B.3.3 Constraints to theoretical background

With the increase of valve head H_0 , the importance of the orifice head-loss relationship (Eq. B.2) begins to decrease. Consequently, the valve flow closing characteristics follows in exactly the same manner as that of a valve area closing. Should the classical RWCT still be applied, Newton's basic equation (Eq. B.1) becomes the only governing function in the RWCT (Eq. B.3).

This implies that based on Johnson's RWCT:

The rigid water column movement approximation is valid for a uniform flow closure which is equivalent to a uniform valve closure ($S=1$) with a high valve head.

This finding contradicts the previous conclusion made in Chapter 5. As shown in Fig. 5.4a, periodic fluctuations with the same magnitude dominate the entire flow closing

period. An upper bound for the valve head must be set for the use of the classical RWCT.

B.3.4 Discussion of RWCT constraints

The constraints discussed above raise a great concern:

It is questionable whether or not the classical rigid water column theory (Eq. B.3), is an appropriate tool for transient analysis.

One of the main tasks in this appendix is to resolve this question.

B.4 Analysis of Rigid Flow Motion in Typical Flow Closures

This section attempts to explain the fundamental background of rigid flow motion theory in a non-frictional system.

The concept of energy balance, characterized by the periodic flow fluctuations at the pipe-end is adopted for the present analysis.

B.4.1 Rigidity determined from flow velocity

B.4.1.1 Rigid water body movement in a closed conduit

For a rigid water column movement, flow rates should be the same at the source and pipe-end, that:

$$Q_s(T) = Q_x(T) \quad \text{B.6}$$

This represents a very strict condition imposed for the rigid water column movement.

Equation B.6 is a valid assumption for some laboratory works, such as U-tube or pulsating mass oscillation. However the assumption of this ideal rigid water column movement is not valid in a transient pipe flow. Some

guidelines for defining the rigid water column region with a realistic allowance of error should be proposed.

B.4.1.2 Transient flows in pipe systems for $T < 1$

It takes one wave travel time for the surge wave to travel from one end of the pipe to the other. Hence at time $T < 1$, the effect of wave reflection is non-existent since the surge wave has not yet arrived at the end of the pipeline. Therefore, the flow leaving the system (Q_x) should be independent of the incoming flow (Q_s) characteristics. This means that:

$$Q_x(T) = 1 > Q_s(T) \quad \text{B.7}$$

where $0 < T < 1$.

B.4.1.3 Flows in pipe systems for $T > 1$

If there is no wave reflection effect at time $T > 1$, the flow rate at the pipe-end, $Q_x(T)$, should be identical to that at the source at the time preceding one wave travel time, $Q_s(T-1)$. This means that the following relationship holds true in terms of magnitude and direction:

$$Q_x(T) = Q_s(T-1) \quad \text{B.8a}$$

Referring to the transient history plotted in Fig. 5.1, it can be seen that a 100 % wave reflection can be induced with an instantaneous flow closure ($T_a = 0$) in a non-frictional system, such as:

$$Q_x(T) = -Q_s(T-1) \quad \text{B.8b}$$

In reality, at a time longer than one wave travel time after the surge initiation ($T > 1$), a transient event with normal flow closure is affected by the reflection of the surge wave in a manner between that described by Eqs. B.8a and B.8b.

B.4.2 Flow deceleration in a pipe system

B.4.2.1 Wave reflection as a criterion

As is well known, the characteristics of periodic flow fluctuation are generally governed by the flow deceleration at source and the wave reflection at the pipe-end. A new relationship of transient properties between the source and pipe-end should be established according to the variation of dV/dT , which is far more complicated than either Eqs. B.8a or B.8b.

During the period of the initial surge wave reflection at the pipe-end (i.e. $1 < T < 3$), results from the numerical model show that the rate of flow change at the pipe-end is twice the inflow at the source at the preceding one wave travel time (i.e. $dQ_{\pi}(T)/dT = 2 dQ_s(T-1)/dT$). This is independent of the shape (S) and time (T_a) of flow closure. In this research, this property, $RQ_{\pi\pi}(T)$ is termed the rate of wave reflection and is defined as:

$$RQ_{\pi\pi}(T) = | dQ_{\pi}(T)/dT / dQ_s(T-1)/dT | \quad B.9$$

Then, the characteristics of first wave reflection is:

$$RQ_{\pi\pi}(T) = 2 \quad B.9a$$

where $1 < T < 3$.

It should be noted that after the period of first wave reflection, the rate of wave reflection ($RQ_{\pi\pi}$) will deviate from 2 and vary as a superposition of the past wave reflection. This means:

$$RQ_{\pi\pi}(T) >< 2, \quad \text{for } 3 < T < T_a \quad B.9b$$

General characteristics of Eq. B.9 can be observed by the example cases shown in part b of Figs. 5.2 - 5.4, in which three typical types of flow decelerations are illustrated.

As shown, during the period of $1 < T < 3$, the relationship expressed in Eq. B.9a is valid for any shape of flow. However, during the period of $3 < T < T_a$, Eq. B.9b becomes:

i) Uniform flow closure ($S = 1$) --- the rate of wave reflection (RQ_{ss}) fluctuates intermittently between zero and two.

ii) Decreasing decelerating closure ($S < -1$) --- $RQ_{ss}(T)$ begins with 2 and then increases steadily with time; and

iii) Increasing decelerating closure ($S > 1$) --- the magnitude of $RQ_{ss}(T)$ decreases gradually from 2 to 1, as time progresses.

The disadvantage of Eq. B.9 is the difficulty in determining the exact value of $RQ_{ss}(3 < T < T_a)$, because the time difference of one wave travel time between two flow decelerations (at the source and pipe-end) is required.

B.4.2.2 Deceleration between flows as a Criterion

An alternative criterion is used in this appendix. A new term elasticity of water column movement (RQ_{ss}) is defined as below:

$$RQ_{ss}(T) = \left| \{dQ_s(T)/dT - dQ_a(T)/dT\} / dQ_s(T)/dT \right| \quad B.10$$

where RQ_{ss} is a good indicator of the rates of flow change at the source and the pipe-end of the system.

In this research, the upper bound of rigid water motion is defined as Eq. B.11. This definition is arbitrary, but reasonable in the practical application. This leads to that:

$$RQ_{ss}(T) < 0.1 \quad B.11$$

where $T_s' < T < T_a$.

Here T_r' is the time required for a transient flow to function as a rigid water column following the initiation of transient flow. The time $T_r' > 3$ should be used, for considering the effect of first wave reflection period.

B.4.3 Rigidity for normal flow closure cases

Typical characteristics of each type of decelerating flows related to pipeline elasticity are discussed here at a screening level as shown in Part b of Figs. 5.2 - 5.4 which are re-inserted at the end of this section.

B.4.3.1 Uniform flow closure ($S = 1$)

Within the whole period of flow closing, the elasticity of flow movement is:

$$RQ_{ss}(T) = 1. \gg 0.1$$

Thus it can be concluded that the water column in a uniform flow closure is not rigid.

B.4.3.2 Decreasing deceleration ($S < -1$)

The elasticity of the water column increases with time as:

$$RQ_{ss}(T) > 2 \gg 0.1$$

This indicates that during the whole period of flow closure, the unsteady flow is governed by the elastic phenomena.

B.4.3.3 Increasing deceleration ($S > 1$)

The elasticity of the water column continues to decrease with time as:

$$d[RQ_{ss}(T)]/dT < 0$$

This indicates that flows with increasing deceleration may reach the state of rigid water column motion if the time of flow closure (T_q) and time of transient (T) are sufficiently long.

B.4.4 Discussion of rigid flow in a typical flow system

The physical phenomena of an unsteady pipe flow is governed by its elastic behavior.

For the purpose of general practice, a guideline (Eq. B.11) has been established in this section, to determine the eligibility of the assumption for the use of rigid water column movement (RWCT) in a uni-directional flow.

At the screening level, it is indicated that for a non-frictional pipe system, the minimum requirement for a rigid water column movement is that the shape of flow closure (S) must be greater than 1, and the time of flow closure (T_q) must be longer than 3.

B.5 Analysis of RWCT Classification in Hypothetical Flows

Further to the screening discussed in the previous section, a systematical classification on RWCT is carried out herein, based on the guideline of Eq. B.11. The existence of friction is included in the present analysis.

B.5.1 Theoretical development of rigid flow movement

B.5.1.1 Rigid motion equation in frictional systems

The effect of friction has been traditionally considered in Newton's basic equation by the incorporation of a friction term:

$$h_{max} = 1/g * dv/dt + f * 1/d * v^2/2g \quad B.12$$

This equation has been regarded as a universal equation for rigid water column movement in a frictional piping system.

The present research determines where Eq. B.12 can be applied without inducing significant error in depicting transients in unsteady pipe flows. Two cases are considered below.

B.5.1.2 Rigid behavior during flow closure (T_x')

Based on the criterion expressed in Eq. B.12, the hydraulic conditions have been classified for the valid application of rigid water column motion, and these are shown in Fig. B.1: The relationship expressed in the figure can be summarized in the following form:

$$T_x' = F(H_x, S, T_q) \quad B.13$$

where T_x' represents the minimum time required for flow motion to become rigid. This equation can also be interpreted in such a way that approximately the same flow energy will leave and enter the piping system as:

$$Q_s(T) = Q_m(T) \quad B.14$$

where $T_x' < T < T_q$.

The relationship expressed in Eq. B.14 is considered to be the most restrictive condition imposed on the approximation of rigid water column movement in a pipe system.

B.5.1.3 Rigid behavior at the end of flow closure (T_x'')

If the flow motion can be considered rigid until the end of the flow closing period, Eq. B.11 should be adjusted as below:

$$RQ_{max}(T) < 0.1 \quad B.15$$

where $T_x'' = T = T_q$. Here T_x'' is the time required to achieve the rigid water column condition at the end of flow closing period.

Accordingly, the relationship between $T_{x''}$ and its governing parameters can be graphically presented in Fig. B.2 as below:

$$T_{x''} = F(H_x, S) \quad \text{B.16}$$

It should be noted that one of the restrictions imposed by Eq. B.11 has been removed, where RWCT is required over part of the flow closing period (i.e. $T_{x'} < T < T_q$). In the application of Eq. B.16, the effect of pipe elasticity can be evaluated towards the end of flow closing period ($T_{x''} = T \approx T_q$).

Since the maximum surge energy (E_{max1}) occurs at the end of the closing period for the increasing deceleration flow condition, the zone of rigid water column motion (shown in Fig. B.2) defines the region where Newton's equation can be used to estimate the maximum surge energy. As shown in the figure, Newton's basic equation becomes difficult to apply for $S < 2$. This is due to the increasing periodicity of alternate surge fluctuations, during the whole period of flow closure.

It is noted that results expressed in Fig. B.1 should be kept as a test of elastic behavior if the transient history during the flow closing period is of concern.

B.5.2 Characteristics of the rigid water column region

B.5.2.1 RWCT in non-frictional systems

The elastic behavior of transient pipe flows can be most conveniently analyzed in a non-frictional system ($H_x=0$). Fig. B.1 shows the elasticity of the flow motion during the closing period. The relationship of Fig. B.1 can be simplified, and presented in Fig. B.2, in which the flow motion becomes rigid towards the end of flow closing period. The Newton's basic equation of motion (Eq. B.1) can be used as an approximation in the region of RWCT, identified in Figs. B.1 and B.2.

i) Shape of flow closure (S) --- The source (Q_s) and the pipe-end (Q_x) outflows at $S = 2$ have the most frequent colinearity in comparison with other shapes of flow closure (reference in Fig. 5.5). Consequently, it requires the least time for the flow to be closed as a rigid body.

It takes a progressively longer time to approach rigid water column motion as the shape factor departs from $S = 2$. In particular, it takes a much longer time to reach the state of rigid water column for $1 < S < 2$. For example, at $S = 1.5$, the time required to achieve rigid flow motion exceeds $T_x' > 100$. This increase of T_x' is due to the existence of alternate periodic fluctuations, which create considerable elasticity in the pipe system.

For the case of an increasing deceleration with a shape factor (S) greater than 2, Newton's basic equation (Eq. B.1) generally overestimates the maximum surge energy. On the other hand, with the shape factor (S) of flow closure less than 2, the basic equation (Eq. B.1) underestimates maximum surge energy (E_{mx1}).

ii) Time of flow closure (T_a) --- Newton's basic equation (Eq. B.1) should be applied only after the end of the first wave reflection period, i.e. $3 < T_x' < T < T_a$. Prior to that, the effect of the first wave reflection dominates the transients in the system (i.e. $RQ_{xx} = 2$).

Generally, the longer the time for the flow to close (T_a), the faster the flow becomes a rigid water column (T_x'). For example, in an increasing deceleration flow with $T_a > 40$, the elasticity of the piping system almost disappears at the end of the first wave reflection period (i.e. $T_x' = 3$, for $2 < S < 10$).

iii) Uniform flow closure --- For a flow with uniform closure ($S = 1$), only half the maximum surge pressure

would be obtained by the basic flow motion equation. The transient energy estimated by Eq. B.1 is constant throughout the whole period of flow closure while the actual transient oscillates between H_{mx1} and zero, symmetrical with the axis determined from Eq. B.1.

iv) Decreasing deceleration flow closure --- The rigid flow motion equation (Eq. B.1) is not applicable in any part of decreasing deceleration ($S < -1$) flow conditions. This finding should not be a surprise as Newton's basic equation would give the maximum value of surge pressure at the first instance of such a flow motion which could only be found with an instantaneous flow closure, i.e. $H_m(T = 0) = H_{mx1}$.

B.5.2.2 RWCT in frictional systems

The regions of RWCT in frictional systems are also identified in Figs. B.1 and B.2. Eq. B.12 is the updated equation for rigid flow movement in a frictional system.

i) Flow closure with $S > 2$ --- In the absence of alternate periodic fluctuations, the increase of friction (H_f) separates the source (Q_s) and the pipe-end (Q_m) flows further apart, during the flow closing period. Consequently, this results in a longer time (T_x') in arriving at an equilibrium state.

The effect of friction on transient behavior was previously shown in the transient history (as shown in Fig. 6.4), which results in an increase of frictional transfer (ET_f) exceeding its steady friction energy (E_f). Figs. B.1 and B.2 summarize effects of friction on T_x' and T_x'' respectively.

ii) Flow closure with $1 < S < 2$ --- In the presence of alternate periodic fluctuations, the existence of friction (H_f) tends to damp out these fluctuations, resulting a shorter time (T_x') in achieving the rigid flow motion.

As shown in Figs. B.1 and B.2, the time (T_{Σ}') required to achieve the equivalent to a rigid water column movement will be greatly shortened with increasing friction head (up to $H_{\Sigma} = 0.5$).

iii) Flow closure with $S < -1$ --- One of the important results in the analysis of frictional systems is that the rigid flow movement can be applied to flow closure problems with decreasing deceleration if the time of flow closure is sufficiently long and the friction head is adequate.

For example, the rigid flow motion can be extended to decreasing deceleration flows, for $0.5 < H_{\Sigma} < 1$ and $20 < T_{\Sigma} < 100$ (see Figs. B.1 and B.2).

iv) Evaluation of friction effects --- The function of the friction acting as a damper can be seen better in the example of a uniform flow closure with various friction heads in the system. Fig. B.3 traces the history of flows at both source and pipe-end. Flow closing properties of $S = 1$ and $T_{\Sigma} = 20$ are selected with $H_{\Sigma} = 0, 0.1, 0.5$, and 2.5 being utilized.

As shown in the figure, severe surge fluctuations still exist in a low frictional system (e.g. $H_{\Sigma} = 0.1$). However, a collinear relationship can be observed between the incoming (at source) and outgoing (at pipe-end) flows for $H_{\Sigma} = 0.5$, long prior to the end of the flow closure (i.e. $T_{\Sigma}' \gg T_{\Sigma}$). In this event, the periodic surge fluctuations have been eliminated.

Conversely, an excessively high friction head may prevent the unsteady flow from reaching its equilibrium state. For example, in the case of $H_{\Sigma} = 2.5$, the above event of flow closing fails to achieve conditions applicable to the RWCT, as the outgoing flow at pipe-end is still far greater than that of the

incoming, even though the surge fluctuations have been totally removed.

Results of the present analysis suggest that the most efficient friction head to reach the rigid water column movement is $H_f = 0.5$, for $S < 2$.

B.5.3 Discussion of RWCT classification

RWCT has been classified on categorized flow movement, based on the periodicity of transient flow fluctuation (Eq. B.11). This has covered both non-frictional and frictional systems.

B.6 Region of new RWCT on maximum surge energy

This section attempts to derive a new method for the RWCT classification, with emphasis on the maximum surge energy.

B.6.1 General purpose

In general practice, the maximum values of transient energy (E_{mx}) usually cause the greatest concern rather than the transient history during the flow closing period. In such a circumstance, the establishment of RWCT should fulfill one principle:

"The new RWCT to be developed in this research should act as a tool to investigate the effect of elasticity on the maximum transient energy (E_{mx}), which is expressed in a form of pressure envelope along the pipeline."

The traditional RWCT (Eq. B.3) was originally developed as a tool for application on uniform valve closure in a non-frictional piping system. In reality, for $S > 2$, the use of rigid water motion equation (Eq. B.12) inside the RWCT region in Fig. B.2 serves the similar purpose for the estimation of elasticity on maximum surge energy (E_{mx1}). In this case, the maximum surge energy occurs at the same time of the flow closure (i.e. $T_{mx1} = T_d$).

As discussed earlier, a severe constraint to general application of RWCT was noted. This can also be observed in the small RWCT region applicable in flows with $S < 1$, as shown in Fig. B.2.

A new approach to classify the RWCT region for a more general application is established as follows.

B.6.2 Relevant parameter governing RWCT on E_{mx1}

B.6.2.1 Relevance of RWCT extension to $S < -1$

A statement was made previously that the RWCT (T_x') can never be applied in non-frictional flows with uniform or decreasing deceleration, due to the existence of periodic surge fluctuations. A question can also be raised as to:

"whether or not the periodic surge fluctuations would affect the magnitude of maximum surge energy for flows with $S < 1$."

In order to have a better insight into this question, an example is selected for the discussion below.

B.6.2.2 Dimensional reasoning for new RWCT interpretation

Assume a decreasing deceleration flow with $H_x = 0$, $S = -2$ and $T_a = 20$. The maximum surge energy, E_{mx1} , is found to be 0.05 from Fig. 5.6.

If the surge wave speed 'a' is increased 5 times (perfectly possible with a change of pipe material), the new time of flow closure T_a , is then 100, which has the new dimensionless maximum surge energy, $E_{mx1} = 0.01$ (Fig. 5.6). The absolute transient energy [$e_x = \gamma * (a*v/g) * A * \frac{1}{2}$] is now 5 times larger with the new 'a'. As such, the dimensional maximum surge energy ($e_{mx1} = E_{mx1} * e_x$) would still be identical for both types of pipe, independent to the change of surge wave speed. The same result can also be

found for tests with other increased surge wave speeds in the above piping system.

From dimensional reasoning, the physical phenomenon governing the maximum surge energy induced by flow closure should be the rigid water column movement if the effect of surge wave speed, 'a' disappears.

Therefore the physical case of flow closing with decreasing deceleration may also be governed by the rigid water column movement for the maximum surge energy computation. This conclusion could also be proven true for flows with uniform closure by the similar method of testing.

B.6.2.3 Energy Balance Concept for new RWCT interpretation

For an explanation of the above conclusion from the perspective of Energy Balance Concept, Fig. 5.2a and 5.3a have been created. For flows with uniform or decreasing deceleration, the maximum surge energy occurs within the first round trip wave travel time (i.e. $0 < T < T_{mx1} < 2$). Within $0 < T < 1$, the flow outflux is constant and equal to the steady flow rate. During $1 < T < 2$, the rate of flow outflux at the pipe-end is twice that of the flow influx at the source at one wave time earlier [i.e. $Q_e(T) = 2 Q_s(T-1)$].

In both cases, the transient behavior is independent of the periodic surge fluctuation effects, in which the surge wave speed 'a' is considered as a relevant parameter to govern the elasticity on the maximum surge energy.

B.6.3 New RWCT region in maximum surge energy application

If a rigid water column movement dictates the determination of maximum surge energy, the maximum pressure envelope would be linear along the pipeline, and the dimensionless surge energy would equal half the surge pressure at source, i.e. $E_{mx1} = 0.5 H_{mx1}$. In a normal piping system, the maximum pressure envelope would be of a somewhat parabolic shape, depending on the degree of pipeline elasticity in the

system. In order ~~to~~ assess this piping elasticity, a new term elasticity of maximum surge energy (E_y) is introduced below:

$$E_y = E_{mx1} / (0.5 H_{mx1}) - 1 \quad B.17$$

where E_{mx1} and H_{mx1} can be obtained directly from the outputs of the mathematical model.

The elasticity term $E_y = 0$ for the disappearance of the surge wave reflection effect on the maximum surge energy, while $E_y = 1$ for an instant flow closure condition.

Based on this new guideline (Eq. B.17), the time (T_x) required to obtain RWCT based on the maximum surge energy is redefined for the maximum surge energy, and this was shown in Fig. B.4 as expressed in the following equation:

$$T_x = F(H_x, S) \quad B.18$$

In addition to conditions required in Eq. B.15, another restriction on this new RWCT is further relaxed since:

The disappearance of system elastic effects on the maximum surge energy (E_{mx1}) does not necessary apply equally to the transient history after the time occurrence of maximum surge energy (T_{mx1}).

For example, the maximum surge energy induced by a uniform flow closure would not be affected by the system's elasticity, for $H_x = 0$ and $T_q > 2$. This finding does not contradict the fact that the transients during the whole uniform flow closing period are governed by the periodic surge fluctuations.

B.6.4 Properties of new RWCT region

The new approach to RWCT by maximum surge energy estimation has the following characteristics:

i) The analysis of pipeline elastic behavior can be extended to cover a large portion of flows with uniform and decreasing deceleration. This has been proven true by both the dimensional reasoning and the energy balance concept. As shown in Fig. B.4, the relationship can be approximated in a broad sense as:

$$E_y(+S) = E_y(-S) \quad \text{B.19}$$

ii) The effect of periodic surge fluctuations is reduced to the new RWCT in flow with increasing deceleration. This can be observed by the comparison between Fig. B.4 and B.2. At $S > 1.7$, the new RWCT region is slightly smaller at $H_s < 0.5$. However, the use of new RWCT has extensively been enlarged for all frictional conditions.

iii) The procedure for estimating the pipeline elasticity (E_y) on maximum surge energy, is simple and straight-forward. It can be computed directly from the maximum surge pressure at source (H_{mx1}) and the maximum surge energy in line (i.e. E_{mx1} , by integrating the maximum surge pressure envelope) which are or can be summarized in a numerical model output. No separate computation is required.

B.6.5 Discussion of new RWCT in pipe transients

The new technique of adopting the shape of maximum surge pressure envelope as a governing parameter is introduced in this research, for the zoning of RWCT on the estimation of maximum surge energy (E_{mx1}). The use of this new criterion greatly extends the effective range of RWCT application, and simplifies the procedures of estimating maximum transient energy.

The effectiveness of this new RWCT has been demonstrated in this section for the wide range hypothetical flows, and this will be further demonstrated below in the construction of design charts for various surge control measures.

B.7 Practical application

In the condition of rigid water column movement, the procedures required for determining transient behavior in a surge control system can be greatly simplified. For example, the surge tank installation and valve operation in many hydraulic conditions can be simplified as follows.

B.7.1 New RWCT in non-frictional and uniform valve closure

The new RWCT region is first applied in a frictionless system with uniform valve closure (solid lines in Fig. B.5). Results of the classical RWCT (Eq. B.3) and the corrections recommended by current researchers (dashed lines in Fig. B.5) are also presented for the comparison purpose.

B.7.1.1 New RWCT application

As shown in the Figure B.5, the rigidity of maximum surge energy increases with the valve head (H_0) and the time (T_a) of valve closure. The time (T_x) required to produce a linear maximum surge pressure envelope (with error margin 10 %) varies from $T_a = 18$ at $H_0 = 0.01$ to $T_a = 2$ at $H_0 = 10$.

In the former case, the equivalent shape (S) of flow closure is high as $S \gg 2$ for $H_0 = 0.01$. On the other hand, $S = 1$ for $H_0 = 10$ in the latter case.

B.7.1.2 Classical RWCT application

For valve head (H_0) below 0.1, the equivalent shape of flow closure (S) is increasing with the decrease of valve head (H_0), with $S > 2$. In this case, both new and classical RWCT regions are virtually the same (see $H_0 < 0.1$ in Fig. B.5).

Conversely, the equivalent shape of flow closure approaches to $S = 1$, as the valve head increases from $H_0 > 0.1$. In this case, the rigid water body motion is difficult to

achieve during the flow closing period. Consequently, the classical RWCT region is diminishing with the increase of H_0 .

At $H_0 = 0.5$, the time required to reach the classical RWCT exceeds 100, with error margin 10 % (Fig. B.5).

A.7.1.3 Current correction factor

As shown in Fig. B.5, Quick (1933) and Parmakian (1963) significantly overestimated the classical RWCT region, while Vallentine (1965) and Wylie (1981) did the opposite.

Though substantial effort had been devoted during his research, the scope of Wood's work is actually very limited (1973 and 1974). The range of hydraulic conditions analyzed is: $H_0 > 0.05$ and $T_a < 10$, which is not sufficient for general practice.

B.7.1.4 Discussion of new RWCT in uniform valve closure

It is concluded from discussions in this subsection that the new RWCT is preferred to the classical RWCT for general application. This is due to:

i) Both new and classical RWCT are the same for $H_0 < 0.1$.

ii) For $H_0 > 0.1$, the new RWCT can still accurately predict the linearity of maximum surge pressure profile along the pipeline, while the classical RWCT begins to lose its capability.

It should be noted that this linear H.G.L. of maximum surge energy is usually used as a guideline for the development of the rigid water column concept (Wylie 1981, page 162 and 164).

iii) The New RWCT is much easier to use, as all the data (H_{max1} and E_{max1}) can be directly obtained from numerical applications.

iv) The classical RWCT can not be applied in other surge control devices, other than a uniform valve.

The new RWCT is adopted in this research for the following practical applications.

B.7.2 New RWCT in common types of valves

The region of the new RWCT has been established for all seven common types of valves, with the valve shape factors varying from -1.85 to 2.75 (see Fig. B6). Friction head adopted herein varies in a wide range ($0 < H_f < 5$).

As shown in the figure, the rigidity (E_y) of the pipe system decreases with higher friction (H_f) and a higher valve shape factor (S). Conversely, it increases with the valve head (H_0) and the time of valve closure (T_q).

B.7.3 New RWCT in system with pump trip-out

As shown in Fig. B.7a, upon pump trip-out the RWCT condition is valid at low pump head (H_0), high friction head (H_f) or large mechanical time (T_m) conditions. This finding is reasonable since the pump flow drops to zero in a step-wise manner. Consequently, a greater time is required for pump flow to approximate RWCT conditions.

In the case of a step-wise pump flow closure, the maximum surge energy envelope is not linear. However, the shape of this parabolic maximum pressure envelope (E_y) may be retained by varying the surge wave speed 'a', resulting the same dimensional maximum surge energy (e_m). Should this new criterion be adopted for pump flow, the mechanical travel time (T_m) required to achieve RWCT (Fig. B.7a) may be

further reduced to approximately 5 times. This criterion related to the step-wise flow is included in the present appendix, for the simplicity and consistency.

B.7.4 New RWCT in system with surge tank

In contrast to traditional analysis, rigid water column movement in a surge tank system was found to be frequently impractical during the course of this research. Particularly in the condition of orifice installation.

As shown in Fig. B.7b, the RWCT may be applied in conditions of a low orifice (H_{ox}), low friction (H_x) with high mechanical time (T_m).

B.7.5 New RWCT in system with air chamber

Elastic water column movement usually dominates transient characteristics of an air chamber installation. As shown in Fig. B.8, RWCT can only be applied in the condition with high chamber head (H_{ox}) and high mechanical travel time (T_m), but with low friction.

The new RWCT can also be adopted in maximum counter-surge energy for two way air chamber, though its results have not been presented in this appendix. In contrast to the general belief (as suggested by Ruus, 1980), the rigidity may actually increase with the orifice head (H_{ox}) and the ratio of differential orifice (K_{ox}). Very often, the orifice throttling (H_{ox} and K_{ox}) reduces the shape of reverse flow closure from a highly increasing deceleration (i.e. $S \gg 2$).

B.7.6 Discussion of new RWCT in common control devices

The region of inelastic flow phenomenon on the maximum surge energy (E_{mx1}) has been defined in these surge control devices. This is made possible by the development of the new RWCT (based on maximum pressure envelope) in this research.

B.8 Storm Sewer Flow Simulations

The following case may not be directly related to the present surge control systems. It is discussed here partly for its importance in general engineering practice, and partly for its popularity in academic research.

The importance of piping elasticity, characterized by the periodic fluctuations, has been demonstrated from both theoretical and practical points of view, throughout this thesis. Significant errors could be introduced by the assumption of rigid water column motion in hydraulic systems which are actually governed by elastic phenomena.

On the other hand, the overemphasis of the importance of the elastic properties in a closed conduit may lead to a gross error in a closed conduit. In a series of technical publications on the subject of surcharged storm sewer hydraulics (e.g. Wiggert 1972, Cunge and Wegner 1980 and Song et al 1983), hypothetical slots were assumed to accommodate the effect induced by the surge wave travel. For a surge wave speed of 1000 m/s, the width of the small slot has to be in the order of 1 mm. Severe transients were indeed observed in such hypothetical systems from their numerical model simulations (the pressure head can fluctuate more than 50 m. along the system).

The results of such a hypothetical system simulation do not appear to be possible or reasonable. This statement can be strongly supported by the following evidences:

- i) In a 10 km long and 2 m diameter storm interceptor without any manhole between (an abnormal scenario made in this type of interceptor), a linear decrease of sewer flow velocity from 5 m/s to zero in just fifteen minutes is assumed (another conservative assumption in the context of a large sewer network). The hydraulic conditions of this system are:

Dimensional --- $t_c = 10$ sec, $h_m = 500$ m, $h_z = 50$ m.

Non-dimensional --- $R = 10^7$, $f = 0.008$

$S = 1$, $T_a = 90$ and $H_z = 0.1$

Based on Fig. 6.7, the maximum surge energy $E_{mx1} = 0.006$, or $H_{mx1} < 0.012$. Therefore, the maximum surge pressure fluctuating in the interceptor system should not exceed 6 m (i.e. $h_{mx1} = 0.012 * 500 = 6$ m).

ii) Assume that a single manhole (with diameter twice that of the sewer, i.e. $d^* = 4$ m) was located at the upstream end of the 10 km interceptor. This size of manhole is a practical design if the inflows are properly intercepted from the upper sewer system. Under an instantaneous flow closure (an unrealistic but conservative assumption), the manhole functions as a simple surge tank, which buffers transient fluctuation with its stored water body. In this assumed case of surge tank, the equivalent values of non-dimensional parameters are:

$H_z = 0.1$; and

$$\begin{aligned} T_m &= [0.5 * (1/g)^{0.5} * (d^*/d)] / t_c & (4.14) \\ &= [0.5 * (10000/10)^{0.5} * (4/2)] / 10 \\ &= 3.2 \end{aligned}$$

As shown in Fig. B.7, the unsteady flow in this interceptor system with a properly designed manhole becomes a rigid water column movement during the whole transient period.

Based on the design nomograph (Fok, 1980), the maximum surge pressure fluctuating along the system should still be less than 50 m.

In reality, numerous large diameter manholes along the interceptor system could be found in a normal storm sewer system with relatively slow flow variations (in terms of

surge wave travel time). This realistic arrangement between manhole and sewer system suggests that the hypothetical slot simulation needs to be rethought.

Further discussions of this slot simulation have been demonstrated by the work conducted by other researchers.

iii) Modeling error --- In one of his comments on this hypothetical simulation, Papatakis (1984) pointed out that the logic of the entire hypothetical slot simulation is fundamentally wrong. The transients thus induced were simply caused by the instability problem of the modeling.

iv) Air entrainment --- Based on the results of laboratory tests, Hamam and McCorquodale (1982) suggested that the air entrainment in the storm interceptor may contribute the interface instability and consequently rapid rise of pressure in the conduit.

v) Hydraulics in interceptor --- During the discussion of the Song et al's publication, Fok (1984) raised concerns on several basic assumptions made by the authors. These include: a) excessively sharp rise and fall of flow variations in a large interceptor; b) unreasonably small slot on top of the conduit; c) neglect of the effect of transient damping from the existence of manholes; and d) absence of the air entrainment induced by the drop of water.

vi) Elimination of manholes --- In their closing remarks of the publication, the authors admitted that a gross error of overestimation may be induced as a result of the elimination of manholes in the simulation (Song et al 1984).

B.9 Conclusions

B.9.1 Theoretical Background

1) The traditional RWCT (developed by Johnson 1915) is found in this research to be applicable under some constraints. These constraints are related to:

i) its narrow scope in relation to general practice, since it is applicable only to uniform valve closure in low head and non-frictional conditions;

ii) the clumsy methods of subsequent modifications, preventing its effective use; and

iii) errors in the assumed boundary condition, in which the basic flow motion equation was wrongly applied in a uniform flow closure.

2) Newton's second law of motion has been applied as a only analytical tool to interpret phenomena of uni-directional flow closure. In reality, this basic equation (if the Eq. of state is not included) should be regarded only as a tool for simplifying the analysis of transients, and only used in the region of RWCT.

3) Past research on improving the traditional RWCT has failed to achieve its objective. One of the main causes of its failure is due to the non-detection of the periodic surge fluctuations during the flow closing period. To achieve this, the understanding of an Energy Balance Concept developed in this research is essential.

4) The existence of periodic surge fluctuations greatly affects the rigidity of water column movement during the flow closing period. However, their effect on maximum surge energy may be greatly reduced for uniform and decreasing deceleration flows.

5) Friction may act as an effective damper, reducing the surge fluctuations, and consequently speeding up the flow movement to reach the RWCT condition (i.e. $S < 1.7$). Conversely, for flows with $S > 1.7$, friction may delay the time arriving the RWCT state.

B.9.2 Classification of RWCT

The region of RWCT can basically be classified from four different approaches depending on the imposed restrictions.

1) The most restrictive condition of RWCT requires the identical flow entering and leaving the system. This basically requires rigidity of the entire water column in the system, over the whole duration during the flow closing period. This is virtually non-existent in most pipe flow conditions.

2) The second most restrictive condition of RWCT requires the same flow leaving and entering the system only after certain time after the surge initiation. The error margin of the rigidity is in the order of 10 percent. This approach provides a realistic condition for the classification of RWCT.

3) The third condition of RWCT is the rigid water column movement considered to occur toward the end of the flow closing period. This condition can be considered as a simplification of (2) from the above discussion.

4) The fourth condition is the least restrictive. It could be used effectively for the classification of RWCT on the maximum surge energy estimation. The key restriction removed in this approach is that the only requirement of RWCT is the disappearance of elastic effect on maximum surge energy (not of the historical transient). This approach has extended the area of application of RWCT to a large region of uniform and decreasing deceleration flow conditions, which Newton's basic equation fails to achieve.

B.9.3 Practical applications

1) The new RWCT has been effectively applied in common surge control devices for the classification of RWCT region. These includes:

- i) seven common types of valve closure conditions;
- ii) pump trip-out;
- iii) surge tank equipped with orifice; and
- iv) air chamber installation.

2) Current analysis of elastic behavior in a surcharged sewer system tends to lead to an over-reaction and consequently, a gross error.

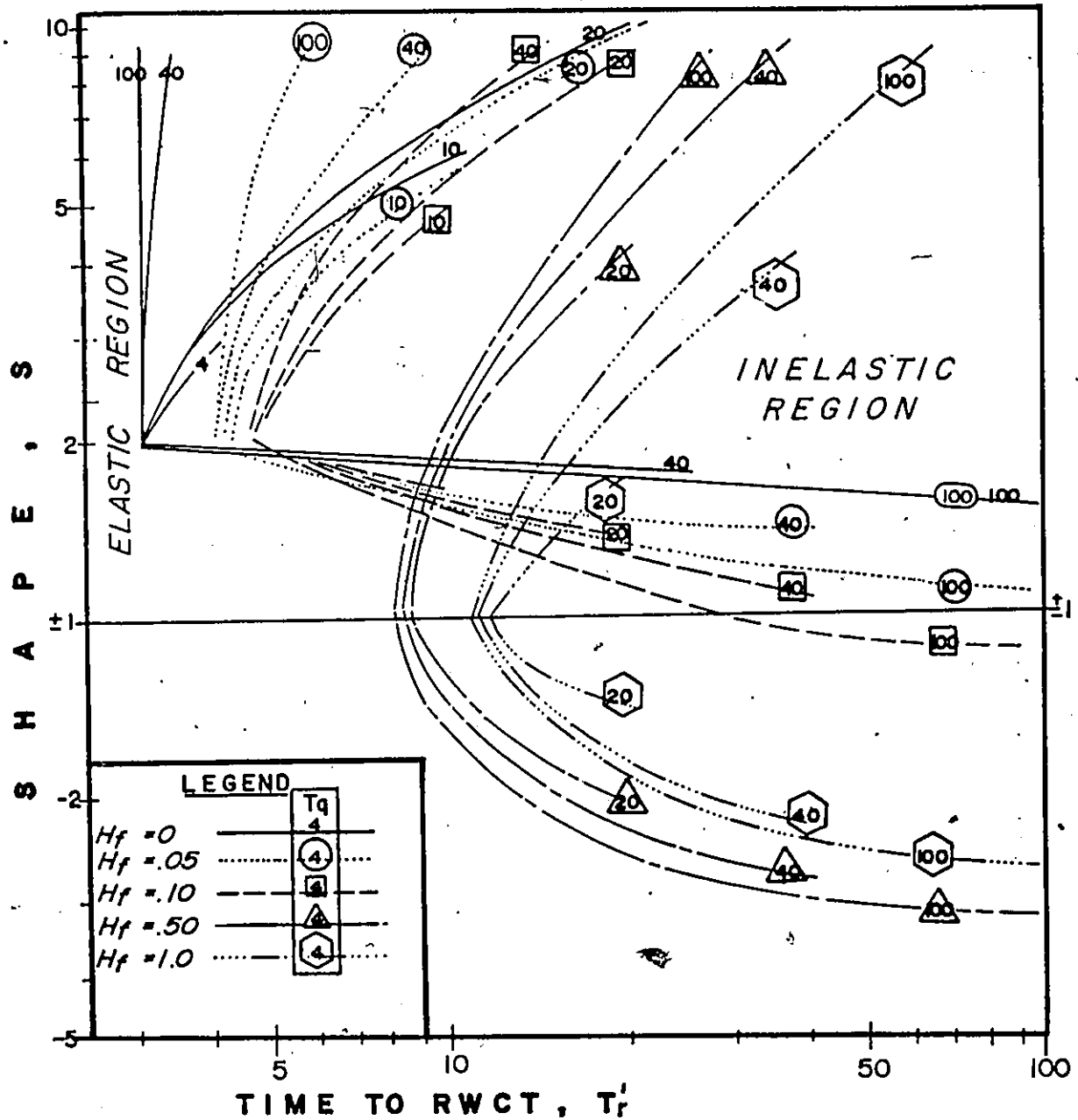


Fig. B.1

Region of RWCT (based on $T_r' < T < T_q$)

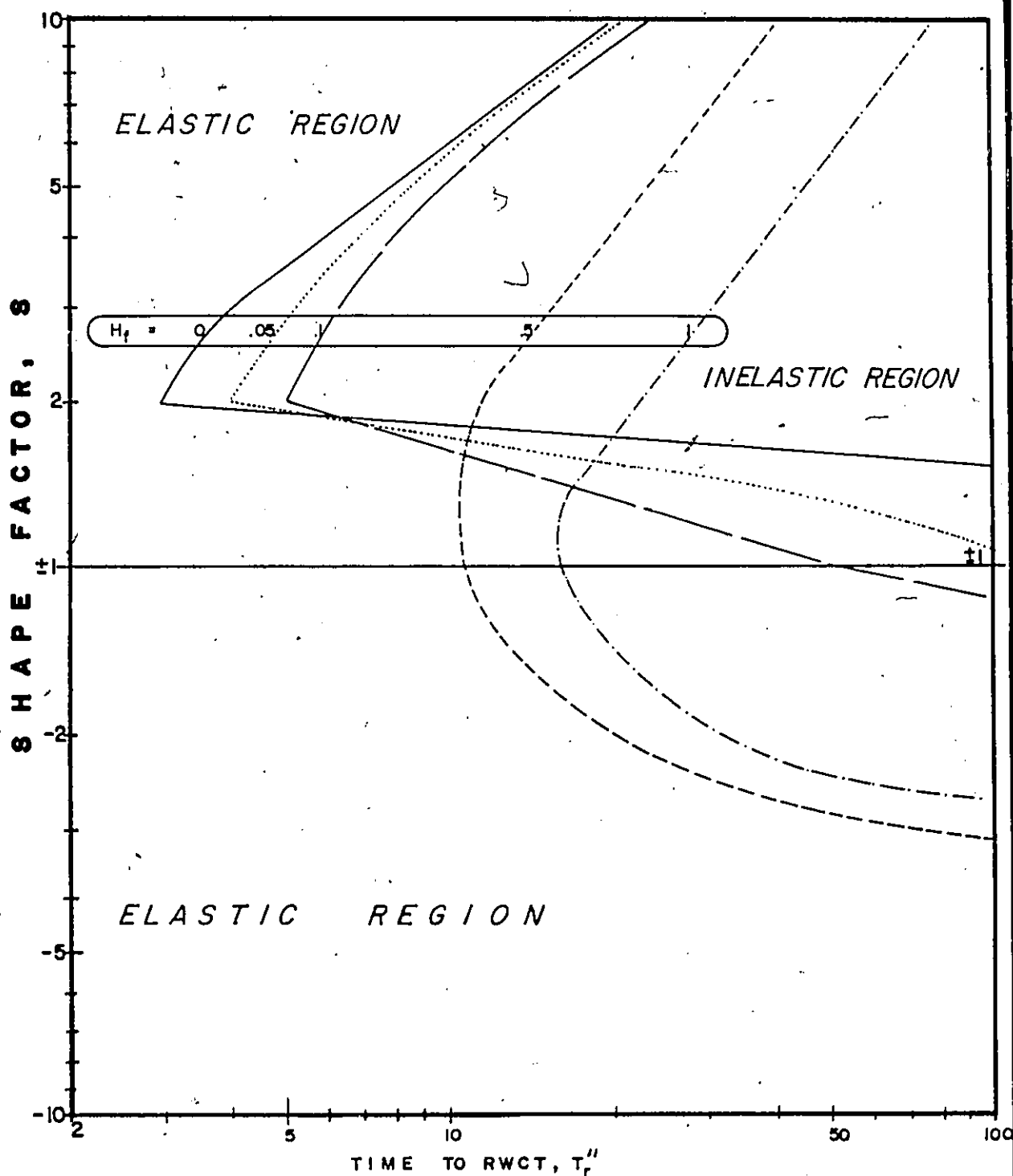
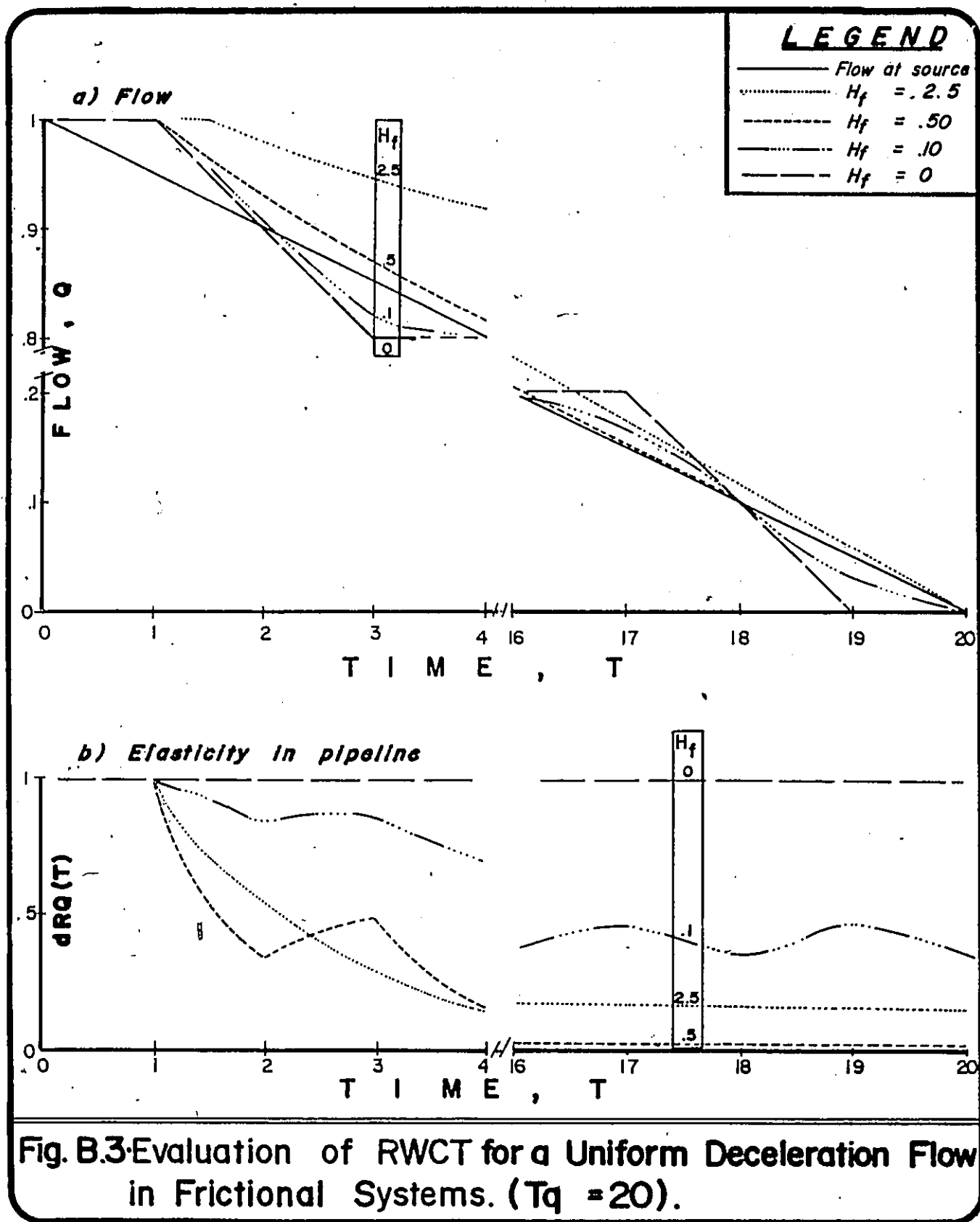
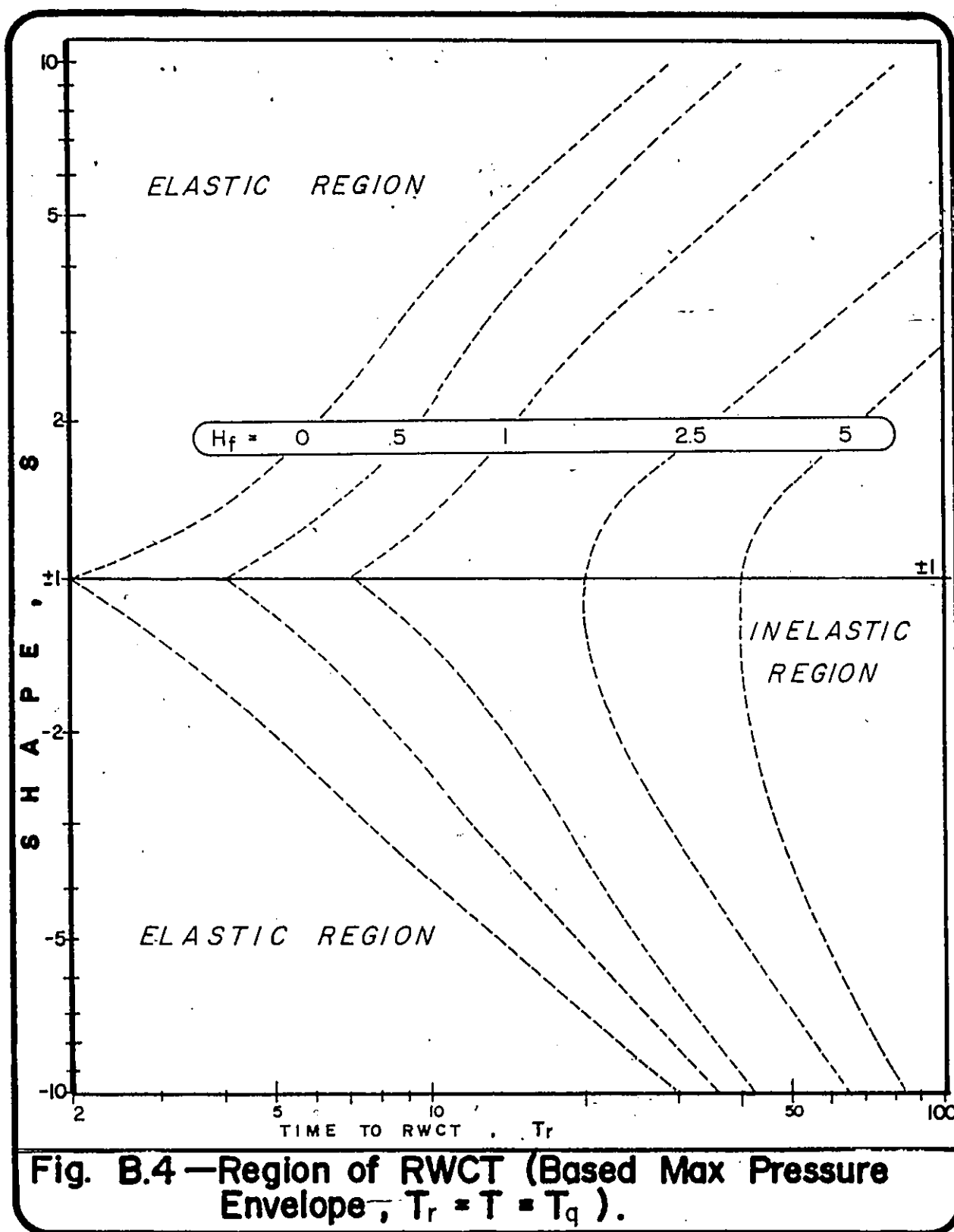


Fig. B.2 — Region of RWCT Based on $T_r'' = T = T_q$.





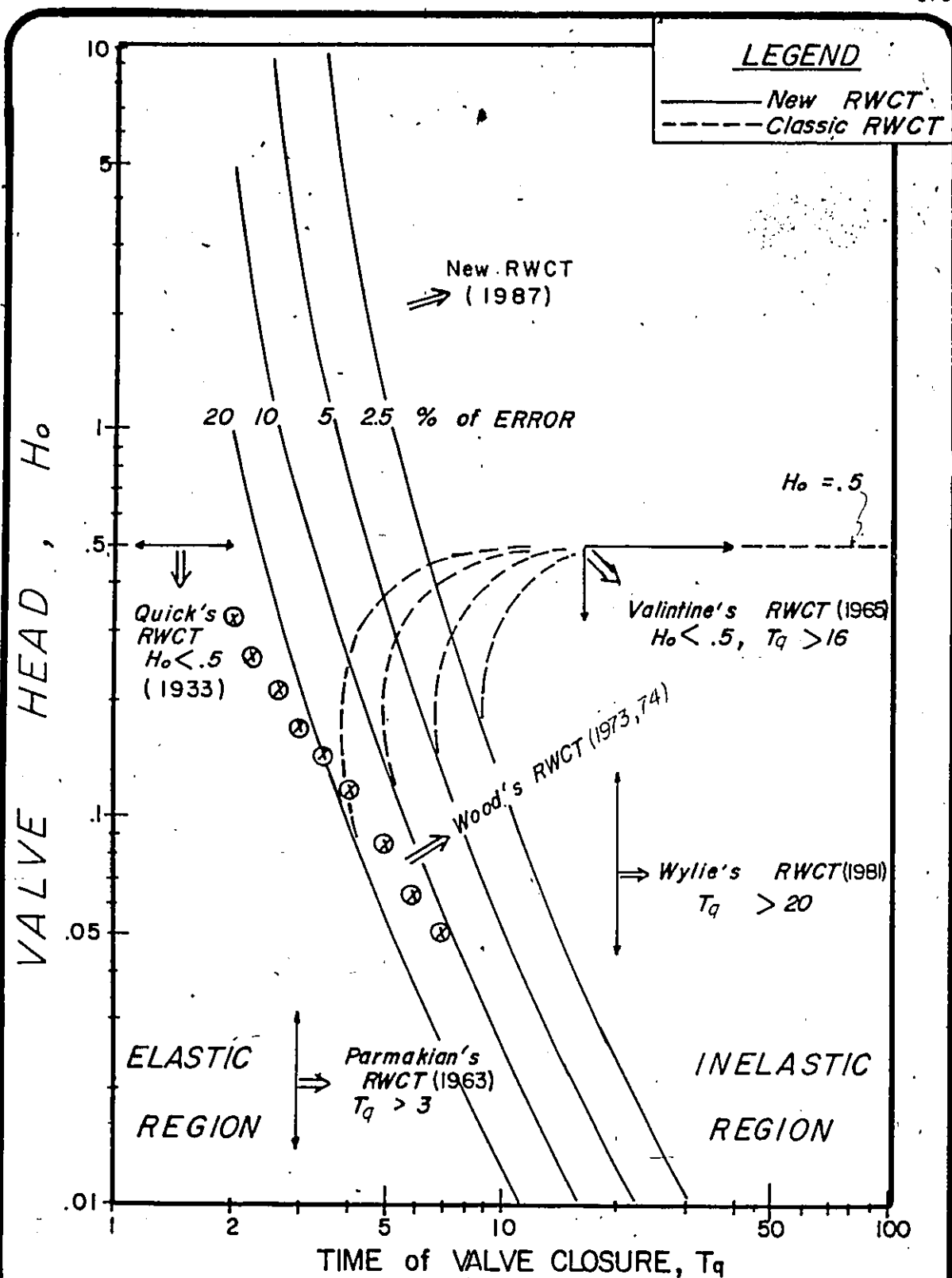
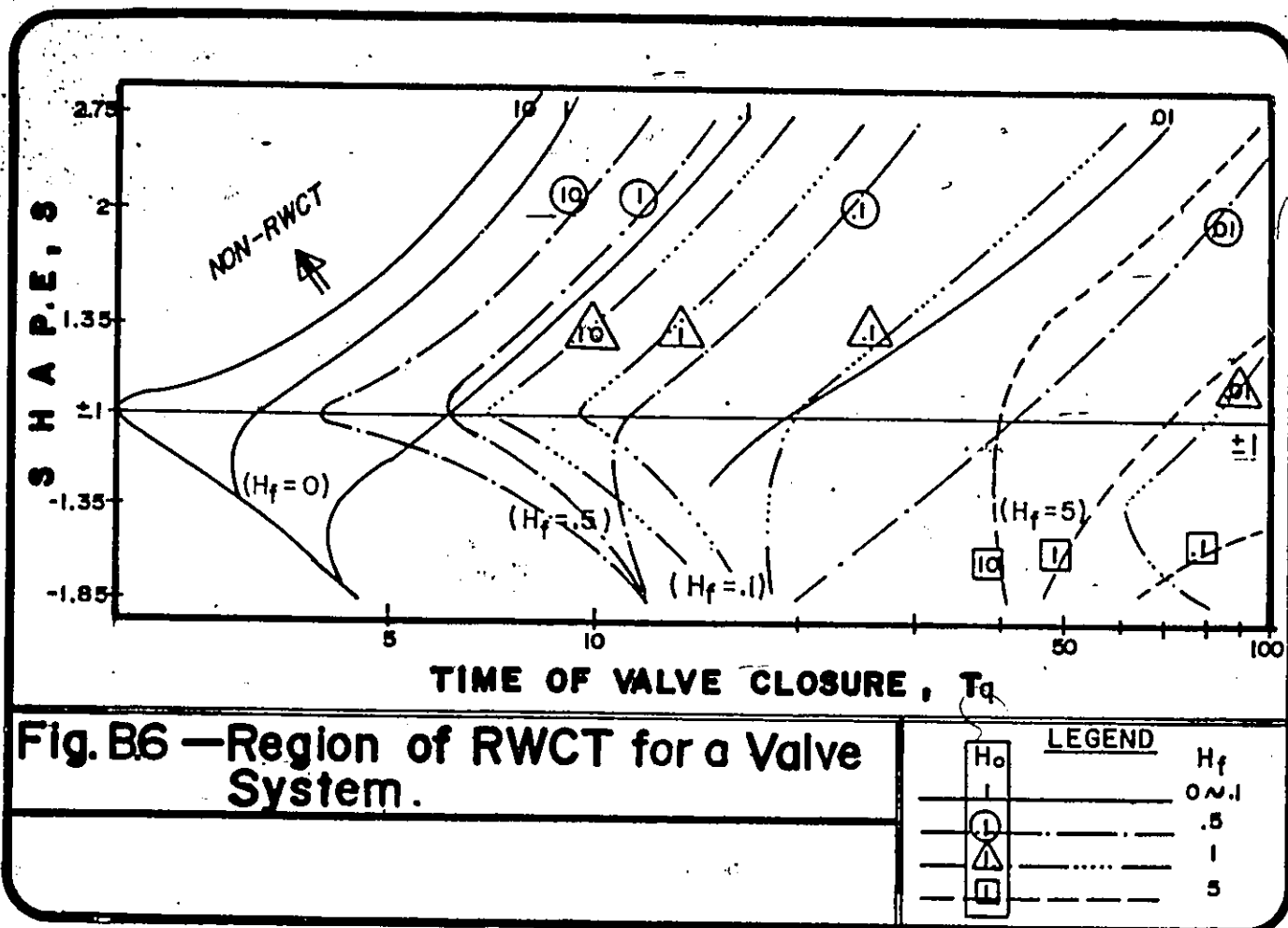
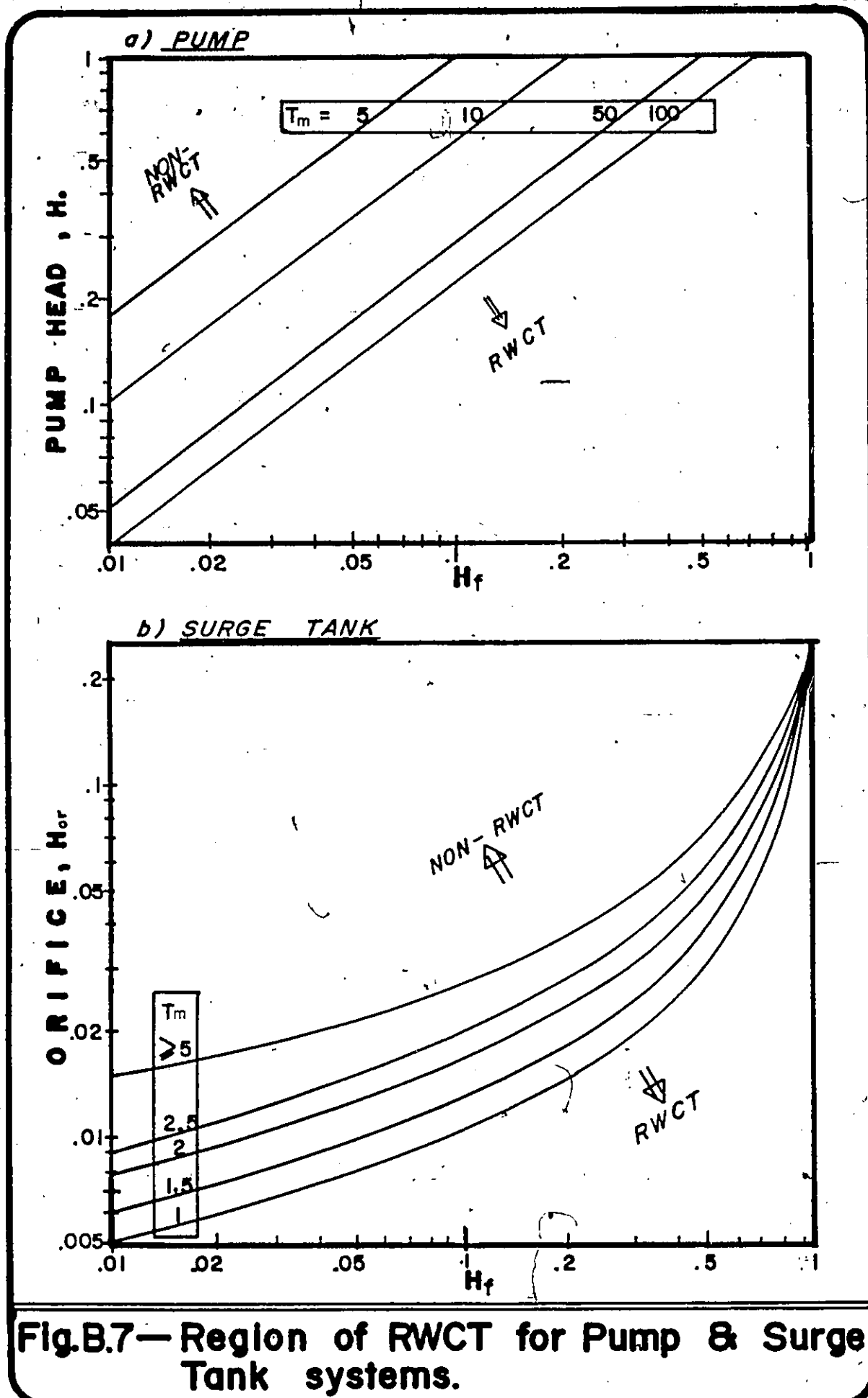
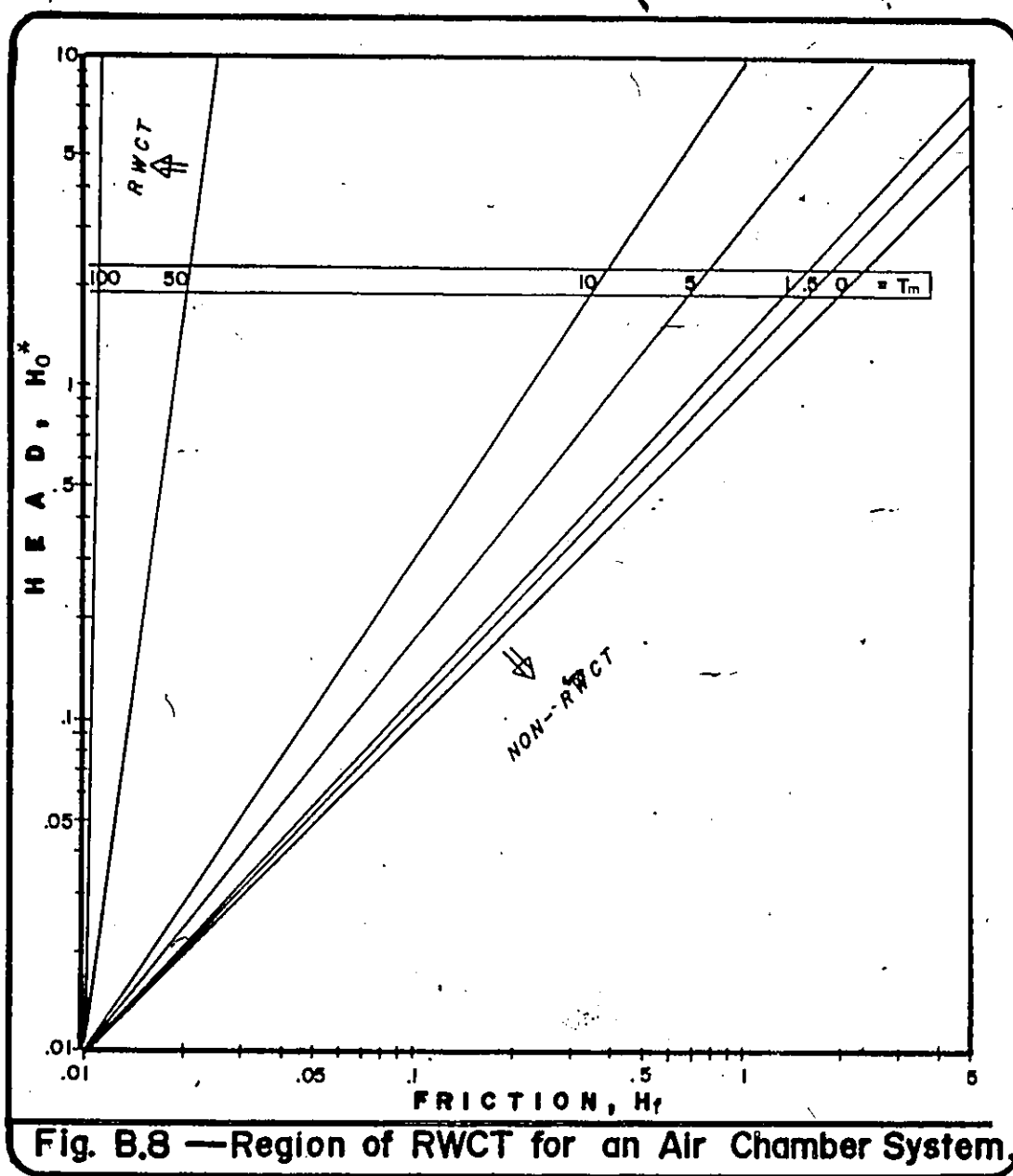


Fig. B.5 — Region of RWCT in a Frictionless Uniform Valve Closure.







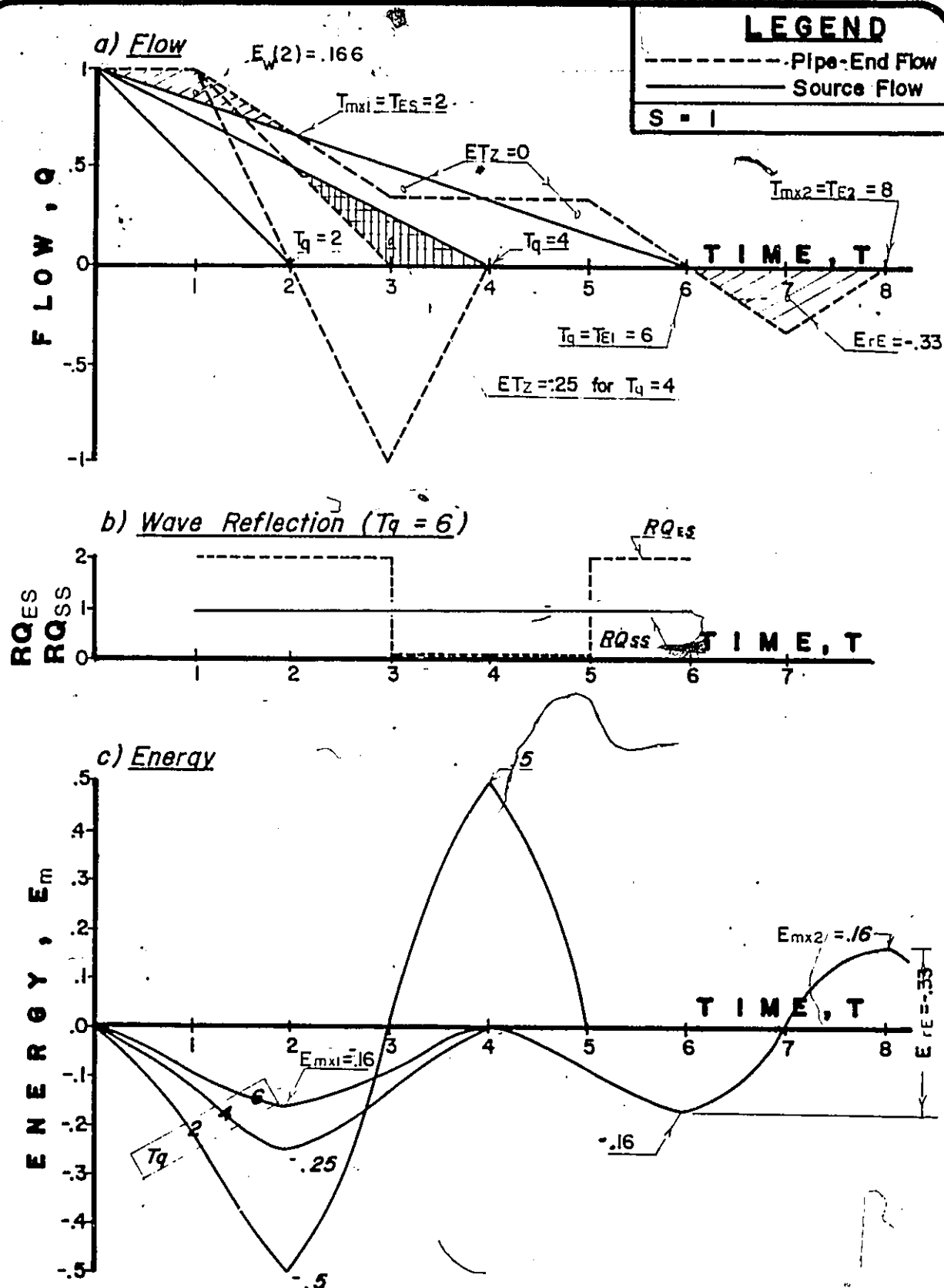


Fig. 5.2—Transient History of Non-frictional, Uniform Deceleration Flow.

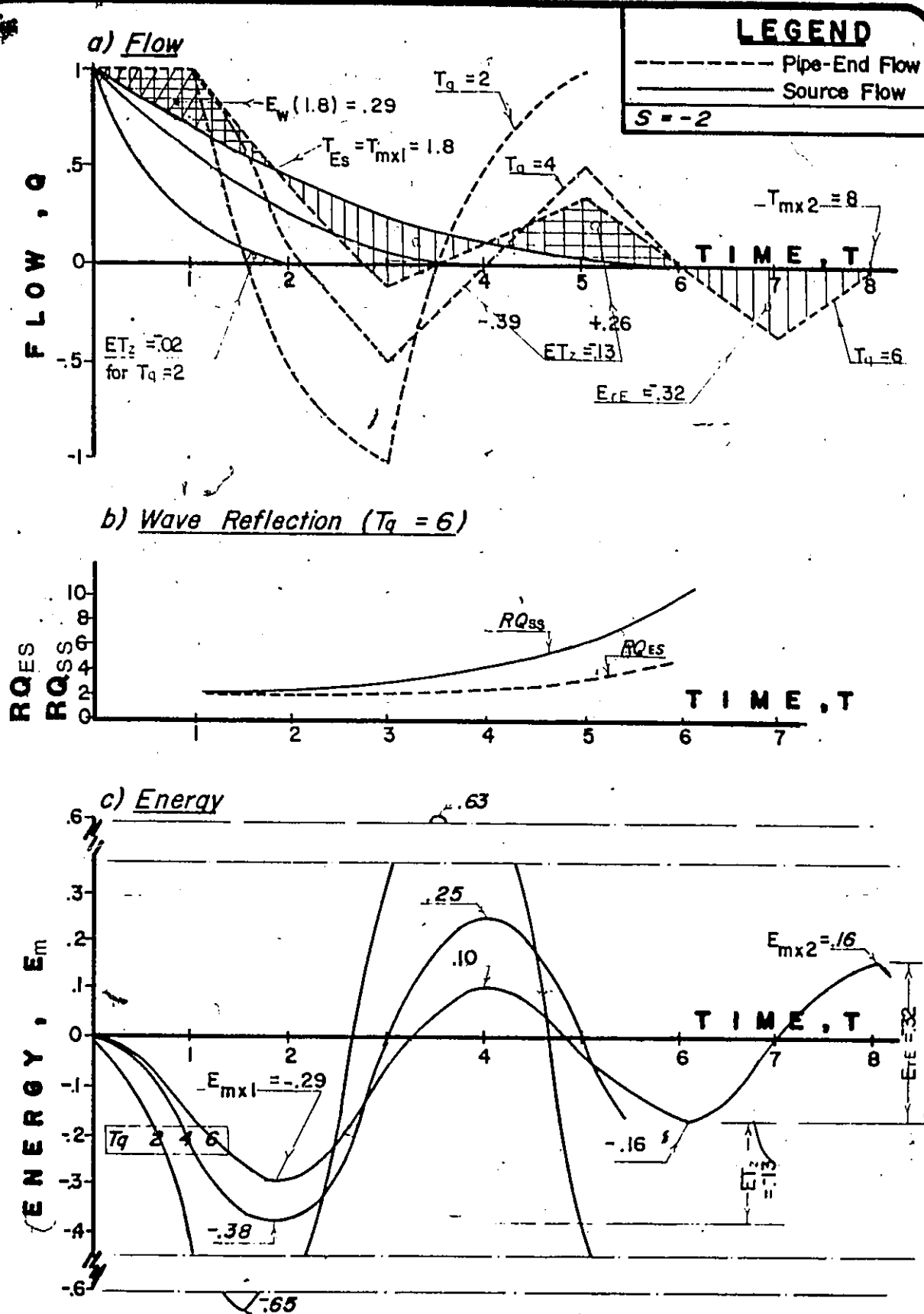


Fig. 5.3 — Transient History of Non-frictional, Decelerating Flow.

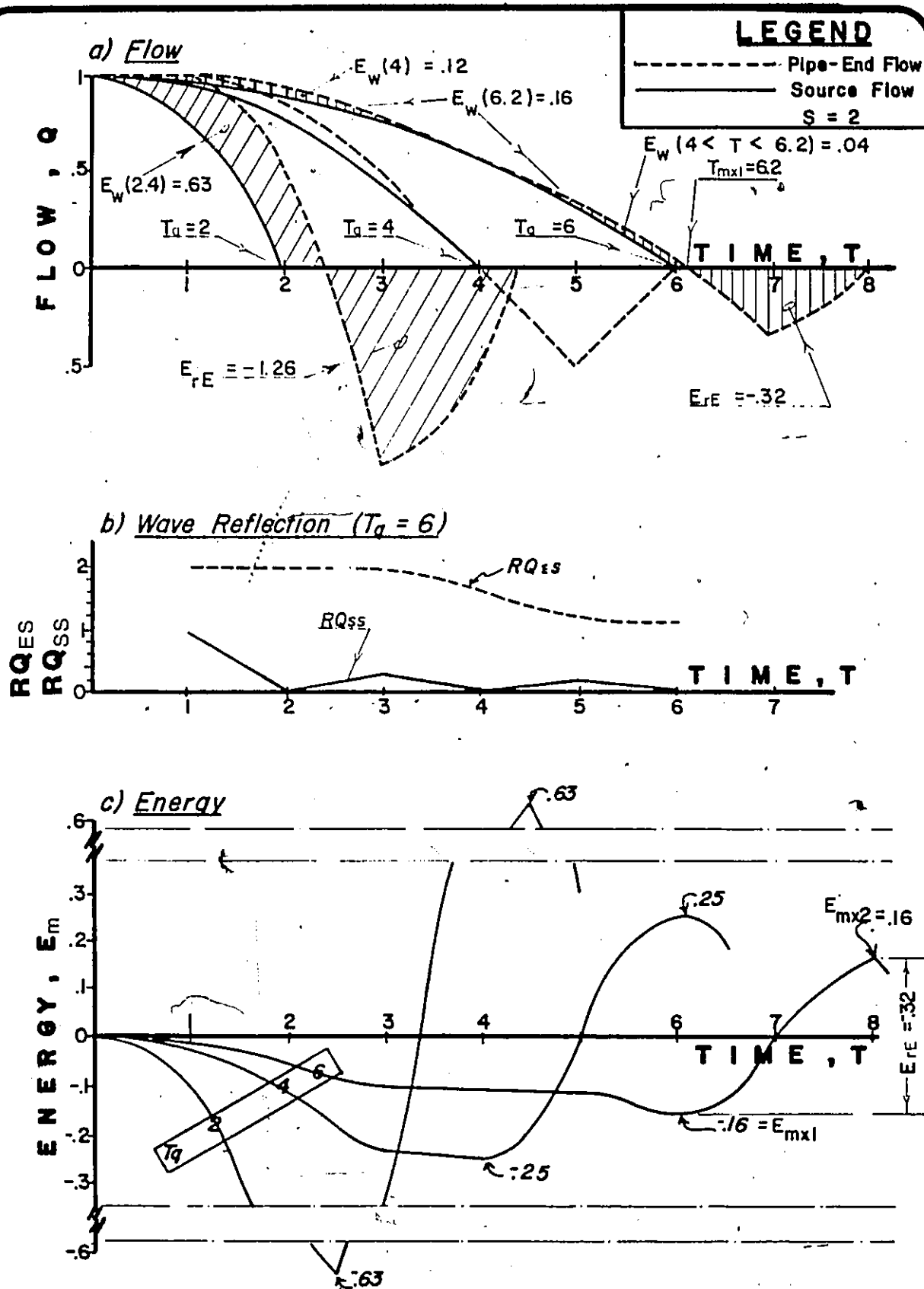


Fig. 5.4—Transient History of Non-Frictional, Increasing Deceleration Flow.

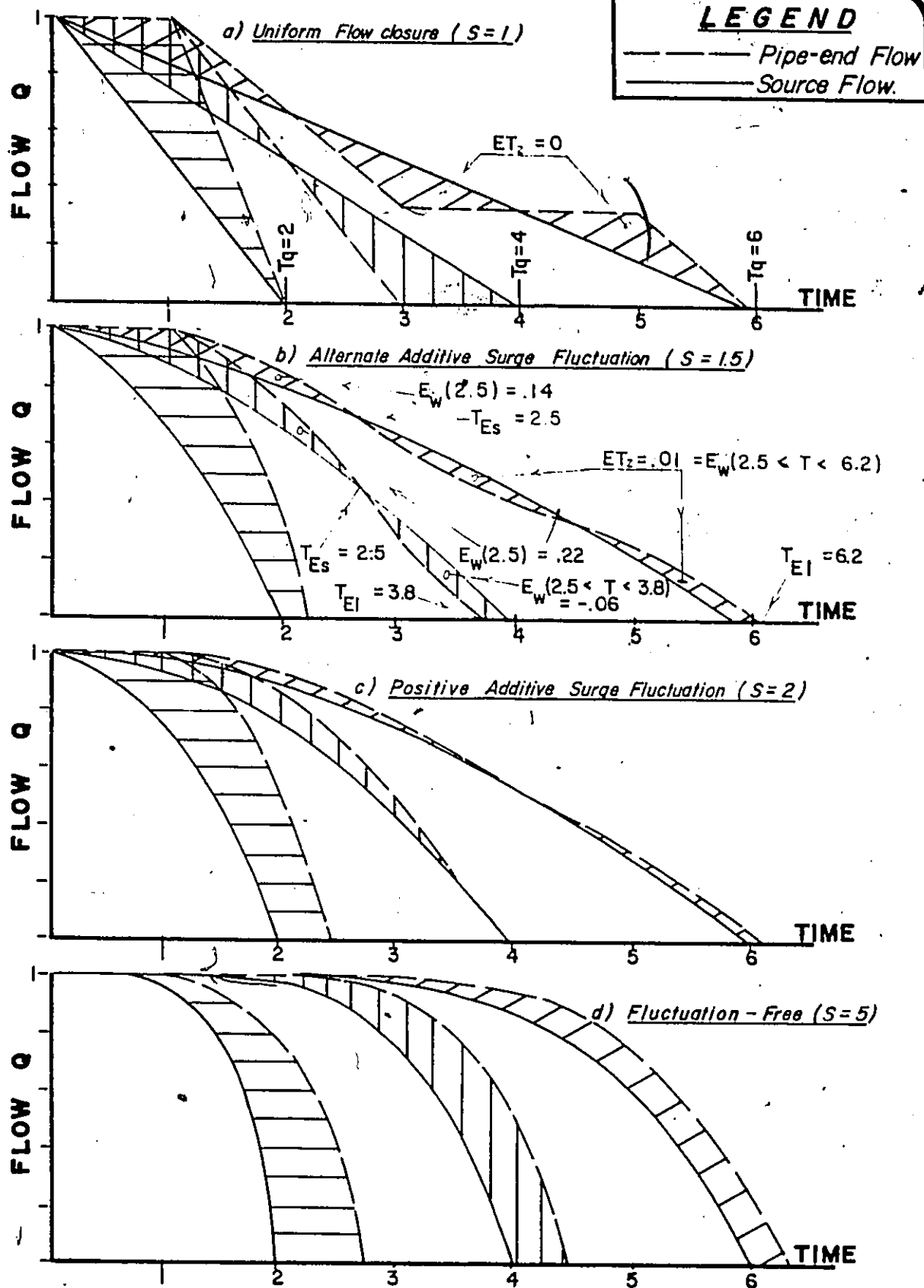
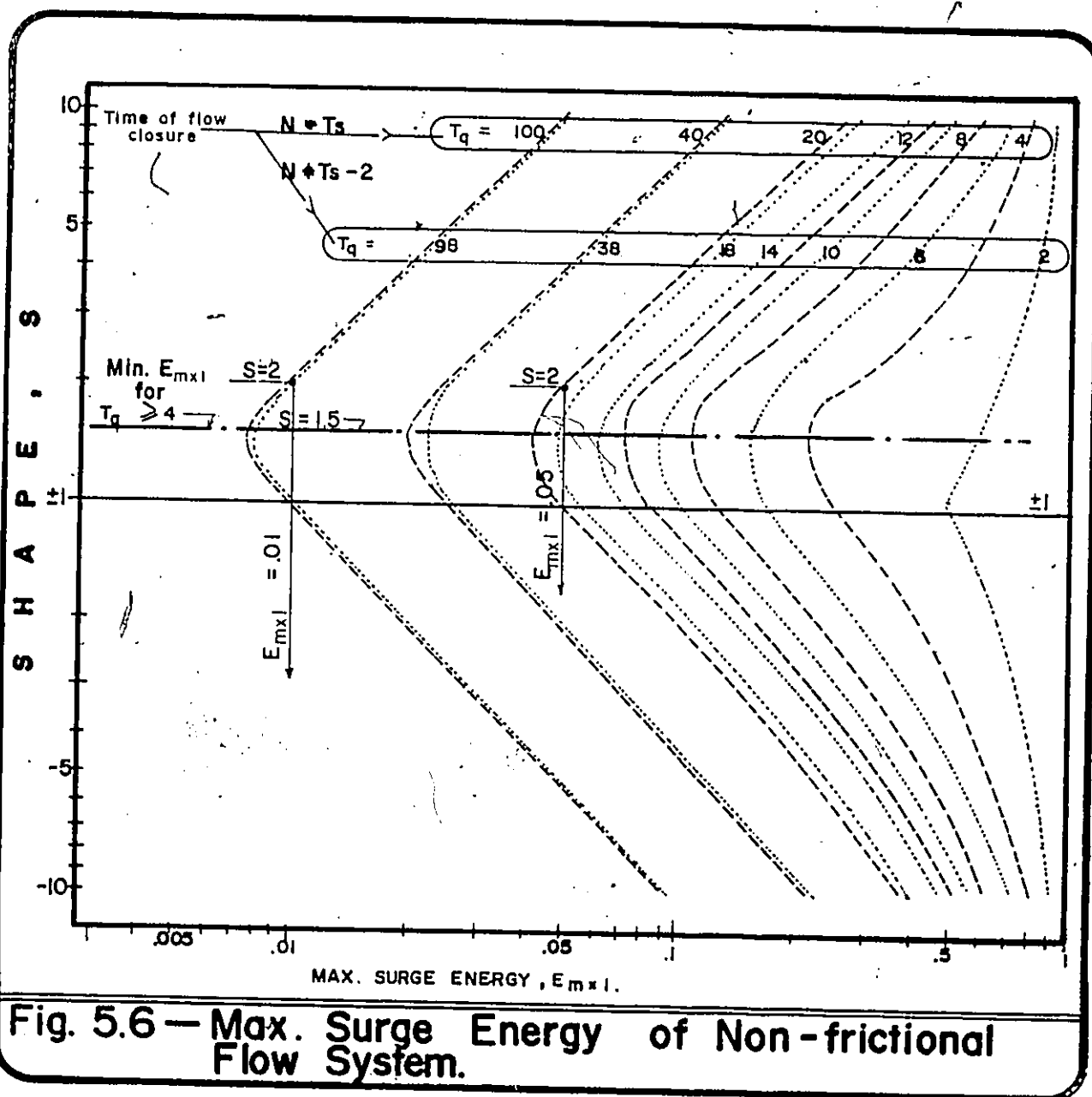


Fig. 5.5 — Flow History of Periodic Surge Fluctuations



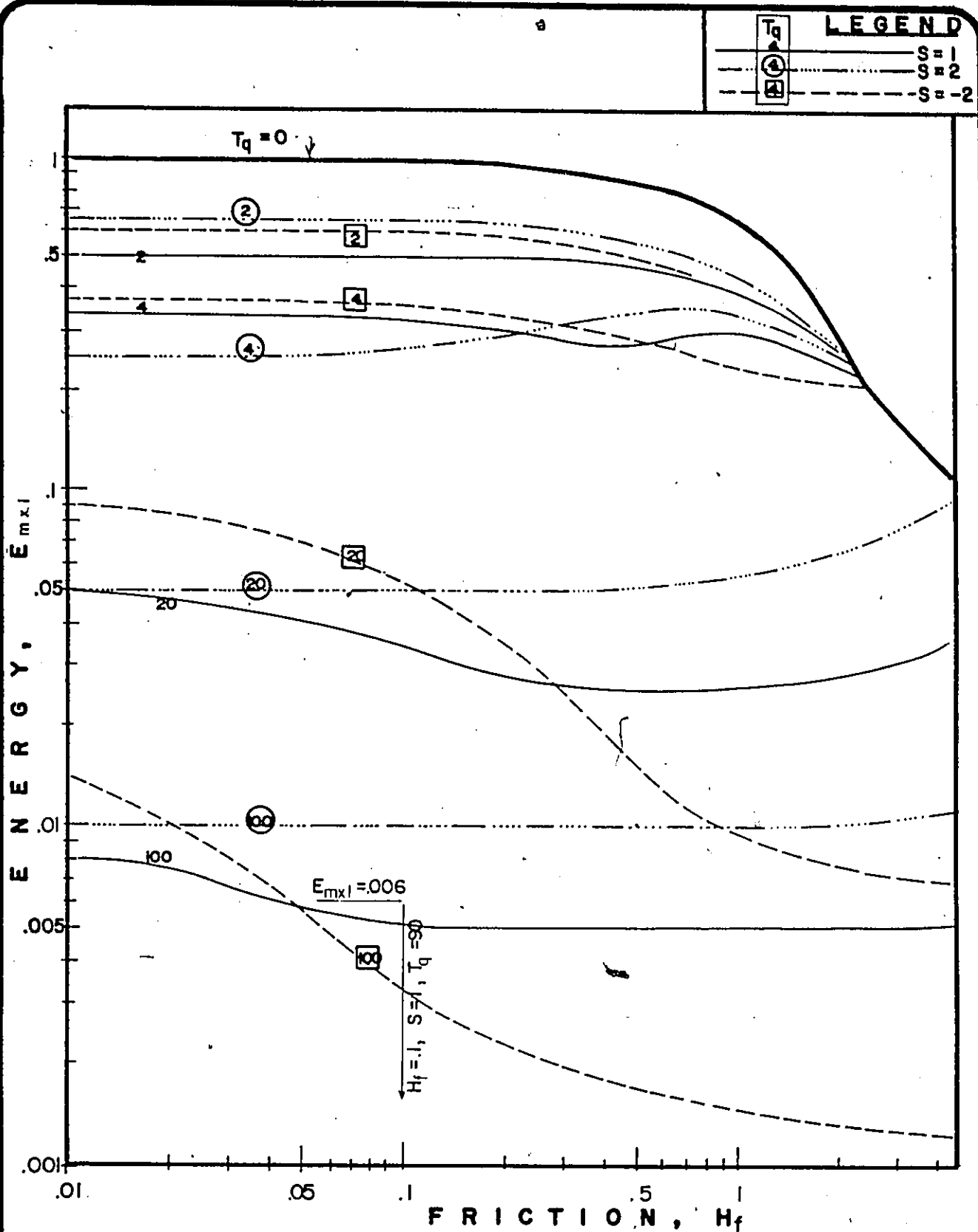


Fig. 6.7 — Maximum Surge Energy induced by Typical Flow Closures.

C.0 ANALYSIS OF COMMON SURGE CONTROL DEVICES IN PIPE SYSTEMS

Analyses, presented in this appendix were conducted in parallel to the research presented in the main body of the thesis. Only abstracts of this work are appended, one page each for the four devices.

Some of the work presented in the thesis applies to all four devices. They are described and summarized herein for each individual abstract for the purpose of independence and completeness. The results of analysis are formulated in a uniform and consistent manner such that they can be used to explore the scientific meaning of the concepts developed as well as to facilitate engineering application.

C.1 Transient Analysis of Valve Closure in A Pipeline

The results of analyzing a broad range of friction, valve head and time of valve closure, or opening are discussed in this section. All seven common types of valve operation in a pipe system are analyzed, namely butterfly, ball, globe, square gate, circular gate, needle and accelerating circular-type gate valves. The valve may be located at either end of the system, and its initial steady state contains both fully open and partially open conditions. A set of valve closure performance curves is developed herein which can be expressed in a simple equation.

Maximum transient pressure at valve and energy in line are presented to produce pressure envelopes along the system. The maximum transients presented here include those for the surge, counter-surge and decaying periods. Results of analysis can be used for the prediction of maximum or minimum pressures along the line as well as for an aid in 'valve stroking' synthesis, which selects the type and time of valve operation under specified maximum pressure conditions. This decaying transient analysis also aids in the determination of when to resume normal valve operation.

The optimal valve closing characteristics vary on the basis of system hydraulic conditions. It is found that the most effective surge protection is the use of the butterfly valve at the low head condition. Conversely, the circular gate valve produces the best surge protection at high valve head.

The effects of friction on valve closure can best be considered using four categories: positive, negative, independent and totally dominant effect on transients. The existence of friction may be irrelevant or actually increase the transients, particular in the low valve head conditions.

Traditional rigid water column theory in a non-frictional, low head and uniform (globe) valve is critically reviewed. A set of new criteria are established and applied to all seven type of valves and a wide scope of frictional and head conditions. In general, the rigidity of flow increases with valve head and time of valve closure, but decreases with the increasing friction.

A steady frictional model is adequate during the valve closing period or early decaying period. However, a quasi-steady friction model may be necessary for laminar or low turbulent flow conditions where frictional energy is significant in the analysis of the transient decay. Should a rapid valve closure occur and the transient decay period become long, a time-dependent friction model is recommended.

Numerical examples are presented, illustrating the application of the design tools formulated from this work.

C.2 Transient Analysis of Pump Trip-out in a Pipeline

The results of analyzing a broad range of friction and pump heads are discussed in this section. The analysis consists of three common types of specific pump speeds, covering from high head centrifugal pumps to low head axial pumps. The existence and non-existence of a check valve at the pump are analyzed for the cases of preventing or allowing flow reversal back to the suction system, respectively.

Maximum transient pressure at pump and energy in line are presented to produce pressure envelopes along the system. The maximum transient events presented here include those occurring at the surge, counter-surge and decaying periods. Results can be used for predicting maximum and minimum pressures along the line, as well as for determining the time to resume normal pump operation for cases with a check valve installed, following a pump trip-out.

For the case where a check valve is absent, it is found that the maximum upsurge transients are significantly reduced, except at high pump head. Without a check valve at pump, the magnitude and time of both maximum and final reversed flows are required to set up the closing performance curves of the control valve to prevent dewatering.

The traditional use of centrifugal pump specific speed may overestimate the resulting transients, particular for cases where the pump head is low and a heavy pump inertial component is added.

A set of new rigid flow motion criteria are established and applied herein. It is found that pump flow can be considered as a rigid body for a low head system.

Maximum surge pressure generally decreases with increasing friction. However, a friction-free zone can still be identified for low friction but high pump head conditions.

A steady friction model is normally adequate during both the downsurge and upsurge periods. However, a quasi-steady friction model may be necessary for laminar or low turbulent flow conditions where frictional energy is significant in the analysis of transient decay. Should the decay of transient become an important issue, a time-dependent friction model is recommended.

Results of analysis are formulated such that they can be used for engineering applications. These are demonstrated by several numerical examples.

C.3 Transient Analysis of Air Chamber in a Pipeline

The results of analyzing a broad range of friction and chamber head are discussed in this abstract. The analysis is performed on both two and one-way air chambers. For the former case (a traditional design), an orifice with a different ratio of head losses between inflow and outflow is considered. For the case where the differential orifice ratio is very large this becomes a one-way chamber.

Applications of an air chamber to both pump forcemain and turbine penstock are considered. The popularity of the latter case is increasing, with a special emphasis on mini-hydro projects. In a low frictional system, transient characteristics induced from these two systems are similar.

Maximum transient pressure at chamber and maximum transient energy in line are presented to produce pressure envelopes along the system. The magnitude and time of critical transient events are also documented, including the maximum surge, maximum counter-surge and peak decaying energies, and various peak flow rates and air volumes. Results documented herein can be used for the prediction of maximum and minimum pressures along the pipeline and the determination of when to resume normal system operation. The complete history of pressure, flow and air inside the chamber, from transient initiation to its decay, can be accurately depicted.

A significant reduction of upsurge pressure can be achieved with the installation of a differential orifice. The variation of air inside the chamber is also a principal factor for sizing an air chamber with an orifice installed.

It is concluded in this analysis that in contrast to traditional design considerations, a small orifice at the chamber produces the optimal transient pressure.

A set of new criteria for rigid flow movement are established and applied herein. In general, the rigidity of flow increases with increasing chamber head and decreasing frictional head.

A steady frictional model is adequate during the transient period in a pipe system with an air chamber.

Examples are presented to demonstrate the application of the design tool formulated from the present work.

C.4 Transient Analysis of Surge Tank in a Pipeline

The results of analyzing a broad range of friction heads are discussed in this section. The analysis consists of both two and one-way surge tanks. For the former case, an orifice with different ratio of head losses between inflow and outflow is considered. A one-way tank is equipped with a check valve with the initial tank water level below steady pressure, and the application of this one-way tank has been increased substantially particularly in sanitary forcemains.

Applications of surge tanks to both pump forcemain and turbine penstock are considered in this work.

Results for maximum transient pressure at tank and maximum transient energy in line are presented to produce maximum pressure envelopes along the pipeline. The magnitude and time of critical transients are also documented, including the maximum surge, maximum counter-surge, and peak decaying pressures, and peak transient flows. Results documented herein can be used for the prediction of maximum and minimum pressures along the pipeline and for the determination of when to resume normal system operation. The complete transient history, both pressure and flow, from transient initiation to complete decay, can be accurately depicted.

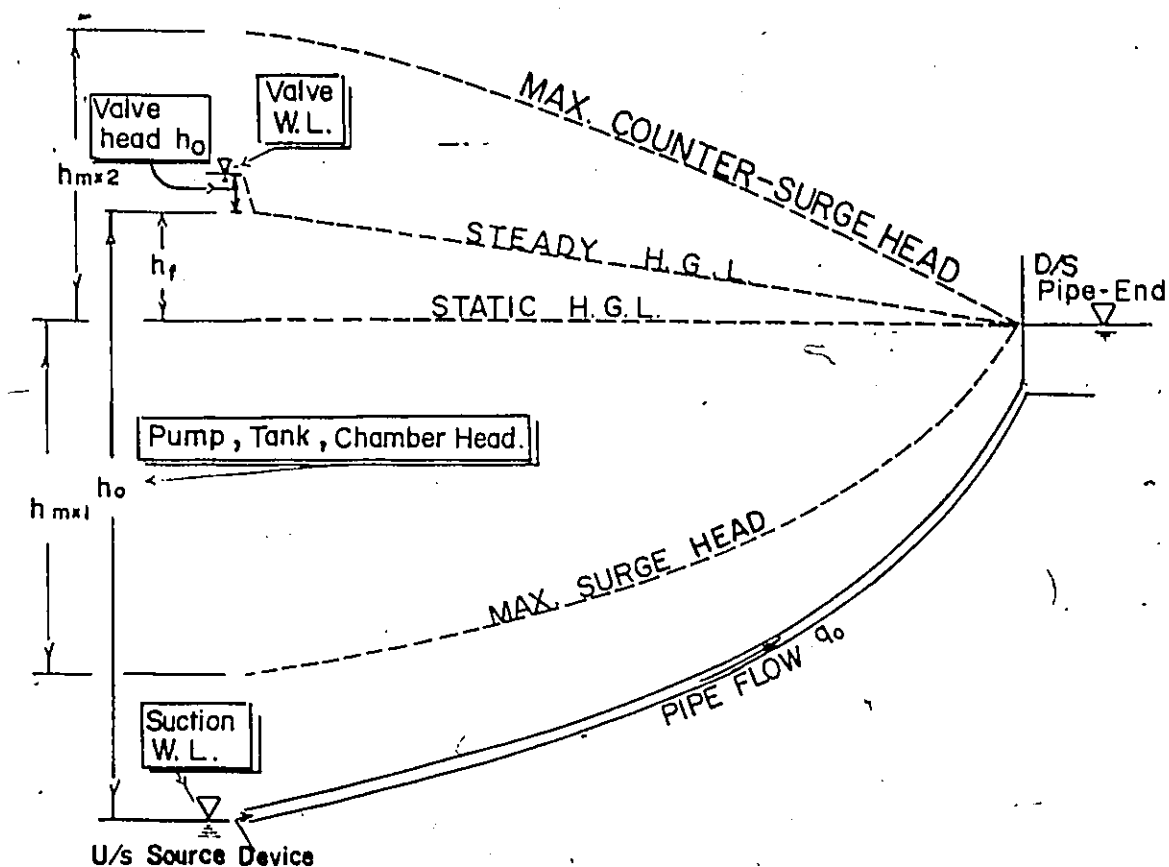
The widely used 'Pressure Balance' design concept is used for accounting the pressure difference between inside and outside the tank due to the head loss of the orifice. The present analysis does not concur with this traditional concept and corrections are made. Further, a significant reduction of upsurge pressure can be achieved with the installation of a differential orifice.

The rigid water column theory is currently used for estimation of the transient characteristics in a system. However, present analysis concludes that inelastic water column theory should be used for cases where a small diameter tank is installed in a long pipeline, or an orifice is present. A new non-dimensional parameter, mechanical travel time of tank, is thus introduced.

A steady frictional model is adequate during the transient period in a pipe system with a simple surge tank. However, a quasi-steady or even a time-dependent friction model may be required for the analysis of transient decay in a one-way tank system.

Examples are presented to demonstrate the application of the design tool formulated in the present work.

a) Hydraulic Conditions



b) Alternative Devices at Source

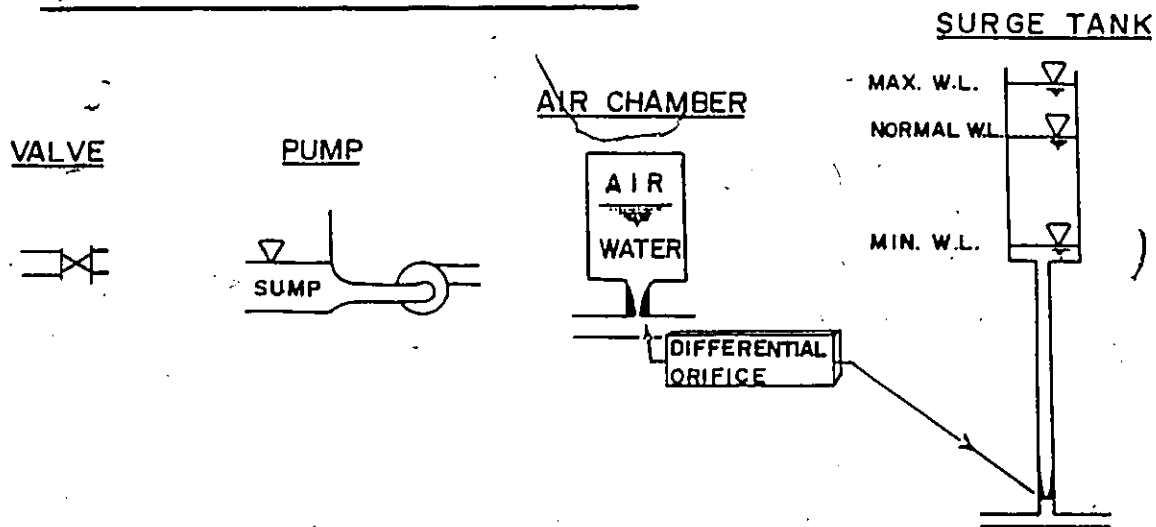


Fig. 1.1 — Schematics of a Pipe System with Various Protective Devices.

D.0 SOME CONTRIBUTIONS FROM THE THESIS

Some of works presented in this research may be extracted or extended for further discussions, and part of them may lead to publications. These include:

TITLE	Extracted or Extended from

1) Dimensional Reasoning as a Tool to Establish Research Direction in Transient Analyses	4.1, 6.3 7.4
2) Model Selection in the Analysis of Unsteady Friction Theory - Numerical vs. Physical	4.2.1, 4.2.3 4.2.4
3) A Numerical Treatment in the Development of Unsteady Friction Models - Simple vs. Complex	4.2.2
4) Characteristics of Transient Flows in a Pipe System - with Various Surge Control Devices	4.3, C.0
5) Energy Balance Concept as a Tool to Interpret Transient Phenomena in a Pipe Flow	5.2, 6.1 7.2
6) Relevance of Periodic Wave Reflections Affecting Transient Characteristics in a Pipe Flow	5.3, 6.2 7.3
7) A Review of Current Unsteady Friction Theories in U-tube and Pulsating Mass Oscillation	2.2, 8.1 A.6
8) Discussions of Current Unsteady Friction Models and their Applications in Transient Pipe Flows	2.3, 8.2
9) Theoretical Background of Time-dependent Friction Model and its Simplified Versions in Pipe Flows	8.3
10) Applications of Friction Models in Transient Pipe Flows- with Common Surge Control Devices	7.5, 7.6 8.4, 8.5
11) Future Development of an Ideal Unsteady Friction Model in Transient Pipe Flows	9.2.1
12) Classification of Frictional Effects on Transient Pipe Flows - with Various Surge Control Devices	A.5
13) Optimal Transient Condition in Pipe Flows - with Common Surge Control Devices	A.4
14) Some Considerations of Classical Rigid Water Column Theory - for a Uniform Valve Closure	B.3

TITLE (Continued)

Extracted
or Extended
from

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| 15) Development of New Rigid Water Column Theory in Pipe Flows - with Friction | B.2, B.4 |
| 16) Application of New Rigid Water Column Theory in Pipe Flows - with Common Surge Control Devices. | B.5, B.6
B.7 |
| 17) Transients induced by Various Types of Valve Closure in a Pipeline - with Friction | C.1 |
| 18) A Design aid on Valve Stroking in a Pipeline - with Friction | C.1 |
| 19) Analysis of Pump Trip-out Equipped with a Check Valve.- from High to Low Head Pumps | C.2 |
| 20) Analysis, Operation and Control of Transients in a Pump Trip-out - without a Check Valve | C.2 |
| 21) Analysis of Air Chamber in a Pumping System - with Differential Orifice and/or Check Valve | C.3 |
| 22) Analysis, Operation and Maintenance of Air Chamber in a Hydro Pipe System | C.3 |
| 23) Design Criteria of an Air Chamber used as a Surge Control Device in a Low-head Sewerage Forcemain | C.3 |
| 24) Analysis of Surge Tank Equipped with Differential Orifice in a Pump or Hydro Pipe System | C.4 |
| 25) Analysis, Operation and Maintenance of One-way Surge Tank in Low-head Water or Sewerage Forcemain | C.4 |
| 26) Impacts of Periodic Shocks during Transient Decay on Protecting Pipeline and Resuming Normal Operation | 7.5, 8.4
C.0 |
| 27) Effect of Minor Losses on Hydraulic Transients - as an Orifice or as a Pumping Station System Loss | 9.2.2 |
| 28) Some Considerations on Analysis and Protection of a Pipe System with the Existence of Vapor Pressure | 9.2.2
/ |
| 29) Some Considerations in the Discretization of Pipe Network for Transient Analysis | 9.2.2 |
| 30) Some Considerations in Surge Pressure Simulations of a Surcharged Storm Sewer System | A.6 |

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1/4, Continued over

HYDROTECHNICAL PROJECTS

During his professional career, Dr. Fok has undertaken work on a wide variety of hydrotechnical projects which include:

- Analysis and design of various waterhammer protective measures including (i) air chambers for Lake Huron water supply mains (London), Easterly filtration plant (Toronto), and Longuil sewerage forcemain (Montreal); (ii) surge tanks for Deacon booster pumping station (Winnipeg), and New Haven pumping station (Ottawa); and (iii) surge relief valves for District #5a watermains (Hamilton) and Kabul water systems (Afghanistan).
- Feasibility study of irrigation, water supply, hydro power and flood control for Clarendon plains development (Jamaica).
- Oil waste dispersion study in St. Clair River for P.A.C.E.
- Ice jam study for South Nation River, Ottawa River (Ontario) and Churchill River (Newfoundland).
- Physical model studies of wave effects for the site selection of Point Lepreau nuclear station (New Brunswick), and analysis and design work for Lepreau intake-outfall alternatives.
- Thermal heat discharge studies, both computer modeling and field dye diffusion tests for Dalhousie power plant and Point Lepreau nuclear station (New Brunswick).
- Water supply and distribution studies for Hamilton-Wentworth, Metro Toronto and Sue Ste. Marie (Ontario).
- Hydraulic structure design of treatment plants for Humber and Thunder Bay, and a diversion channel in Rambo Creek (Ontario).
- Erosion and sediment control studies on outlets of Rambo Creek (Burlington), Don River (Toronto), and Neebing/McIntyre River (Thunder Bay).
- Flood relief to urban drainage areas for Calgary, Edmonton, Winnipeg, St. Catharine, Toronto and St. John.
- Storm water management application for new development areas in North Edmonton, Winnipeg Wonderland and West St. Catharine.
- Floodplain mapping and control studies for conservation authorities and cities in Ontario and Eastern Canada.

During the above studies, Dr. Fok has improved and developed techniques in computer modeling and application. Moreover, he has also contributed numerous technical papers to various journals, conferences, etc.

TECHNICAL PUBLICATIONS

I. Hydraulic Transients (Waterhammer)

- 1) Fok, A., "Design Charts for Air Chamber on Pump Pipelines", J. of Hyd. Div., ASCE, Sept. 1978
- 3) Fok, A., Ashamalla, A., and Aldworth, G., "Considerations in Optimizing Air Chamber for Pumping Plants", Symposium on Fluid Transients and Acoustics in the Power Industry, San Francisco, U.S.A. Dec. 1978
- 2) Fok, A., "Design Charts for Surge Tanks on Pump Discharge Lines", BHRA 3rd International Conference on Pressure Surges, Bedford, England, Mar. 1980
- 4) Fok, A., "Waterhammer and Its Protection in Pumping Systems", Hydrotechnical Conference, CSCE, Edmonton, May 1982
- 5) Fok, A. and Holme, R., "Hydraulic Transient Pressure Considerations", 23 AWWA Conference on Design, Application and Corrosion in Water Pipe, Toronto, Nov. 1982
- 6) Fok, A., Discussion of Song's paper on "Transient Mixed-Flow Models for Storm Sewers", J. of Hyd., ASCE, 1984
- 7) Fok, A., Perks, A., and Fisher, D. "Alternative Approaches for Waterhammer Analysis and Pipeline Protection", Conference on Advances in Underground Pipeline Engineering, ASCE, University of Wisconsin, Madison, 1985

II. Computer Applications

- 8) Fok, A., and Robinson, M., "Computer Aided Hydraulic Design of Treatment Plants", Specialty Conference on Computer Application in Hydrotechnical and Municipal Engineering, Toronto, 1978
- 9) Fok, A., Wigle, R., and Vatagoda, S., "Application of Computer Programs in the Analysis of Water Supply Schemes", Specialty Conference on Computer Application in Hydrotechnical and Municipal Engineering, Toronto, 1978
- 10) Fok, A., Perks, A., and Pataky, L., "Application of Stormwater Management Techniques on an Urban Drainage Relief Study", International Symposium on Urban Storm Relief, Kentucky, 1979

III. Stormwater Management and Urban Drainage

- 11) Fok, A., et al, "Application for Stormwater Management Concepts for a Master Drainage Planning Study", International Symposium on Urban Storm Relief, Kentucky, 1979
- 12) Fok, A., and Tan, C., "Detention Lake Application in Master Drainage Planning", SWMM Users Meeting, Toronto, 1980
- 13) Fok, A., and Wolfe, W., "Analysis of Heavily Infiltrated Sewer Systems", 2nd International Conference on Urban Storm Drainage, Illinois, 1981
- 14) Vatagoda, S., and Fok, A., "A Review of Methodologies and Techniques of Hydrologic Analysis", 5th Canadian Hydrotechnical Conference, CSCE, Fredericton, N.B., 1981
- 15) Fok, A., and Vatagoda, "Some Considerations in Detention Lake Design", 8th International Symposium on Urban Hydrology, Hydraulics and Sediment Control, Kentucky, 1981
- 16) Vatagoda, S., and Fok, A., "Application of Stormwater Management Criteria to New Development Planning", 9th International Symposium on Urban Hydrology, Hydraulics and Sediment Control, Kentucky, 1982
- 17) Fok, A., and Fraser, H., "A Review of Methodologies and Techniques of Sewer Network Planning and Analysis", International Conference on Planning, Construction, Maintenance and Operation of Sewerage Systems, England, 1984

IV. Coastal Engineering

- 18) Fok, A., and Marche, C., "Wave Runup on Irregular Beach Profiles", Conference on Water Resource Engineering, ASCE, Texas, 1977
- 19) Fok, A., et al, "Experimental Verification of Theoretical Solutions for Wave Runup on Irregular Beach Profiles", International Conference on Water Engineering, Bangkok, 1978
- 20) Fok, A., Vatagoda, S., and Bishop, C., "On the Estimation of Scour Depth in a Channel Outlet", 4th National Hydrotechnical Conference, CSCE, Vancouver, 1979
- 21) Fok, A., and McNeely, D., "A Hydrotechnical Study of a River Snow Disposal Facility", 5th Canadian Hydrotechnical Conference, CSCE, Fredericton, N.B., 1981
- 22) Fok, A., and Perks, A., "A Hydrotechnical Study for a River Outlet with Tidal Gate Control", 5th CSCW-CWRS Atlantic Region Hydrotechnical Conference, Fredericton, N.B. 1982