FORCED COOLING OF UNDERGROUND

ELECTRIC POWER TRANSMISSION LINES

PART IV OF IV

ANALYSIS OF PUMPING SYSTEMS FOR THE COOLING OF UNDERGROUND TRANSMISSION LINES

by

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ABSTRACT

Various pumping arrangements and their pressure profile control for forced cooling of long pipe type transmission lines were investigated.

In order to overcome the extensive friction head losses and provide ample cable cooling, a number of pump stations has to be used. Since the inner line segments cannot be provided with pressure control head tanks, line blockages, flow resistance changes, flow rate changes, pump shutdowns, or other imbalances in one segment can alter the pressure profile along the entire line, and, when two head tanks are used, create transverse flow.

Using experimental and analytical methods, it was determined that the pump - relief valve combination operating as a constant flow source is superior to the pump - relief valve combination operating as a constant pressure source, and that the configuration consisting of an even number of loops, each loop having the opposite flow direction from its neighbor's, is the best solution when operated with only one pressure control head tank.

The simplest, and yet effective, line pressure profile control appears to be the pump bypass, which could be easily implemented on existing installations. The head tank pressure adjustment, however, is the most effective line pressure profile control scheme, and should be considered when a new system is being designed. From the analysis performed on an electric analogy model it was found, that the head tank pressure adjustment or the pump bypass would be sufficient to mainain the line pressure profile within its working limits for all practical imbalance sizes, and that, to extend the range of either of these line pressure profile controls, the emergency pump shutdown and the pump bypass itself should be based on the pump discharge, rather than the pump inlet pressure.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A	=	cross sectional area
A	=	nonlinear flow resistance under normal operating conditions
A'	=	linear flow resistance under normal operating conditions
a	=	nonlinear flow resistance
a'	=	linear flow resistance
BPS	=	partial pump bypass
D	=	pump displacement
DL	=	discharge line or its flow resistance
DPC	=	dual pressure control
DWN	=	pump shutdown
d	=	pipe diameter (equivalent pipe diameter)
FSS	=	flow source system
f	=	friction factor
H	=	pump head
H_{max}	=	maximum pump pressure head
HE	=	heat exchanger or its flow resistance
HT	=	head tank
LED	=	light emmitting diode
L	=	pipe length
ML	=	main cable line (its flow resistance)
m	=	pressure - voltage conversion factor

NOC = normal operating conditions

NPSH = net positive suction head

n = flow rate - electric current conversion factor

P_{ij} = absolute pressure in loop i ($j = 1$ inlet, $j = 2$ discharge, $j = 3$ point on ML closer to discharge, $j = 4$ point on ML closer to inlet)

$P_{1\ min}$ = minimum inlet pressure

$P_{2\ max}$ = maximum discharge pressure

$P_{4\ min}$ = minimum main line pressure

P_{cav} = pump cavitation pressure

ΔP = pressure drop under normal operating conditions

$(\Delta P_{HE})_i$ = pressure drop across heat exchanger in loop i

$(\Delta P_{DL})_i$ = pressure drop across discharge line in loop i

$(\Delta P_{ML})_i$ = pressure drop across main line in loop i

Δp = pressure drop

PSS = constant pressure source system

Q = main line flow rate under normal operating conditions

Q_l = pump leakage flow rate

Q_o = ideal pump delivery

Q_r = pump cavitation losses

Q_T = transverse flow rate

q = flow rate

q' = ideal model flow rate

R = pump leakage flow resistance under normal operating conditions

Re = Reynolds number

r = nonlinear pump bypass (leakage) resistance

r' = linear pump bypass (leakage) flow resistance

SPC = single pressure control

t = time

V = battery voltage in loop i

v = flow velocity at steady state (Q = v A)

u = flow velocity (q = u A)

x = distance

β = bulk modulus of elasticity

η_v = pump volumetric efficiency

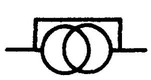
μ = fluid viscosity

ω = pump speed

ρ = fluid density

Diagram symbols

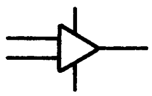
 = pump - relief valve as a constant flow source

 = pump - relief valve as a constant pressure source

 = heat exchanger

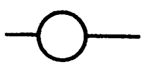
 = flow blocking diaphragm

 = switch relay

 = operational amplifier

 = LED (light emmitting diode)

 = voltmeter

 = microammeter

1. INTRODUCTION

We live in an era when the energy consumption and demand rapidly increase with every day, and when the energy shortage is a bitter reality. Thus increase in load carrying capacity, reduction of losses and increased equipment life of high voltage cable lines have suddenly become a primary concern of electric transmission industry.

Artificial cooling by forced circulation of oil, gas, or water has been used on a variety of apparatus and as the loads become heavier and space becomes more and more limited in critical locations, the use of this technique with dielectric oil in underground cable work was suggested and has been implemented on a few installations by electric utility companies such as the Consolidated Edison Co. of New York, N.Y., Inc.[3], or the Boston Edison Co.[8]. The forced cooling by refrigerated oil significantly increases the load carrying capacity and life span of pipe type electric cable lines. Since many of the older oil filled pipe type feeders readily lend themselves to the application of forced oil cooling, first such systems were built on these installations.

If the total pipe length is not too great, that is if one pumping unit (with a pressure control head tank at the pump inlet and a differential pressure relief valve) is sufficient to drive the oil through the conduit and the heat exchanger, no problems can arise from system pressure profile changes. When, however, the distance between the potheads is such that, in order to overcome the resulting flow resistance, the system has to be separated into smaller units, the interaction of individual units, or loops, becomes a matter of concern. If each pair of the loops can be provided with a pressure control head tank and is separated from the rest of the system by diaphragms in the cable carrying pipe, then each loop is independent of the others and no interaction of the circuits is possible. A line imbalance is defined here as a pipe line blockage, pipe flow resistance change, pump flow reduction, or pump shutdown. A partial blockage is most likely to occur at the heat exchangers, pipe junctions, and around valves and measuring devices, such as flow meters and pressure gages. A pipe flow resistance change may be due to a change in oil viscosity and density (caused in turn by change in temperature level) or, in the main cable line, to twisting of the cable. Pump flow reduction can be caused by poorly maintained or faulty pumps, and a pump may be shut down due to a power failure.

It may not be practical to build head tanks along the entire feeder route since it tends to be very expensive and since the utility company building such a system or adapting its older facility to forced cooling does not usually own enough land along the route to make such installation. Most important of all, the diaphragms available today are not strong enough ($\Delta P_{max} \simeq 40$ psi) for the differential pressures which can be expected to develop when certain line imbalances occur (~ 200 psi). The only solution left is then to build only two pressure control head tanks, each on one end of the pipeline. If this is the case, then imbalances are allowed to "propagate" outside the loop of their origin and alter the pressure profile in the entire system. If an imbalance is large, or if more imbalances occur at one time, a discharge and/or differential pressure of one or more pumps along the cable route may exceed specified limits, or a main cable line pressure may drop below the oil breakdown pressure and the oil insulation capacity will be reduced, or a pump inlet pressure may drop so low that cavitation will ensue. In order to keep these pressures within working limits, further system changes, such as head tank pressure readjustment or pump shutdown or bypass, must be implemented. These changes may, in turn, adversely affect the system, that is the oil flow around the cable may be further reduced and the cable failure probability increased. If two head tanks are in simultaneous operation, a slightest imbalance will cause ratcheting, which is the oil flow from one head tank to the other. To offset this effect, a return line could be built, but again its installation may be impractical.

The following presentation describes, analyzes, and compares the various solutions to the problem. The main criterion in evaluating the merits of individual ideas was a system ability to maintain oil flow through the main cable line at, or close to normal. Other criteria were the minimum line pressure profile variation when an imbalance occurs, and a simple imbalance control with minimal adverse effects on the system.

In Chap. 2 the operation of the present oil filled pipe type system, employed by the Consolidated Edison Co. of New York, N.Y. on its Dunwoodie - Rainey installation, is described, and a survey of possible solutions to its adaptation for forced cooling is presented in Chap. 3. In order to simplify the search for new solutions and to enable the analysis of the more complicated control schemes, an electric analogy model for steady state simulation was built and its operation and hardware are described in Chap. 4; Sec. 4.1 contains general functional relationships of the prototype and its model, and Sec. 4.2 presents the actual electronic componenents and circuits. The model does not include simulation of transient effects during a pump start-up since the size of the system does not allow lumped parameter modeling (Chap. 7). Due to the simplicity of the constant pressure source system and its pressure control scheme, it is possible to apply analytical methods to obtain the effects of imbalances on its line pressure profile and the main line oil flow rates. These results were compared with the response to imbalances of other solutions, such as the constant flow source system, which was obtained experimentally from the model. The experimental program is described, and the various configurations and their pressure control schemes for selected imbalances are compared in Chap. 5. In Chap. 6 an attempt was made to generalize the line blockages so that the results of this work could be applied to systems with different number of loops. A procedure for determination of the transient effects during a pump start-up or a shutdown is suggested and outlined in Chap. 7. In Chap. 8 the results of Chap. 5 and Chap. 7 are summarized, and an optimal solution for the system described in Chap. 2, Fig. 1 or 2, and Tab. 1 is presented. A reader interested mainly in the applied aspects of this work can, without a loss of continuity, skip Sec. 3.1.2 - 3.1.4, Chap. 4, and Chap. 6. Reading Sec. 4.1 together with the error analysis in the Appendix A may be helpful in establishing the range of validity of the experimental program.

2. DESCRIPTION OF THE OPERATION AND THE CHARACTERISTICS OF THE PRESENT DUNWOODIE - RAINEY SYSTEM

The Dunwoodie - Rainey electric power transmission pipe type cable line of Consolidated Edison Co. of New York, N.Y., Inc. is located in the Weschester county north of New York, N.Y. The total length of the feeder route between the Dunwoodie terminal and the Rainey terminal in Queens, New York is about 15 miles. The pressure level increase at Rainey due to the elevation difference is approximately 100 psi. The cable is designed for 345 kV and serves double purpose to carry electric power in either direction as needed. The pipe strength is approximately 800 psi in the discharge line and 600 psi in the main cable line, and, in order to maintain the insulation properties of the oil, the main line pressure can not drop below 150 psi. The pothead on each end of the line is rated at 400 psi, and the gear pumps to be used for the forced cooling operation have an approximate pressure rating of 450 psi.

The system is operated as an oil filled cable circuit with negligible oil flow. It is planned to adapt this system to forced cooling so that the cable power carrying capacity be increased by initialization of rapid oil circulation at the peak load periods. The flow rates needed for the forced cooling operation were determined from the desired cable loading and the cable-to-oil heat transfer

Pipes:

Loop	Length (ft) ML	Length (ft) DL	Q (gpm)
1	13,970	13,980	312
2	14,170	14,370	336
3	13,360	13,560	312
4	13,340	14,140	312
5	13,860	14,660	312
6	12,800	12,800	288

ML I.D. = 10.250 in
DL I.D. = 5.047 in

Cables:

Three with O.D. = 3.935 in (4.135 in across skid wires). It is assumed that the skid wires contribute to turbulence only.

Oil:

Low viscosity polybutene with hot temperature 49°C, cold temperature 25°C.
Assume that the oil temperature in the entire pump discharge line is 25°C and properties of oil in the main line are found by averaging those at 25°C and 45°C.

Tab. 1. System specifications used in modeling and analysis.

considerations[15] and are listed in Tab. 1. Under normal operating conditions ($\sim$ 300 gpm) the flow would be in the turbulent regime but it could become laminar if the flow rate is significantly reduced, especially in the main cable line.

In order ot overcome the flow resistance and provide ample cable cooling during the forced cooling operation, the feeder has to be divided into smaller segments. As shown in Fig. 1a or 2a, there are six loops, each provided with a pump station and a heat exchanger. The heat exchangers are located upstream from each pump since they are much easier to manufacture for lower pressures. The flow direction in each loop is opposite from its neighbor's as indicated on Fig. 1a or 2a. Since there are pressure control head tanks at the line ends only, if an imbalance occurs in one of the loops, the imbalance can propagate in the direction away from the head tank. This imbalance propagation is discussed in greater detail in Chap. 5. The system specifications pertaining to the planned forced cooling operation are listed in Tab. 1.

3. SURVEY OF POSSIBLE SOLUTIONS

The purpose of this study was to find a system configuration and its pressure profile control which, with imbalances in the system, would maintain constant, or, at least, maximize the main cable line (ML) flow rates. Imbalance effects on the system should be minimal and the pressure control scheme should be easy to implement with limited adverse effects on the system.

This chapter contains a brief description of various design ideas (system configurations and line pressure control schemes), some of them perhaps impractical, and the merits of individual ideas are evaluated and compared. The feasible solutions are selected for closer analysis and evaluation in Chap. 5.

Following design limitations were observed:

The feeder has to be divided into smaller segments. The length of each segment is selected according to available sites for pumping and refrigeration stations and on the basis of the pumps capacity and their pressure rating. As discussed in Chap. 8, to improve the system controllability the segment lengths may vary.

Due to the relatively low pothead strength ($\sim$ 400 psi), the flow direction in the terminal loops should always be toward the potheads and not away from them, so that the pressure drop across the main cable line of the last segment would reduce the pothead pressure.

For obvious reasons it is not practical to build pressure control head tanks (HT) along the entire cable route. A head tank on each end of the line should be built. The operating HT pressure is governed by the route profile and the oil demand considerations. The upper limit of the HT pressure (~ 400 psi) is determined by the strength of the potheads and the pipes. The lower limit is determined by ionization tests of the cooling oil[6] and by the existing line pressure profile. Nowhere in the ML of the system should the pressure be allowed to drop below the oil breakdown pressure when the oil begins to lose its insulation properties. As a safety measure, when using the oil whose properties are listed in Tab. 1, a ML pressure should never be allowed to drop below 150 psia[3].

Each pump must be provided with a relief valve, protecting it from excessive differential pressures. With the gear pumps to be used on the Dunwoodie - Rainey installation the pump pressure rating is approximately 450 psi. If a constant pressure source (relief valve maintains constant pump head H) is not used, to maintain a pump discharge pressure below the pipe strength limit (here about 800 psia), additional pump control, independent of the pump head control, must be provided. Either the pump inlet or the pump discharge pressures can be followed to determine the instant at which the pump discharge pressure control should begin. For this purpose either the HT pressure adjustment, pump bypass, pump shutdown, or one of the artificial blockage type imbalance controls, discussed in Sec. 3.2 and Chap. 6, can be used. The selection of a method should be such, so as to limit its effect on the system, in a sense that it should cause the smallest possible reduction of the ML flow. For this reason a pump shutdown should be avoided. It should be used only in cases when other schemes fail to provide sufficient amount of control.

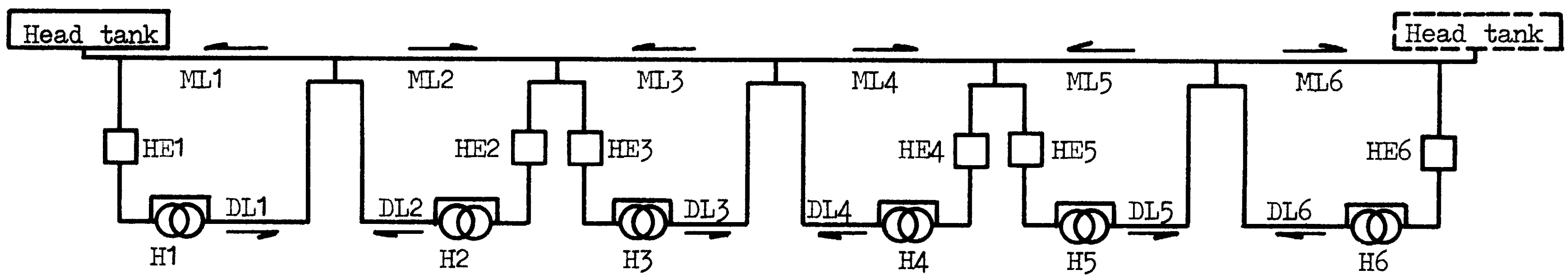
3.1. System Configurations

3.1.1. Configuration 1

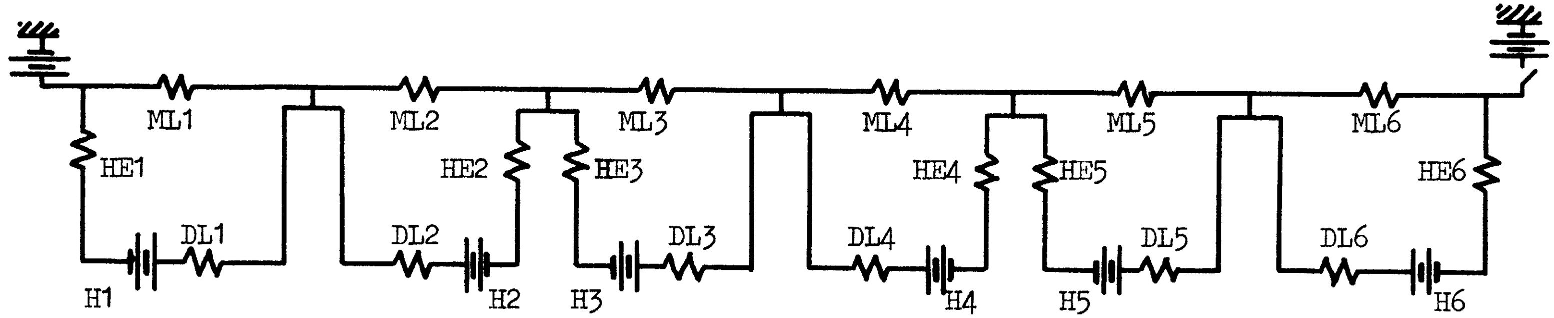
This configuration corresponds to that of the present Dunwoodie - Rainey installation, and is described in the first paragraph of Chap. 2. Fig. 1 represents this configuration with constant pressure sources and Fig. 2 is Configuration 1 with constant flow sources.

3.1.2. Configuration 2

Shown in Fig. 3, this configuration requires flow blocking diaphragms placed in the ML at locations where the discharge lines (DL) of all unit loops are connected to the ML. The DL connections are crossed and thus circuits formed of two units each are created. A unit loop in the Configuration 1 is equivalent to a circuit in

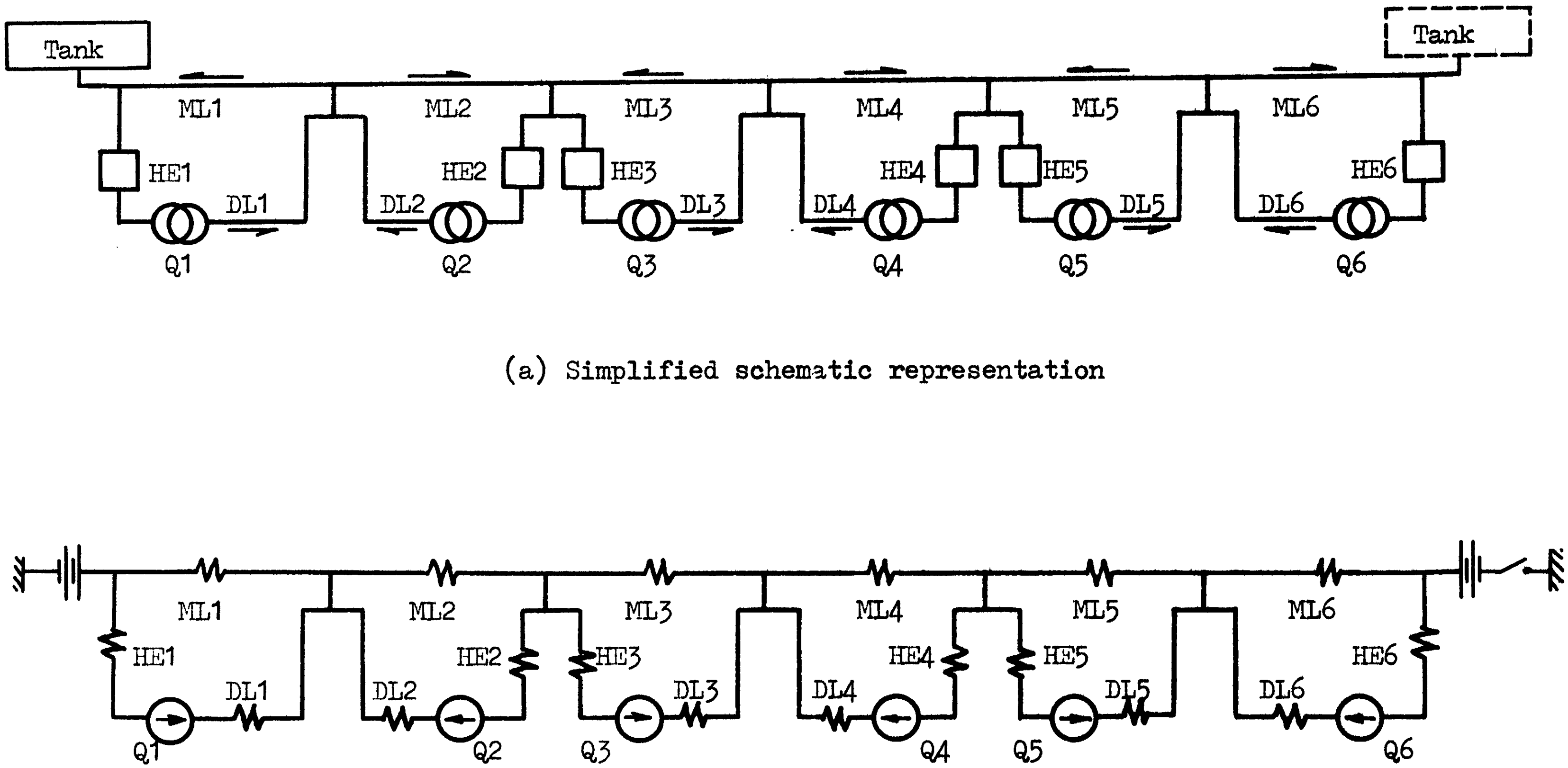


(a) Simplified schematic representation



(b) Electric analogy model

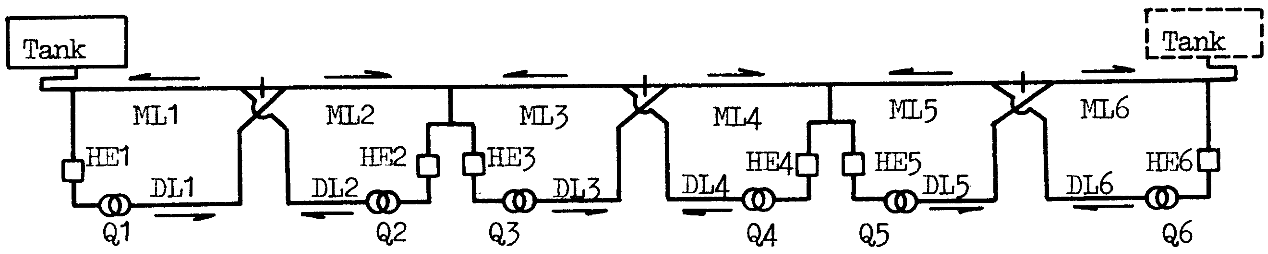
Fig. 1. **Configuration** 1 with pump - relief valve arrangement as constant pressure sources.



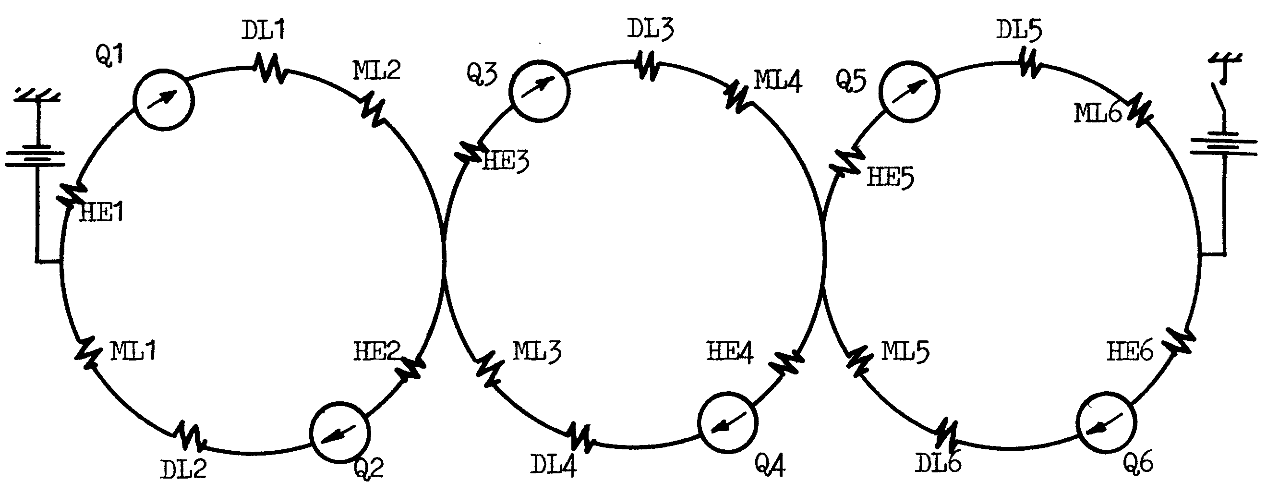
(a) Simplified schematic representation

(b) Electric analogy model

Fig. 2. Configuration 1 with pump - relief valve arrangement as constant flow sources.



(a) Simplified schematic representation



(b) Electric analogy model

Fig. 3. Configuration 2 with pump relief valve arrangement as constant flow sources.

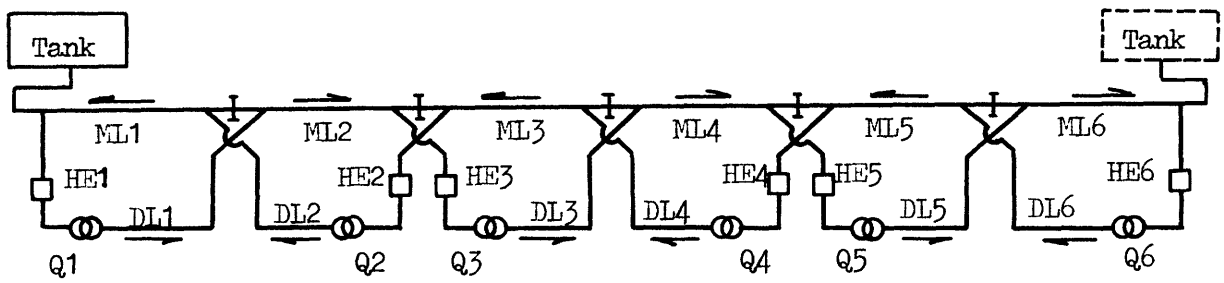
this configuration. This concept will prove to be useful in the treatment of imbalances in Chap. 6.

3.1.3. Configuration 3 (Fig. 4)

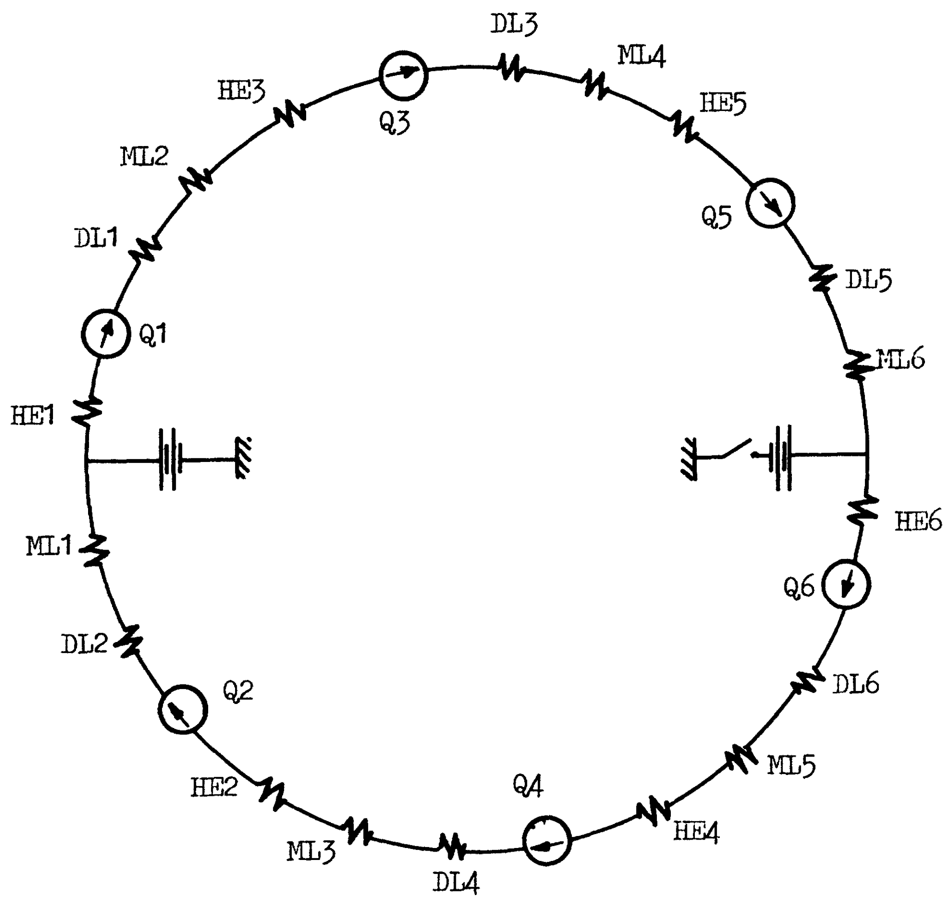
The flow blocking diaphragms are located between each pair of the unit loops and all pipes joining the ML are crossed so that a single circuit is formed. For this reason the flow rates are identical everywhere as is apparent from Fig. 4b.

3.1.4. Configuration 4 (Fig. 5)

The liquid is pumped in one direction along the ML and is sent back through the return line by additional pumps. The flow blocking diaphragms are placed across each ML pump and must therefore be capable of withstanding a pressure equal to the maximum pressure rating of these pumps (here 450 psi). Since the distance between the pump stations is determined by the cable cooling requirements, the ML pump separation and their capacity must be the same as in the Configurations 1, 2, and 3, but, due to the absence of long discharge lines (needed in Configurations 1, 2, and 3), the ML pumps load is smaller and so their pressure rating can also be much smaller than the pressure rating of pumps in Configurations 1, 2, and 3.

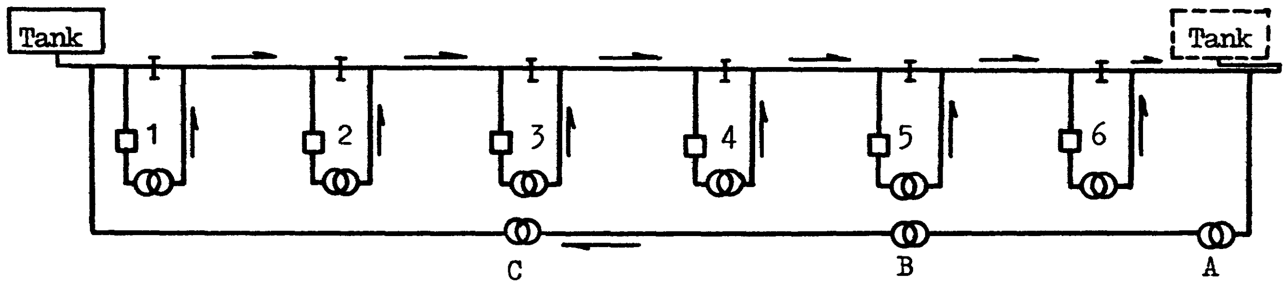


(a) Simplified schematic representation

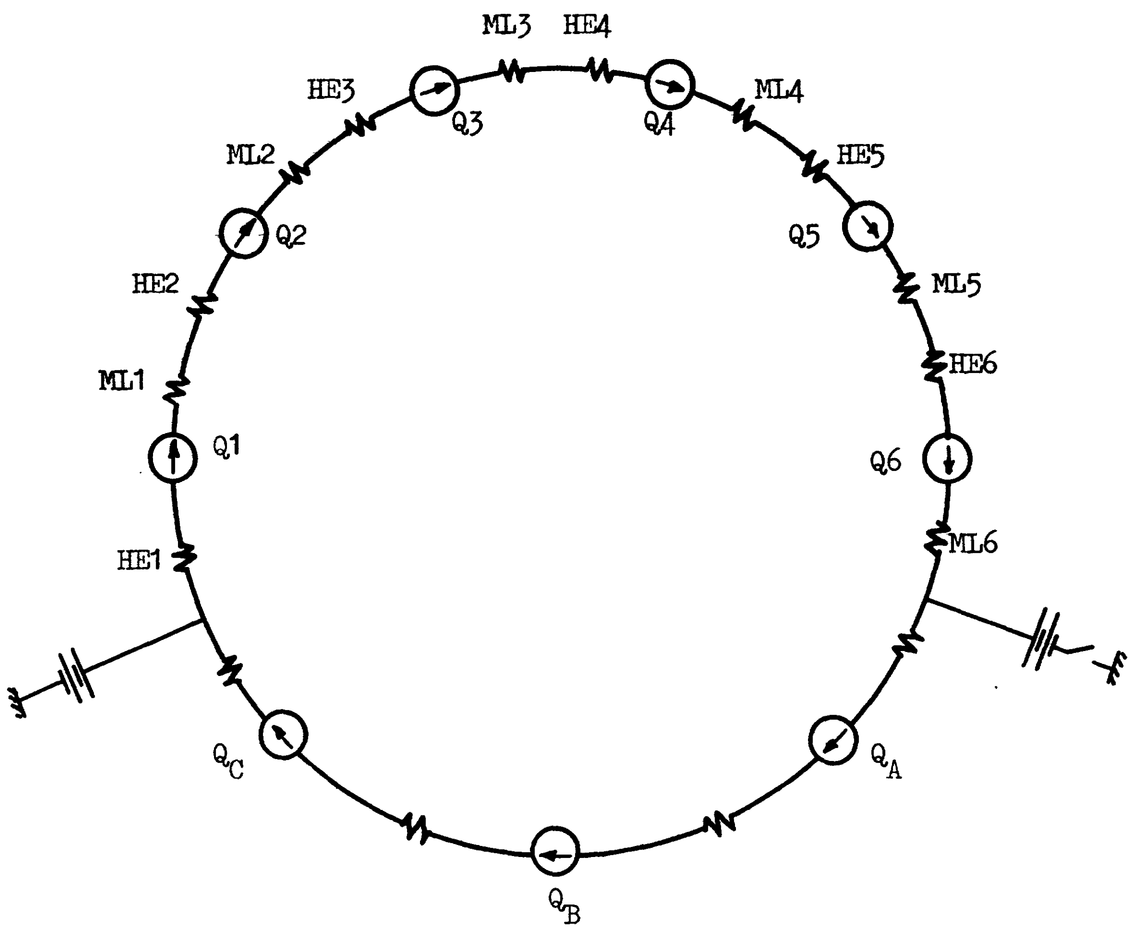


(b) Electric analogy model

Fig. 4. Configuration 3 with constant flow sources



(a) Simplified schematic representation



(b) Electric analogy model

Fig. 5. Configuration 4 with constant flow sources.

3.1.5. Conclusion

The advantage of the Configuration 1 over the other configurations is the fact, that for its operation it does not require the use of the flow blocking diaphragms. Another argument against the use of Configurations 2, 3, and 4 is the possibility of forming of hot spots in the diaphragm neighborhood.

In the Configuration 4 the flow direction is away from one of the potheads and the pressure on this pothead may, for some imbalances, exceed the pothead strength. This could be used as a major argument against the application of this configuration.

3.2. Line Pressure Profile Control

Since there can be pressure control head tanks (HT) at the line ends only, any imbalance can affect the pressure profile and the flow rates of the entire system. An ideal line pressure profile control should maintain all system pressures within their specified limits with no further reduction of flow due to the control application. For the present six loop Dunwoodie - Rainey system the pressure limits are presented in Chap. 2 and for convenience again here:

Max. pump discharge pressure	$P_{2\ max}$	= 800 psia
Max. pump head	H_{max}	= 450 psi
Min. pump inlet pressure	P_{cav}	= 15.3 psia

Max. main line pressure $P_{3\ max} = 600$ psia

Min. main line pressure $P_{4\ min} = 150$ psia

The determination of P_{cav} is discussed in Sec. 4.1.1.

In this section various line pressure profile control possibilities are presented. Except in the case of the head tank pressure adjustment, only the control effects on the loop in which the control is applied are discussed here; the more complex control effects on the rest of the system are discussed in Chap. 5, Chap. 6, and in the Appendix B, and the presentation there is limited to the viable solutions only.

3.2.1. Pump Pressure Relief Valve As the Pump Discharge and Inlet Pressure Control

(a) Pump and its relief valve as a constant pressure source:

The pump relief valve is adjusted for the desired ML flow rate and the resulting pump head is thereafter maintained constant by further proportional opening or closing of the relief valve. Thus, if the absolute pressure level of a loop is controlled by some other means, such as the HT pressure adjustment discussed in Sec. 3.2.2, the relief valve can simultaneously control the pump head and pump discharge pressures.

The disadvantage is the relatively large ML flow reduction when a blockage type imbalance occures as compared to a constant flow source of the following section. Much of the pump power is wasted since the gear (positive displacement) pump continues to supply the same amount of flow, a large portion of which has to be bypassed and is not used for cable cooling. This could be somewhat corrected by using a centrifugal pumps (constant head), but the large ML flow rate reduction would still be realized when a blockage occurs. This system will be shown in Chap. 5 to be inferior to the constant flow source system.

(b) Pump and its relief valve as a constant flow source:

In this method the pump pressure relief valve is initially adjusted for the proper main cable line (ML) flow rate. Further opening of the relief valve is delayed until the pump head reaches its specified maximum value (here $\sim$450 psi) and then the valve maintains the pump pressure head constant. The flow rate is maintained almost constant; some flow reduction is observed when larger blockage develops in the line. Since the relief valve controls the pump differential pressure only, additional control on the pressure level must be provided (just as in the case of the constant pressure source) to protect the pipes from overpressure. Any of the imbalance controls, such as the HT pressure adjustment, pump bypass, pump shutdown, or an artificial blockage can be used as the pressure level control.

3.2.2. Head Tank Pressure Adjustment As the Total Line Pressure Profile Control

With this method alone, the pressure profile can be moved up or down. The method consists af varying the head tank pressure according to the behavior of the line pressure profile. The control can be applied continuously (proportionally to the highest or lowest absolute pressure along the line) or only at instances when a line pressure is outside its limits.

(a) Single pressure control (SPC):

The pressure at one end of the line is controlled by a HT and the pressure at the other end is allowed to freely vary according to the conditions existing within the system.

The implementation of this method consists of the observation of the inlet and discharge pressure of all pumps along the cable route. When one or more of the pressures deviates from its limits, a new HT pressure is determined in such a way that all the pump inlet and discharge pressures remain within their limits. If this is not possible, that is if an inlet pressure and a discharge pressure are both simultaneously outside their limits (one too low and the other too high), an additional control may be necessary. Either the pump bypass or pump shutdown may be used for this purpose.

(b) Transverse flow dual pressure control (DPC):

Pressures at both ends of the line are controlled by simultaneous operation of the head tanks located on both ends of the line. In this method of imbalance correction, a return line has to be built, or some other means of transferring the oil between the two head tanks must be provided

The advantage of this scheme is the fact that pressure deviations from the normal operating conditions (NOC) due to imbalances are generally lower than in the non-transverse flow schemes, such as the SPC, with comparable imbalance sizes (see Chap. 5 and Fig. 16). At the same time, however, some ML flow rates are reduced and some increased by the amount of the transverse flow. This flow reduction coupled with the requirement for a return line are the major disadvantages of this method.

(c) Non-transverse flow dual pressure control:

As in (b) both head tanks are operated simultaneously, but as soon as the oil starts to flow from or into a HT, the flow is stopped by changing either HT pressure setting. As in (a) the normal HT pressure setting is determined by monitoring all pump pressures and keeping them within their limits. This scheme is actually a single pressure control (SPC) of (a) since the same pressure profile and flow rates can be achieved by using only a single HT.

3.2.3. Pump Shutdown As Emergency Pump Discharge Pressure Control

Either the pump inlet or the pump discharge pressure can be used as the control input to determine the instant at which the pump shutdown should be initialized. When a constant pressure sources are used, the two alternatives are equivalent, but in constant flow source systems they are not (see next section and Chap. 5).

(a) Pump inlet pressure as the control input:

In this method, when a pump inlet pressure exceeds a set limit, the pump is shut down. The pressure for which the control is set is equal to the maximum allowable pump discharge pressure $P_{2\ max}$ minus the maximum allowable pump head H_{max} (here $P_{2\ max} - H_{max} = 800 - 450 = 350$ psia).

The major drawback of this method is the fact that in constant flow source systems, when the inlet pressure exceeds a set limit, the discharge pressure may still be safely far from its limit. This situation may be worsened if the ML pipe pressure rating is lower than the pressure rating of DL pipe. Then, in order to assure, that a ML does not exceed its upper limit, the maximum allowable pump discharge pressure must be lowered (a situation like this will exist on the Dunwoodie - Rainey system since the maximum ML pressure there is 600 psia — 200 psi less than the DL pipe pressure rating). A treatment of this possibility is presented in Sec. 5.3. The reason for the difference between the inlet pressure observation and the discharge pressure observation for the purpose of the pump shutdown is the fact that the imbalance raising the inlet and discharge pressures has originated outside the loop in consideration and could not therefore increase the differential pressure.

(b) Pump discharge pressure as the control input:

The control in this method is applied directly by measuring the pump discharge pressure. It permits continuous pump operation in situations where the pump inlet pressure observation in constant flow source systems would already call for the pump shutdown.

A disadvantage lies in the fact that if a pump shutdown is necessary, the resulting discharge pressure fall, which must follow, would cause the pump to be turned on again and thus chattering would ensue. Ther are several ways to cope with this phenomenon:

(aa) Manual pump start-up:

At the moment of a pump shutdown, the control is discontinued and an operator has to determine, by observing the pump inlet pressure, whether he could turn on the pump again, and would do so manually. The manual pump start-up would be coupled with the reinitialization of the automatic pump shutdown.

(bb) Inlet pressure controlled pump start-up:

As a pump is shut down, the control input is switched from the pump discharge pressure to the pump inlet pressure. Thus, the possibility of chattering would be removed, since after a pump is shut down its inlet pressure rises, rather than decreases as does the pump discharge pressure. The automatic pump start-up would be coupled with switching the control input from the pump inlet pressure back to the pump discharge pressure.

3.2.4. Pump Bypass As the Pump Discharge And Inlet Pressure Control

This line pressure profile control is an alternative to the head tank pressure adjustment or the pump shutdown and is best suitable for application on systems with constant flow sources. It can also be used as a secondary control to back-up the HT pressure adjustment control.

A combination of a pump bypass and a constant flow source creates a component, which, during normal operation, has a constant flow source characteristics, but which, when the bypass opens to correct a pump pressure, has a flow - pump head characteristic equal to a constant pressure source.

The pump bypass can be implemented by further opening the pump pressure relief valve, or by providing the pump with an additional bypass pipe and a valve. The valve control for the pump discharge pressure control can be based on either the pump discharge or inlet pressure, just as the pump shutdown was in Sec. 3.2.3, with similar consequences. Obviously, using the pump discharge pressure as the bypass valve control input is the better alternative, since the pump head is not, in general, constant.

The valve control for the pump inlet pressure control should be based on the pump inlet pressure since, again, the pump head is not constant in the constant flow source systems.

Thus the bypass valve opens when the pump inlet pressure falls below a set limit (here 130 psia — Sec. 5.3), or when the pump discharge pressure exceeds its set limit (here 700 psia — Sec. 5.3). This dual function of a bypass valve is possible because reduction of flow around the loop simultaneously raises the pump inlet pressure and lowers the pump discharge pressure.

3.2.5. Artificial Line Blockages As the Pump Discharge And Inlet Pressure Control

Even though obviously impractical, this method is presented here in order to demonstrate all possible pressure control solutions. A natural line blockage has the same effect as an artificial one and thus the following can also be viewed as a description of the Configuration 1 response to various natural line blockages.

Flow limiting valves placed at a pump inlet or discharge line or even a ML can alter the pump inlet and discharge pressures. If the pressure at one of the ends of the ML is kept constant, the effect of closing down such a valve is the same as a line blockage would have on the loop.

In a loop with a constant pressure source (see Sec. 3.2.2) closing down a pump inlet valve (or HE blockage) will reduce both the inlet and discharge pressures equally, and reduce the ML flow rate whereas in a loop with a constant flow source (Sec. 3.2.3), where the pump head H is allowed to increase, the discharge pressure will be reduced less than the inlet pressure,and the ML flow rate will be reduced less than it would with a constant pressure source.

Closing down a pump discharge valve (or DL blockage) in a loop with a constant pressure source will increase both the inlet and discharge pressures equally, and reduce the ML flow rate. In a loop with a flow source the discharge pressure will be increased more than the inlet pressure and the ML flow rate will be reduced less than it would, had a constant pressure source been used.

The effect of closing down a valve in the ML (or ML blockage) depends on which ML pressure is kept constant. In any case the ML flow rate will be reduced, and more so if a constant pressure source is used. If P_3 (pressure at the point between ML and DL) is not allowed to vary, then in a loop with a constant pressure source closing down a ML valve will equally reduce the inlet and discharge pressures, and with a constant flow source, since H is allowed to increase, the inlet pressure drop is larger than the discharge pressure drop. If P_4 (pressure at the point between ML and HE) is constant, then in a loop with a constant pressure source closing a ML valve will equally increase the inlet and discharge pressures, and with a flow source the discharge pressure increase will be larger than the increase of the inlet pressure.

3.2.6. Conclusion

The effects of pump shutdown, bypass, and line blockage outside the loop of their origin depends on the number of operational head tanks and their location. These effects are discussed in Chap. 5 and illustrated in the figures of Chap. 5 and Appendix B.

It is apparent that both the constant pressure sources and the constant flow sources are possible pump configurations. Of the HT pressure adjustment methods only the single pressure control (SPC) is a viable solution. For the pump bypass or shutdown the discharge pressure observation is the method to be used. When a pump is shutdown, to prevent chattering, the (aa) method of manual pump start-up seems to be the simplest solution. All these mentioned methods of the line pressure profile control are further analyzed and evaluated in Sec. 5.3.

4. STEADY STATE SIMULATION BY THE ELECTRIC ANALOGY METHOD

In order ot simplify the search for new configurations and pressure profile control schemes, and to ease the steady state analysis of the more complicated systems, it was decided to build an electric analogy model.

4.1. Comparison of Prototype and Model Functional Relationships

Two fundamental analogies exist between the performance of an incompressible fluid in a pipeling network and of electricity in a resistive circuit. With electric current representing flow, the total current approaching a terminal equals the total current leaving it, just as fluid flows balance at a pipeline junction. With voltage drop representing friction head loss, the voltage drop around a closed circuit is equal to zero just as fluid head losses balance around a pipeline loop.

If an electric circuit is connected to simulate a pipeline network, and suitable conversion factors are used to relate electric and hydraulic quantities, the performance of the pipeline network is indicated by conditions in the electric circuit. Complete proportionality of corresponding quantities does not occur, however, unless the voltage drop across each resistor in the electric circuit is related to the current through it in a manner analogous to the nonlinear relation of turbulent flow between head loss and flow rate for the pipeline that it represents. Two general methods have been developed[7] previously for satisfying this nonlinear relation. The first is a direct analogy that involves one or more succesive approximations, between which the settings of ordinary linear resistors must be changed in the direction indicated by the preceding trial, the second method consists of the analysis of pipeline networks by means of electric circuits whose resistors automatically represent an accepted relation between head loss and flow rate in the turbulent regime. The positive variation of resistivity of tungsten with temperature, and therefore with resistor current, is employed in the nonlinear resistors used in the later method. Excellent correspondence between the hydraulic and the electric systems was obtained by both methods[7]. A model utilizing nonlinear resistors, however, is relatively expensive and complicated for the use in this work. Also tungsten resistors are not easily available.

It was decided therefore to use a different approach from the two methods just described. The following presentation describes the electric analogy model used in the analysis of a six loop pipeline network with its basic configuration corresponding to Fig. 2 with the pump-relief valve arrangement corresponding to a constant flow source (see 3.2.3), and with its specifications listed in Tab. 1. In building the model, linear resistors were used throughout since in a linear system the effects of individual imbalances can be superimposed on each other and since such a model is inexpensive and relatively easy to build and operate. No iteration or succesive approximations are necessary. After the collection of data, this method involves computations for corrections of the results as shown in the Appendix A. Since, however, the qualitative analysis is more important here than the actual numerical results, no corrections were applied to the data presented in the figures of Chap. 5 and the Appendix B.

4.1.1. Characteristics of Positive Displacement Pumps[1,4,17-19]

It is assumed that identical gear pumps are used in each loop. Since the length of each ML segment is different, it is necessary to provide the pumps with a special bypass to obtain the required flow rates. It is also assumed that the unit with the largest flow rate would govern the pump selection, and would not be provided with a bypass. The maximum pump head is given as 450 psi and the maximum discharge pressure is 800 psia; the volumetric efficiency of a typical gear pump without the special bypass is approximately 90% and the pressure drop across the HE in the unit with the largest given flow rate (which is #2) is set at 40 psi.

The following analysis is applicable to all positive displacement pumps at steady state. The delivery of a pump can be divided into three factors:

$$Q = Q_o - Q_l - Q_r \tag{4-1}$$

The ideal pump delivery $Q_o = Q / \eta_v$ is a function of the pump physical dimensions and its shaft speed:

$$Q_o = D\omega \tag{4-2}$$

The leakage Q_l is caused by the flow through the small clearence spaces between the various parts that separate high and low pressure regions, and here, considering the pump special bypass to be a part of the pump, the bypassed flow is an additional source of leakage. The cavitation losses Q_r become significant when the pump inlet pressure approaches the pumping liquid vapor pressure.

An exact pump model is a current source in parallel with a resistor, representing the pump leakage. In general, the leakage resistance is nonlinear (turbulent flow through the bypass, laminar flow through the small clearence spaces), but here, since a linear model is being built, the leak resistance must be linearized. Using the Ohm's law:

$$H = r' \, Q_l \tag{4-3}$$

where H is the pump pressure head and r' is the linearized leak resistance.

Knowing the pump volumetric efficiency without the special bypass, η_v , the pump head H (from Tab. 2) and the actual desired ML flow rate under NOC (normal operating conditions) Q, the linear leak resistance r' for each pump can be determined from (4-3) and from:

$$Q_1 = Q_o - Q = Q_2 / \eta_v - Q \tag{4-4}$$

since it is assumed that the flow rate of pump #2 would be equal to the required ML2 flow rate and therefore the leakage of pump #2 would be due to the flow through the small clearence spaces of the pump only.

Thus

$$r' = H / Q_1 = H \eta_v / (Q_2 - \eta_v Q) \tag{4-5}$$

and the values of r' for each loop are listed in Tab. 3.

The appearence of cavitation usually is evidenced by the drop in pump head and efficiency below the well established values under ample net positive suction head (NPSH) conditions[10,12,13]. NPSH is defined as the absolute suction pressure less the vapor pressure at suction temperature. Since it is known that in the case of mixtures of oils the required NPSH is lower and cavitation less severe than in the case of cold water[10], the cold water NPSH at cavitation inception is a good approximation for limiting NPSH. Since oils have generally lower vapor pressures than cold water, assume that the inlet oil pressure at which cavitation begins is the pressure corresponding to cavitation with water:

$$\begin{aligned} P_{cav} &= NPSH_{water} + P_{water\ vapor} + P_{atm} \\ &= 0.1722 + 0.3887 + 14.696 \qquad (4\text{-}6) \\ &= 15.257 \text{ psia} \end{aligned}$$

It can be therefore safely assumed that cavitation will not occur when

$$P_{cav} \geq 15.3 \text{ psia}$$

The cavitation model consists of a variable resistor and a switch relay connected in parallel between the pump and the circuit. When the inlet pressure drops below 15.3 psia the switch is closed and the flow and pump head reduced. The actual pump cavitation performance is compared with the model performance in Fig. 6.

The pump relief valve is modeled by a zener diode connected across the pump. The flow through such a diode is virtually zero

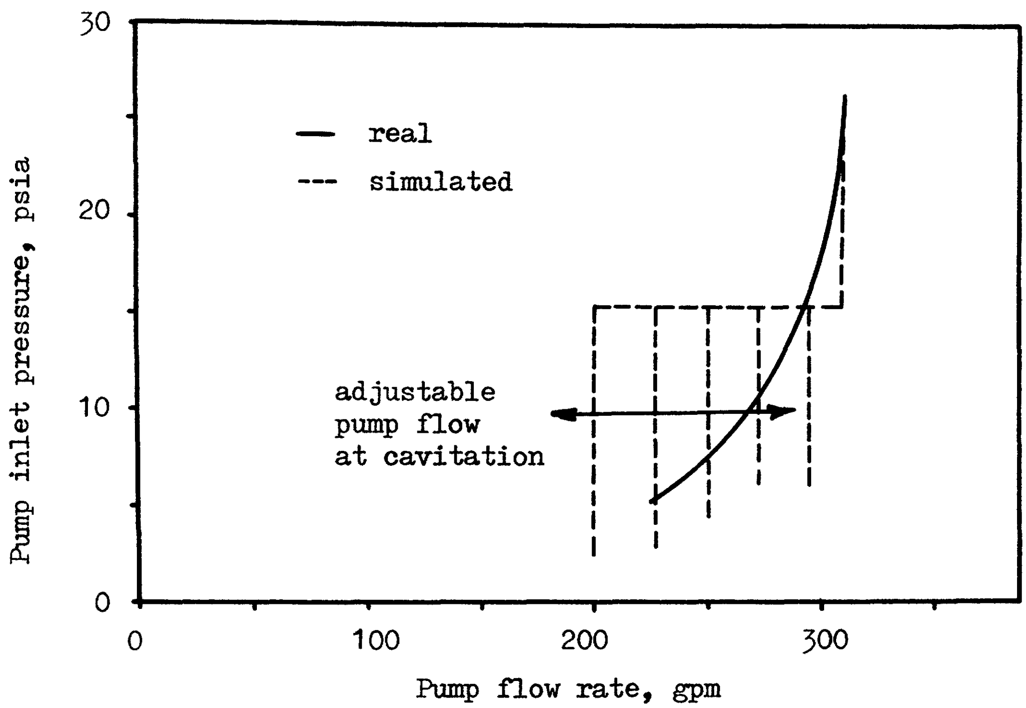


Fig. 6. Comparison of the pump cavitation performance with the simulated performance.

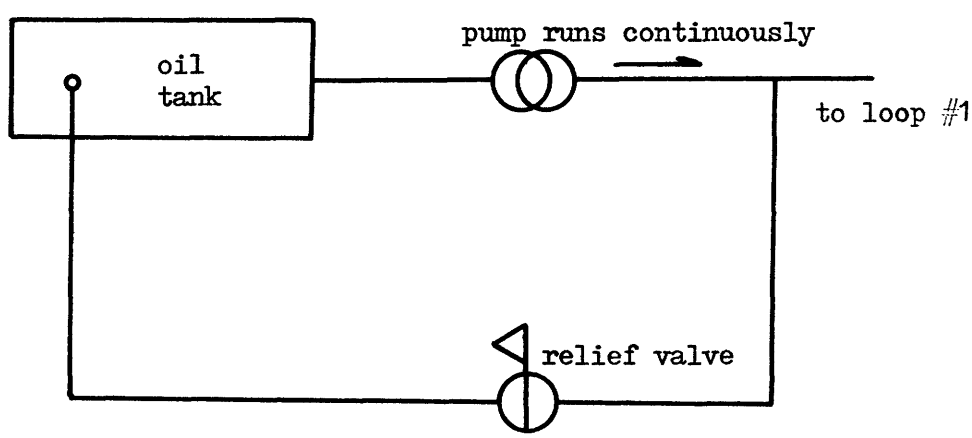


Fig. 7. Head tank system used on the Dunwoodie - Rainey installation.

until the pump head (voltage across the zener diode) equals the diode face value. Then the pump head remains constant and all additional flow is directed through the diode. The diode face value is equal to the maximum allowable pump head H_{max} divided by the conversion factor m (Sec. 4.2).

4.1.2. Pipe Flow At Steady State[9,11,15]

It is known that pressure changes along a pipe in steady, fully developed turbulent flow functionally depend on the Reynolds number and the relative roughness of the pipe. This unknown function is in practice known as the friction factor f. The friction factor is defined by:

$$f = \Delta p \,/\, (\,2\, L \rho v^2 \,/\, d\,) \tag{4-7}$$

For flow in circular pipes the Moody diagran (e.g. Ref. 11) can be used to determine f as a function of the Reynolds number. For the flow in pipe type cable systems an f versus Re correlation was obtained by Slutz et. al.[15]. Given the flow rate, the pressure drops are found from (4-7) and listed in Tab. 2 :

$$\Delta p = a\, q^2 \qquad a = 2 \rho L \,/\, d\, A^2 \tag{4-8}$$

Loop (i)	Q_i (gpm)	Δp (psi)			
		$(\Delta P_{ML})_i$	$(\Delta P_{DL})_i$	$(\Delta P_{HE})_i$	H_i
1	312	120	157	37	314
2	336	138	184	40	362
3	312	115	152	37	304
4	312	115	159	37	311
5	312	119	164	37	320
6	288	96	125	34	255

Table 2. Calculated pressure drops using specifications of Tab. 1.

A simple linear approximation gives:

$$\Delta p = a' \, q' \qquad a' = a \, Q \tag{4-9}$$

where a = nonlinear pipe flow resistance

a' = linearized pipe flow resistance

L = pipe length

A = effective pipe cross-sectional area

d = equivalent pipe diameter

v = flow velocity

q = fluid flow rate associated with a

q' = fluid flow rate associated with a'

ρ = fluid density

4.1.3. Head Tank Modeling

The head tank system consists of a pump which continuously sends pressurized oil from a reservoir through a valve back to the reservoir[3], and is sketched in Fig. 7. The valve is set for pressure needed for satisfactory system operation and this pressure is further called the HT pressure. When transverse flow from a HT exceeds the pump capacity, the HT pressure begins to drop. The model maintains a set pressure and the pressure drop due to the transverse flow is simulated manually.

4.1.4. Elevation Modeling

The increase of pressure level due to elevation was taken into account in a lumped model form in loops #1 and #3. The exact and approximated pressure increase due to elevation are compared in Fig. 8.

4.2. Electronic Components And Circuits Used In Modeling

The availability of electric and electronic components governed the selection of the scale factors relating voltage to pressure and current to flow rate. These factors are:

$$m = 165 \text{ psi / volt}$$
$$n = 6.1 \text{ gpm / } \mu a$$

Operational amplifiers were used extensively to operate the switch relays used for pump shutdown and in the cavitation model, and to provide lossless voltage outputs. A voltage comparator circuit is shown in Fig. 9. The absolute value of the output voltage of such a circuit is constant, but the voltage polarity abrupt-

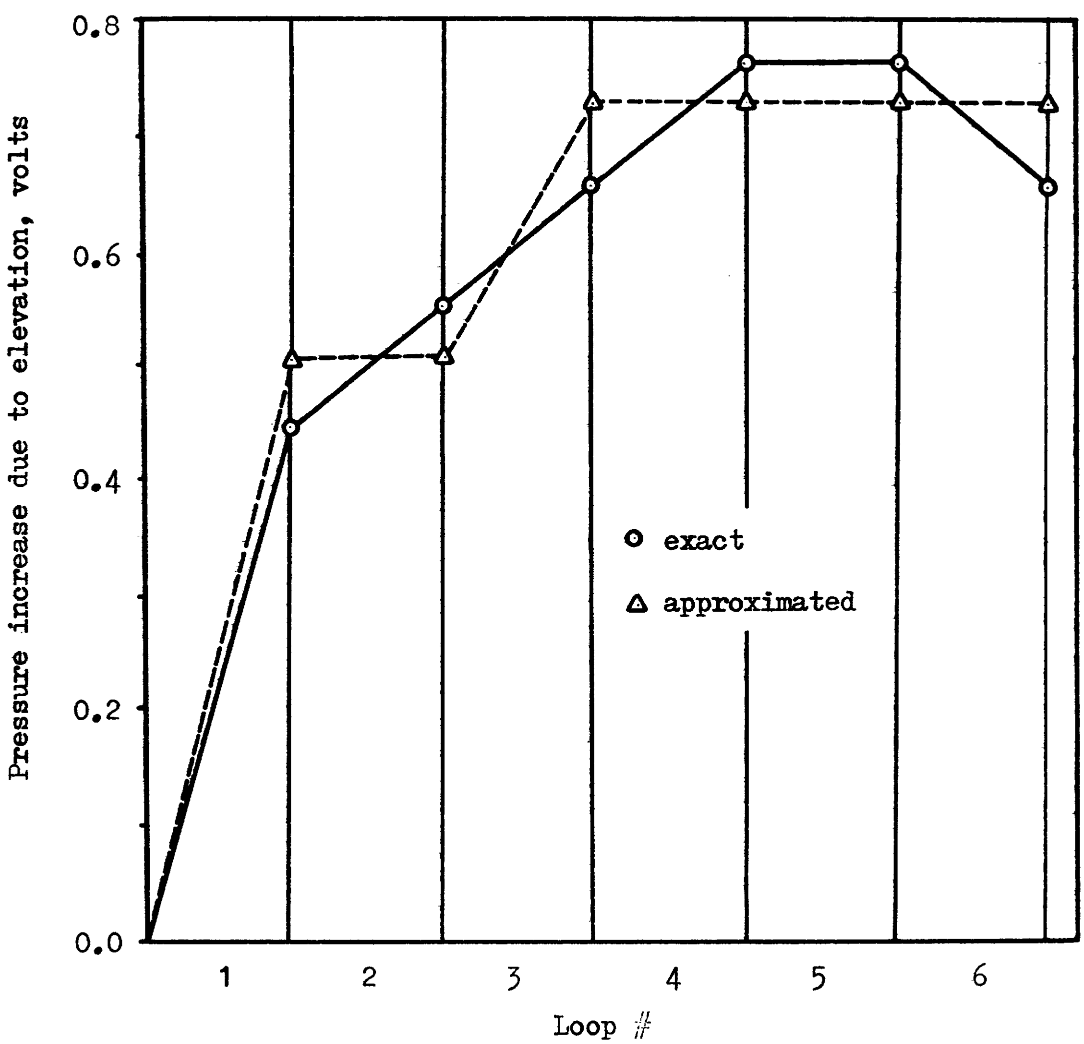


Fig. 8. Comparison of the exact and approximated pressure increase due to elevation.

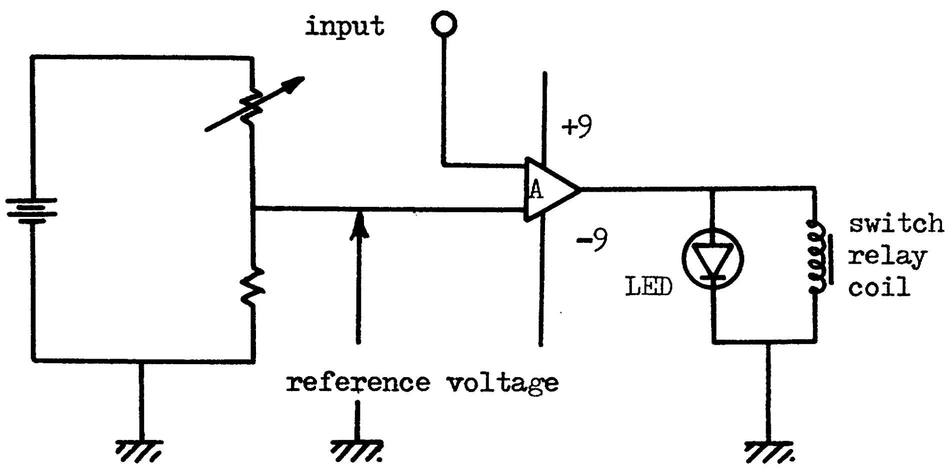


Fig. 9. Voltage comparator circuit.

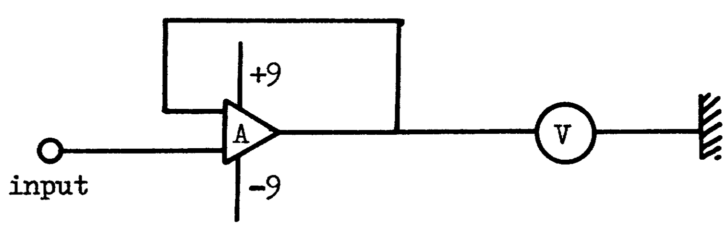


Fig. 10. Voltage follower circuit.

ly changes when the input voltage reaches a reference voltage. The operational amplifier output connected to a switch relay will then close or open the relay depending upon the operational amplifier input voltage. A voltage follower circuit is shown in Fig. 10. The output voltage of such a circuit is equal to its input, but the circuit draws a negligible amount of current ($\sim 10^{-9}$amps).

4.2.1. Pipeline Network Modeling

Linear variable resistors were used to represent each pipe segment, with the minimum resistance equal to that existing in the prototype under NOC. Using the pressure drop values of Tab. 2 and Eq. (4-9), the NOC flow resistance values were found and are listed in Tab. 3, after being multiplied by the factor of n/m, and after the resistance of the current meters and elevation modeling circuits were taken into account. Increase in pressure due to elevation was simulated by battery and resistor circuits placed in loops #1 and #3 as shown in Fig. 14.

4.2.2. Pump Modeling (Fig. 11 and Fig. 12)

A flow source and a resistor in parallel can equally be represented by a voltage source and the resistor in series, if linear

Loop	Line resitors				Battery Voltage
	(Ω)				(volts)
i	r_i	HE_i	DL_i	ML_i	V_i
1	390	10.0	39.0	28.8	12
2	645	10.0	41.9	31.3	23
3	390	10.0	37.2	28,8	12
4	390	10.0	39.0	28.8	12
5	390	10.0	40.1	28.8	12
6	270	10.0	33.0	24.8	8

Table 3. Model resistor and battery voltage values.

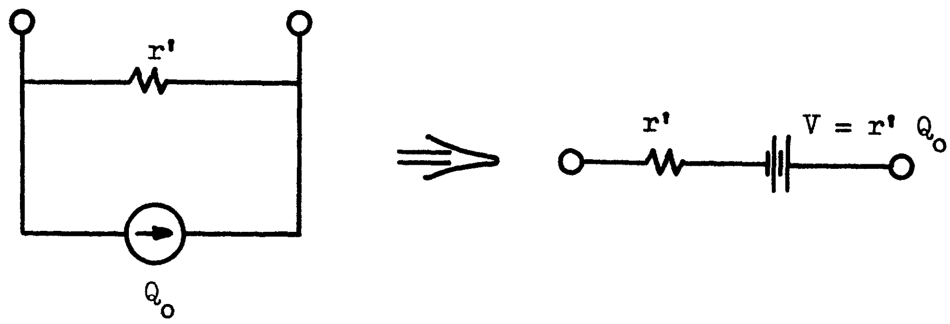


Fig. 11. Pump modeling.

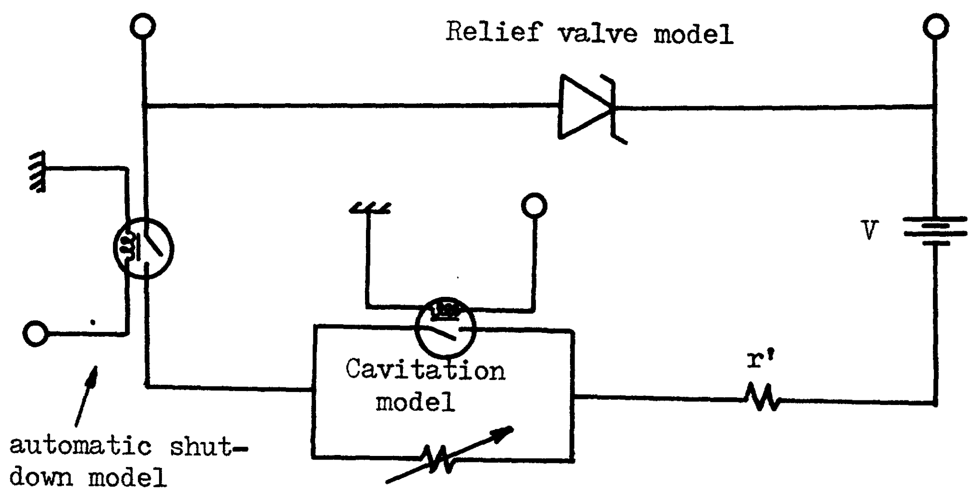


Fig. 12. Complete pump model.

relationship between pressure drop and flow rate is assumed. Thus the pumps are modeled by batteries and linear resistors in series. The leakage resistor values were found from (4-5) and are listed in Tab. 3, after being multiplied by the factor of n/m. The battery voltage values were found by using the Thevenin theorem:

$$V = r' Q \tag{4-10}$$

and are also listed in Tab. 3, after being multiplied by the factor of 1/m .

The face value of the zener diodes representing the pump relief valve was found by scaling the value of $H_{max} = 450$ psi by multiplying it by the factor of 1/m .

For the pump shutdown a switch relay was operated by a voltage comparator circuit which utilized either the pump inlet or discharge pressure as its input at the decision of the model operator.

As mentioned in 4.1.1, the pump cavitation was simulated by a variable resistor which could be adjusted for the desired cavitation extent. The resistor was added in series with the battery by the action of a switch relay operated by a voltage comparator circuit based on the pump inlet pressure (Fig. 9)

4.2.3. Head Tank Modeling

The model as shown in Fig. 13 has built-in all of the functions of the real HT system shown in Fig. 7 and described in Sec. 4.1.3. For satisfactory HT model performance, the electric current corresponding to the HT pump capacity should be much larger than the minimum current needed to cause a voltage drop across the zener diode to equal to its face value V_z. That is, it should be of the order of 20 - 30 ma. Since the maximum HT pump flow is only about 5 gpm, the conversion factor would have to be of the order of 0.2 gpm / ma. Since the pump flow rates are around 300 gpm, the model currents would have to be in the neighborhood of 1.5 amperes, with the need for correspondingly large batteries or power supplies. It was decided therefore, in order to be able to use regular size heavy duty batteries, to keep the current level down at 20 - 30 ma corresponding to the NOC flow rate of 300 gpm. Thus the model HT pressure will not drop when the transverse flow reaches the capacity of the HT pump ($\sim$5 gpm). It is very simple, however, to perform this function manually by changing the HT pressure setting in such a way so as to maintain the transverse flow at or below 5 gpm.

4.2.4. Voltage Measurements

For the voltage outputs, in order to limit current losses through

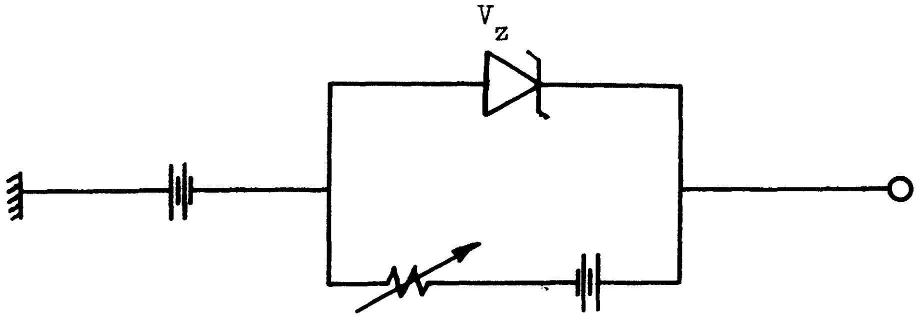


Fig. 13. Head tank model

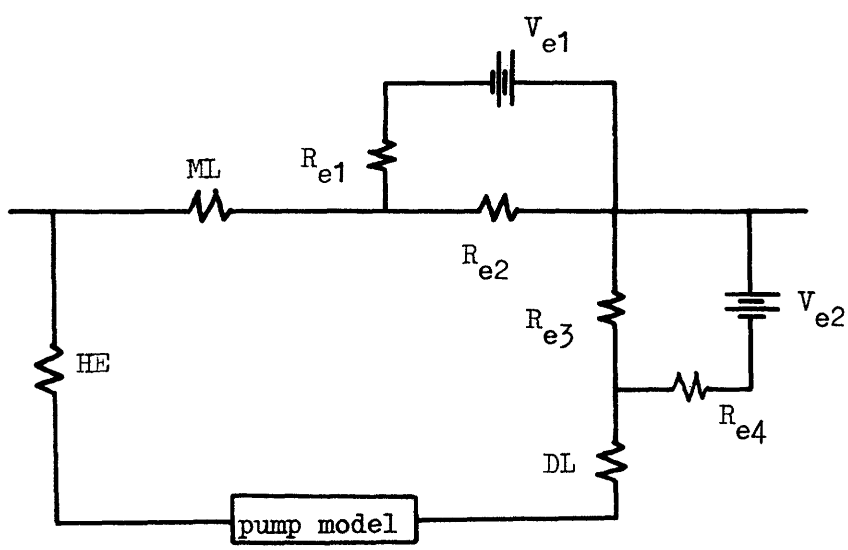


Fig. 14. Elevation modeling.

voltmeters, the voltage follower circuits were utilized as shown in Fig. 10. The voltmeter resolution was 0.02 volts.

4.2.5. Current Measurements

Each ML current was measured by microammeters and their internal resistance was included in the ML resistance. Since the current through the ML's was in the 20 - 30 ma range, the microammeters were connected across shunt resistors and therefore the scale factor n has the units of gpm / μ a.

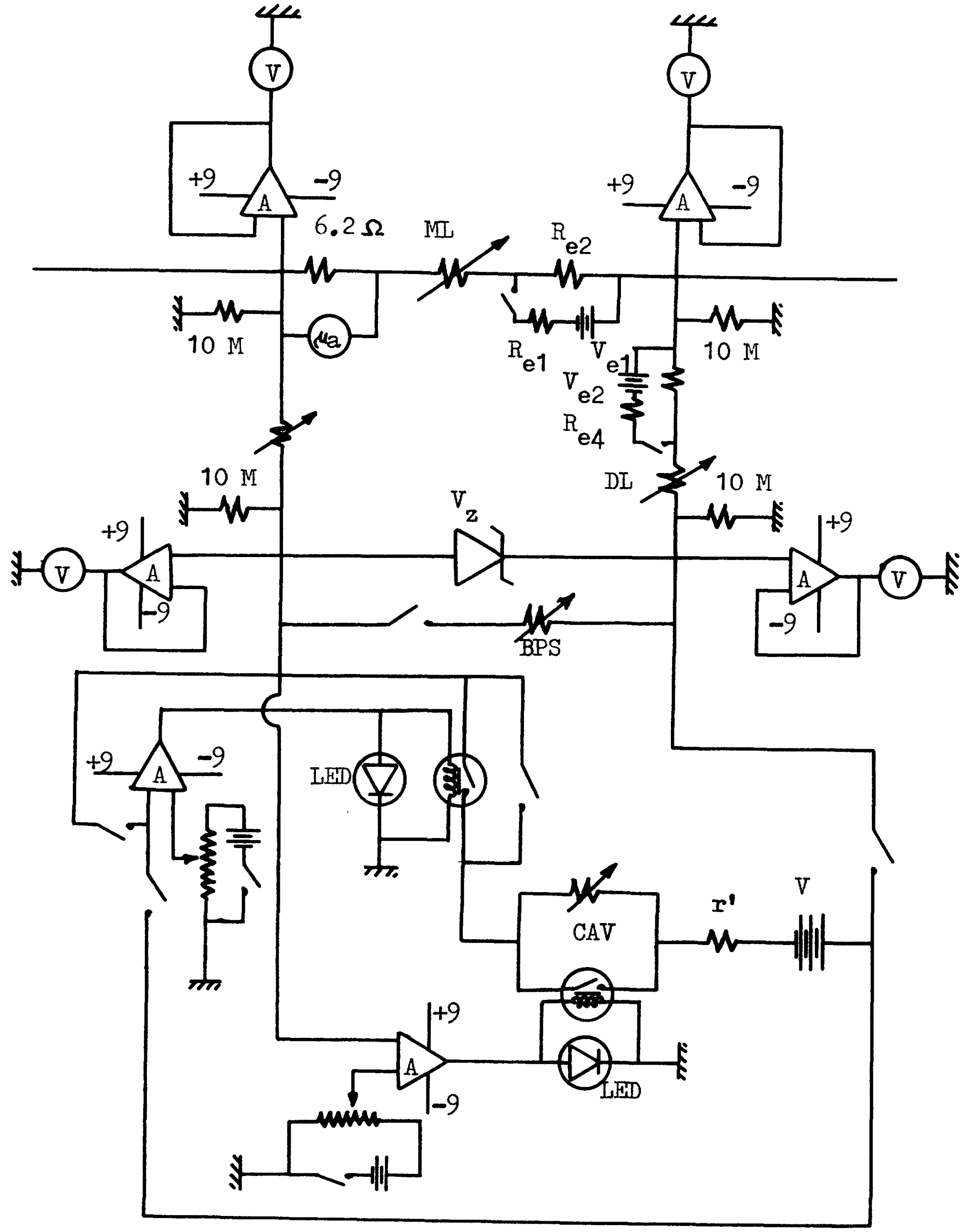


Fig. 15. Complete model of a unit loop.

5. EXPERIMENTAL PROGRAM AND RESULTS

Since only the Configuration 1 is applicable today (requires no flow blocking diaphragms for its operation), imbalance effects on this configuration were investigated in greater detail. An imbalance is defined as a pipe flow blockage or resistance increase, pump flow bypass, or a pump shutdown which alters the pressure profile or the flow rate in the system. The difference between the schemes using single HT (single pressure control — SPC) and two HT's (dual pressure control — DPC) is very small, since the HT pump capacity is only 5 gpm. When the pump capacity is increased and transverse flow allowed by building a return line, this difference may become significant, but from the simulation tests it was discovered that all pressure deviations from NOC due to a practical size blockage would be smaller, but not significantly smaller, in DPC than in SPC, everything else being equal. At the same time, the ML flow rate in loops having oposite ML flow direction to the direction of the transverse flow would be reduced by the amount of the transverse flow, whereas in loops with the ML flow direction in the direction of the transverse flow the flow rate would be increased by the same amount. An example is given in Fig. 16 (for better understanding, the reader may find it convenient to postpone the study of this figure until after he finishes reading Sec. 5.3).

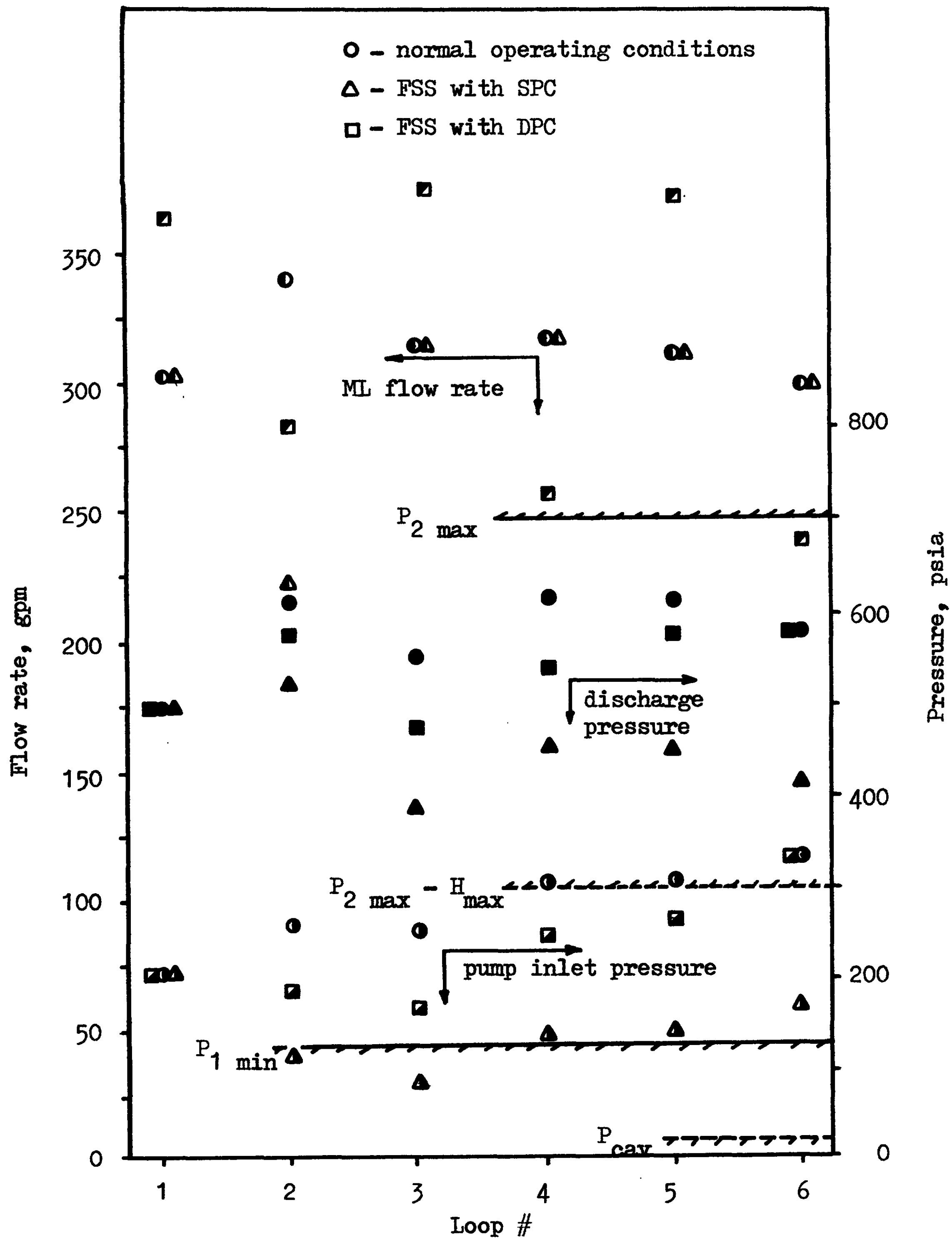


Fig. 16. Effect of ML2 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Conf. 1 with FSS (comparison of single pressure control — SPC, and dual pressure control — DPC).

The reduction of flow in some line segments is one of the major drawbacks of the DPC system. Another disadvantage is the return line requirement. The improvement of the pressure profile behavior is not large enough to justify an application of a DPC system and therefore only the results of analysis performed on Configuration 1 with a SPC are presented here.

5.1. Imbalance Effects on Configuration 1 With Constant Pressure Sources

As discussed in 3.2.2, a constant pressure source is a pump - relief valve arrangement in which relief valve maintains a constant pressure across the pump.

If a linear pressure drop - flow rate relationship is assumed, such a system can by easily analyzed by analytical means. Representing a constant pressure source by a battery and a flow resistance by a linear resistor, a loop can be represented by the diagram shown in Fig. 17. Only the main line pressure closest to the operational head tank (in SPC) is maintained constant and thus, since the flow direction changes from loop to loop, in order to calculate the system response to an imbalance, two cases must be considered:

(1) P_3 = const.

(2) P_4 = const.

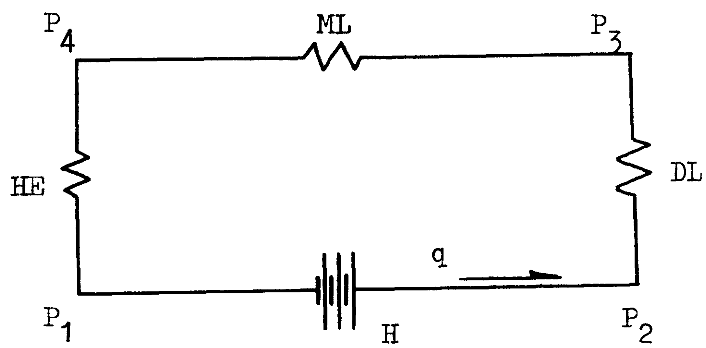


Fig. 17. Simplified model of a loop with constant voltage source during normal operating conditions.

5.1.1. Line blockages

From Fig. 17 and using the fact that pressure drop around a closed loop is zero:

$$(DL + ML + HE)\ q = H \tag{5-1}$$

(1) P_3 = const.

$$\begin{aligned} P_1 &= P_3 + q\ DL - H \\ P_2 &= P_3 + q\ DL \\ P_4 &= P_3 - q\ ML \end{aligned} \tag{5-2}$$

(2) $P_4 = \text{const.}$

$$\begin{aligned} P_1 &= P_4 - q\ HE \\ P_2 &= P_4 - q\ HE + H \\ P_3 &= P_4 + q\ ML \end{aligned} \tag{5-3}$$

The loops between the operating HT and the loop with the imbalance are unaffected, and in the rest of the loops the pressure level moves together with either P_3 or P_4 , whichever is not kept constant by the operational HT. The flow rate is reduced in the loop with the imbalance only, since SPC is used. The results are shown in Figures 17 - 47.

5.1.2. Pump shutdown

In case of a pump shutdown $q = Q$ and therefore $P_1 = P_2 = P_3 = P_4$. The effect on the other loops is in the form of a pressure level change in the loops on the side of the loop with its pump down and away from the operating HT. This pressure increase or drop is equal to the normal ML pressure drop in the loop containing the imbalance.

5.1.3. Pump bypass

The pump bypass can be implemented by further opening the relief valve and thus the pump - relief valve system is no longer a constant pressure source. To simplify the analysis it is assumed here that the amount of bypassed flow q_b is known.

Then

$$q = Q - q_b \tag{5-4}$$

Using (5-4) in (5-2) and (5-3) with HE, DL, ML being constant, the effect of a pump bypass on the system can be obtained.

Since cavitation has the same effect on the pump flow and head as the pump bypass, pump cavitation can be simulated by a pump bypass.

The results of the analysis in 5.1.1 - 5.1.3 are presented in Figures 17 through 47 and compared with the results obtained experimentally on the model representing the Configuration 1 with constant flow sources (3.2.3) and with the NOC pressures and flow rates.

5.2. Imbalance Effects On Configuration 1 with Constant Flow Sources

A constant flow source (3.2.3) is a pump - relief valve arrangement in which the pump head is allowed to vary up to a specified limit and the net flow rate is held approximately constant. An analytical evaluation of the performance of this system under the influence of an imbalance (blockage) would be much more involved than in the constant pressure source system, and therefore the imbalance effects were obtained experimentally by electric analogy simulation. The results are presented in Figures 17 through 47, and compared with the system of 5.1 and NOC.

5.3. Comparison of Effectiveness of Imbalance Control In Constant Pressure Source and Constant Flow Source Systems

Recall that only the Configuration 1 is a practical system configuration (Sec. 3.1), and only SPC (single pressure control — Sec. 3.2.1) is a practical HT arrangement. Thus it remains to choose from either the PSS (constant pressure source scheme — Sec. 3.2.2) or the FSS (constant flow source scheme — Sec. 3.2.3). It will be shown that, as expected, when the constant flow source scheme is used, the discharge pressure input is far superior to the inlet pressure input for the pump bypass ar shutdown, as discussed in Sec. 3.2.4. Either of the two pressures can be used as the input for the same purpose if the PSS is employed, since now the pump head remains constant.

Control of selected typical imbalances is outlined in Figs. 18 through 60. In each figure the response to a blockage type imbalance of the PSS and the FSS is compared with the NOC (normal operating conditions) existing before the imbalance was introduced. Only the inlet pressure, discharge pressure, and flow rate of each loop are recorded, since these are the parameters that can be observed and controlled on the actual system.

A proper imbalance control (head tank pressure adjustment or pump bypass) is initialized if an inlet pressure falls below $P_{1\ min}$ = 0.80 volts (130 psia) in order to maintain ML (main line) pressures above 150 psia. The 20 psi difference represents the minimum expected pressure drop across a HE (heat exchanger); normal HE pressure drop is 40 psi.

An imbalance control is also initialized when a discharge pressure exceeds $P_{2\ max}$ = 4.25 volts (700 psia). This value was obtained by adding the minimum expected pressure drop of 100 psi across the DL (discharge line) to the ML pipe pressure rating of 600 psia. A value of 200 gpm as the minimum allowable ML flow rate was assumed for finding both, the minimum expected HE and DL pressure drops. The limit on the minimum flow rate is necessary to assure that the ML pressure will not exceed the ML pipe pressure rating. This could happen in situations where the loop pressure level is very high (caused by an external imbalance) and so the DL pressure drop, decreased by the flow reduction, is not sufficient to keep the ML pressure within bounds. For example if the pump discharge pressure is 800 psia and if the flow rate is reduced from 300 to 200 gpm, the DL pressure drop will be decreased from 170 to, say, 120 psia. Then, clearly, the ML pressure will be 650 psia, exceeding the limit of 600 psia by 20 psi. Thus a pump must be shutdown if its discharge pressure exceeds $P_{2\ max}$ (= 700 psia) and if, at the same time, the ML flow rate of the same loop is below 200 gpm. The limit imposed on the flow rates would not be necessary if it were possible to measure the ML pressures and used them as input for the line pressure profile controls (HT pressure adjustment, pump bypass or shutdown).

Two other important pressure values are indicated on each figure: P_{cav} — corresponding to the pump cavitation pressure of 0.1 volts (15.3 psia), and $P_{2\ max} - H_{max} = 1.52$ volts (251 psia) corresponding to the maximum inlet pressure if the inlet pressure is used as the determining factor in initialization of the pump bypass or shutdown.

The response of the PSS system was obtained by methods outlined in 5.1.1 and 5.1.3, and the response of the FSS was obtained experimentally (Chap. 4).

To avoid possible misinterpretation of the following discussion, the automatic line pressure profile control action of the head tank and the pump bypass should be recalled here (for more detail see Sec. 3.2.2 and 3.2.4).

The HT pressure adjustment is initiated when any of the system pump inlet pressures falls below $P_{1\ min} = 130$ psia or any of the system pump discharge pressures exceeds $P_{2\ max} = 700$ psia, and then the HT pressure is proportionally adjusted to keep the previously out-of-line pressure within limits.

The pump bypass is initiated when the pump inlet pressure of the same loop is below $P_{1\ min} = 130$ psia or when the pump discharge pressure exceeds $P_{2\ max} = 700$ psia, and then the pump bypass is proportionally adjusted to maintain the previously out-of-line pressure within bounds.

And again, a pump is shut down if the discharge pressure of the same loop exceeds $P_{2\ max}$ and if the ML flow rate of this loop falls below 200 gpm.

Fig. 18 represents the effect of a blockage in HE1 equal to 100% of the normal HE1 flow resistance. As can be noticed, most affected is the loop #1 where the imbalance originates. The inlet and discharge pressures are both reduced for FSS as well as PSS. The loop #1 flow rate drops only slightly in FSS but is significantly reduced in PSS. Pressures in the rest of the system are slightly reduced, but flow rates remain unaffected. Since all pressures and flow rates are within their working limits, no control is necessary.

Fig. 19 shows the effect of increased HE1 blockage to 200% of the normal HE1 flow resistance. All system pressures and ML1 flow rates are further reduced and in the case of FSS the loop #1 inlet pressure is just below the minimum allowable value of $P_{1\ min}$.

In Fig. 20 the low inlet pressure in FSS of loop #1 caused by HE1 blokage of Fig. 19 is relieved by partially bypassing the pump #1. The control was needed for FSS only and therefore no correction was applied to PSS. The result is increased inlet pressure, decreased discharge pressure and reduction of ML flow rate in loop #1, and a slight reduction of the pressure level in the rest of the FSS system. The loop #1 flow rate in FSS after control remains to be significantly higher than the loop #1 flow rate in PSS.

In Fig. 21 the HE1 blockage of Fig. 19 was corrected by increasing the HT pressure. The result is increased pressure level in the entire system and no further reduction of flow beyond the blockage-caused reduction. Again, only FSS was controlled.

Fig. 22 represents the effect of further increased HE1 blockage to a very high 500% of the normal HE1 flow resistance. The reduction of all system pressures outside the #1 loop continues, and the loop #1 inlet pressure of FSS as well as PSS is below the allowed $P_{1\ min}$. In the case of FSS the pump #1 even cavitates. Recall, however, that the blockage size here is unrealistically large. Notice the huge #1 loop flow rate reduction in PSS.

Fig. 23 shows the correction of the HE1 blockage in FSS as well as PSS of Fig. 22 by partially bypassing the pump #2. The ML flow rate in loop #2 is below 200 gpm but the discharge pressure is still well below $P_{2\ max} = 700$ psia and therefore there is no need to shut down this pump. Notice that now, after imbalance correction of both FSS and PSS these two different schemes have equivalent line pressure profile and ML flow rates.

In Fig. 24 the 500% HE1 blockage in FSS and PSS of Fig. 22 was corrected by increasing the HT pressure. The superiority of HT pressure adjustment over the partial pump bypass is clearly demonstrated by Fig. 23 and 24.

Figures 25 - 27 represent the effect of increasing HE2 blockage on the system pump pressures and ML flow rates. No control was necessary.

Figures 28 - 30 show the effect of increasing HE3 blockage, and Figures 31 and 32 its control by pump #3 partial bypass and HT pressure adjustment, respectively.

Fig. 33 presents a variation to the method of partial pump bypass. The HE3 blockage of Fig. 30 is corrected by bypassing pump #2. The result is a large reduction of flow in loop #2 and increase of the pressure level in the system away from the HT and loop #2. No improvement over the conventional partial pump bypass has been achieved.

Figures 34 -39 show the effect of DL1 and DL2 blockage on the system pressures and ML flow rates. Again notice the large flow decrease in PSS as compared to FSS. Only the unrealistic DL2 blockage of 150% in FSS (Fig. 39) required correction of one of its discharge pressures (#2). Fig. 40 represents the control of this imbalance by partially bypassing pump #2. The HT pressure adjustment control of this blockage would also be possible, but is not presented here; for this purpose the HT pressure would be lowered by the amount equal to the difference between the discharge pressure of pump #2 (Fig. 39) and $P_{2\ max}$ = 700 psia.

Figures 41 - 43 represent the effect of increasing ML1 blockage on the system pressures and the ML flow rates. A control is necessary only for the large 200% blockage in FSS of Fig. 43. As in the HE and DL blockage-affected systems, the ML flow rates in PSS are significantly lower than in FSS.

Fig. 44 shows the correction of the line pressure profile of FSS of Fig. 43 by lowering the HT pressure setting. Notice that some discharge pressures are now very close to $P_{2\ max}$ and that the inlet pressure of pump #1 is at $P_{1\ min}$. This situation suggests that a further increase of the ML1 blockage beyond the 200% could not be controlled by the HT pressure adjustment alone. Additional partial pump bypass would have to be used on the pump #1. It should be pointed out here, again, that the pump bypass action is fully automatic and is activated by high discharge pressure or by low inlet pressure of the pump; the HT pressure adjustment is also fully automatic and is activated when one of the line pressures deviates outside the limits.

In Figs. 45 - 47 the effect of increasing ML2 blockage is shown. Notice again the large flow reduction in PSS.

Fig. 48 shows how the ML2 blockage of Fig. 47 can be corrected by increasing the HT pressure, or equivalently, by switching the pressure control from the left HT to the right one (attaching HT to loop #6).

Further examples of blockage effects on FSS and PSS can be found in the Appendix B.

Conclusion:

In general, the effects of odd numbered HE and DL blockages on the system pressures, which are separated from the HT by the blockage, are identical, and the same is true for even numbered HE and DL blockages. The difference between HE and DL blockages is in the effect on the loop containing the blockage; HE blockage reduces both pump pressures, whereas DL blockage lowers inlet pressure and raises discharge pressure. The pressure imbalance propagation outside the loop of its origin is more pronounced in the case of ML blockages than for other blockages; for flow rate reduction it is immaterial, however, where the blockage has occured — all blockages have the same effect on the loop flow rates as long as the blockage absolute sizes are equal.

All examples, and most of all Fig. 48, clearly demonstrate, that the further an imbalance is from the operational HT, the smaller effect it can have on the entire system.

All examples show that the discharge pressure input is by far superior to the inlet pressure input for the line pressure profile control of FSS; in fact here some NOC pressures are already above $P_{2\ max} - H_{max}$.

The HT pressure adjustment is superior to the partial pump bypass in every respect, but nevertheless the pump bypass is also an effective line pressure profile control, and applied simultaneously the two methods are able to control almost any blockage size.

The most important result of all is that , in general, pressures within the loop containing an imbalance deviate more from NOC in FSS than in PSS, HE and DL blockages in FSS have smaller effects on other loops than in PSS, and ML blockages in FSS have larger effects on other loops than in PSS; the flow rate reduction is much larger in PSS than in FSS. For practical imbalance sizes (probably only the first step in each of the types of blockage increase) FSS does not require more frequent or extensive line pressure profile control than PSS, but maintains ML flow rates much closer to NOC. Since, in order to provide ample cable cooling, it is necessary to maintain the flow rate at the highest possible level, the FSS is clearly the superior line pressure profile control scheme to the PSS.

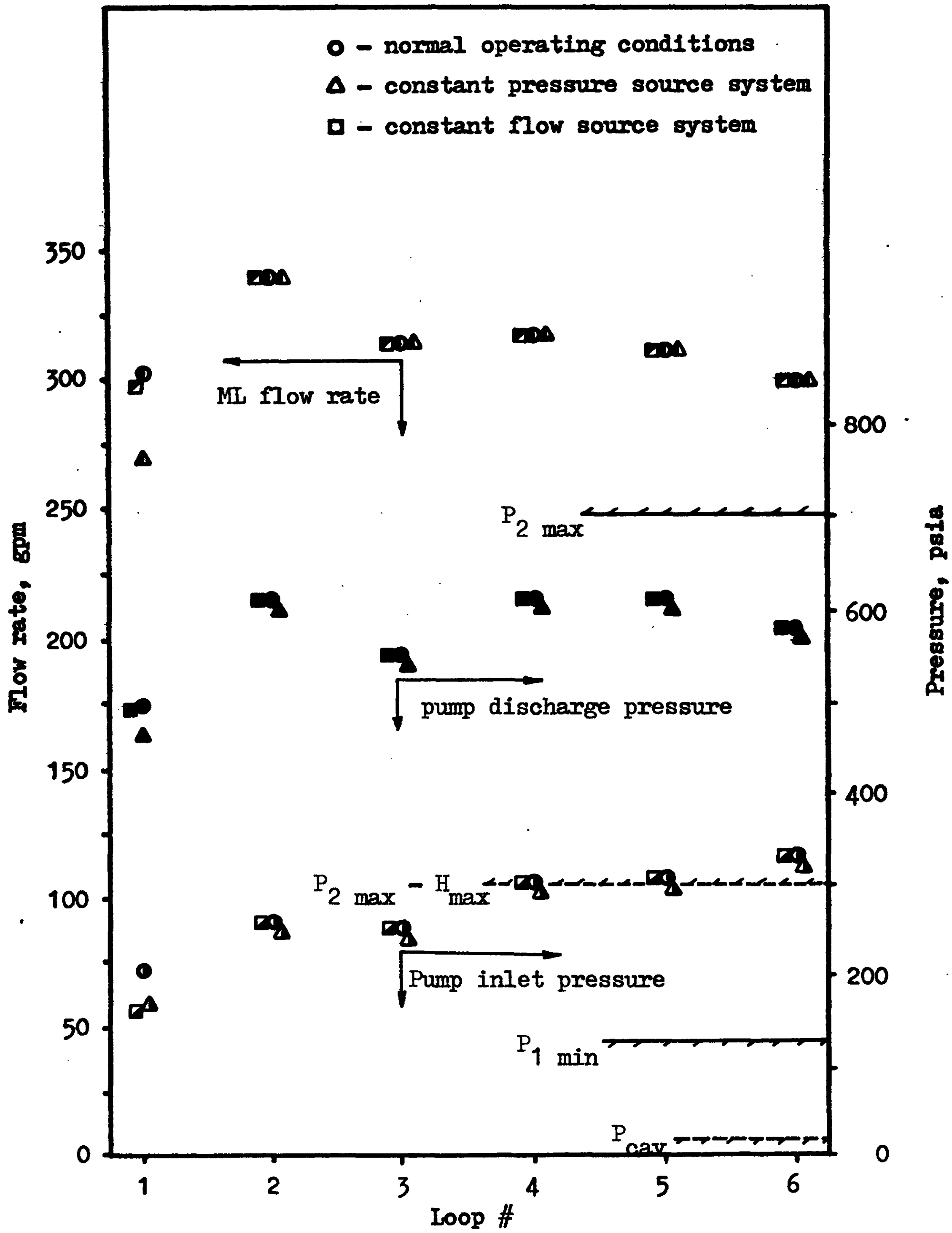


Fig. 18. Effect of the HE1 flow resistance increase of 100% on the inlet and discharge pressures and the ML flow rate in Conf. 1 with single pressure control at the line left end (HT attached to loop #1).

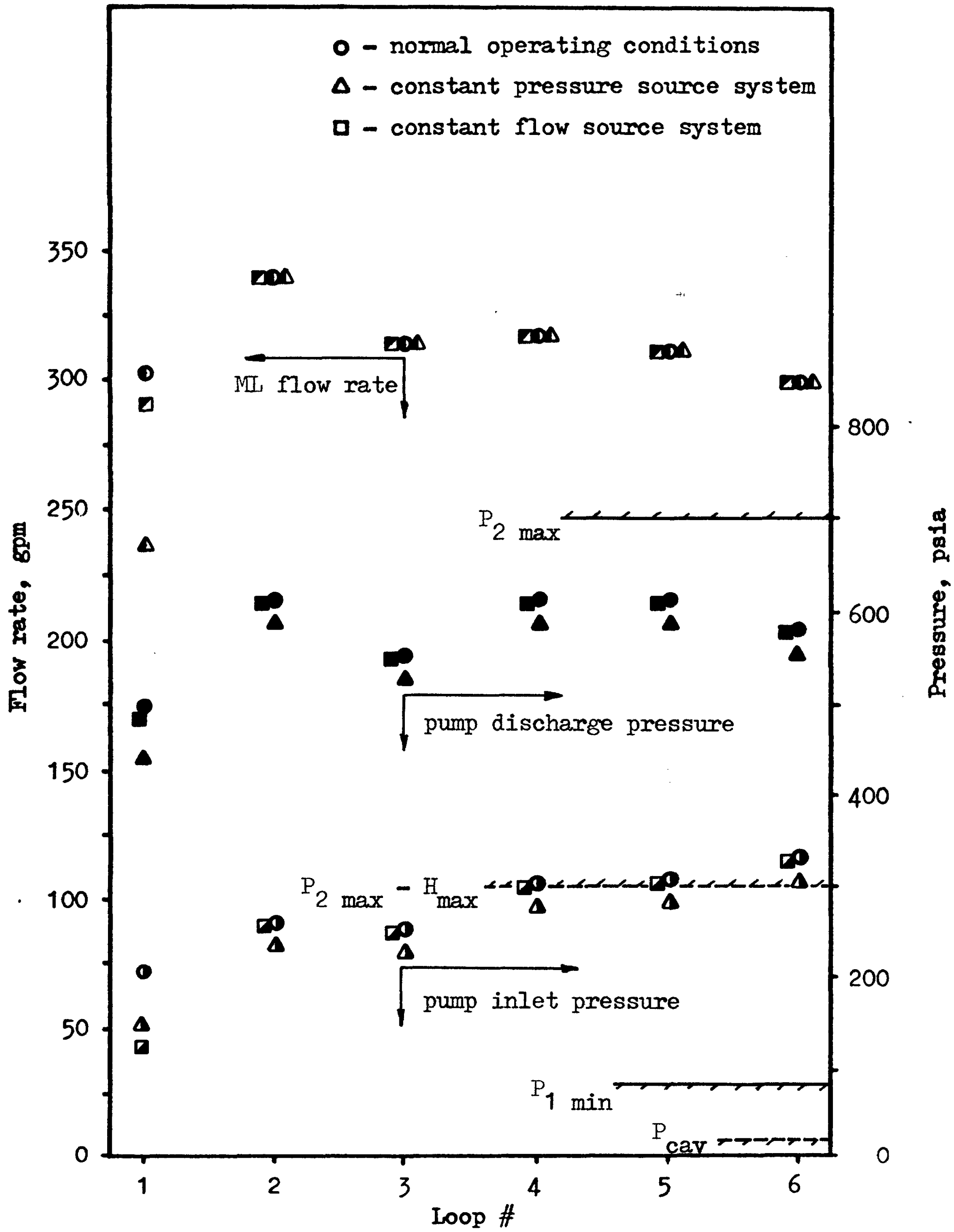


Fig.19. Effect of the HE1 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

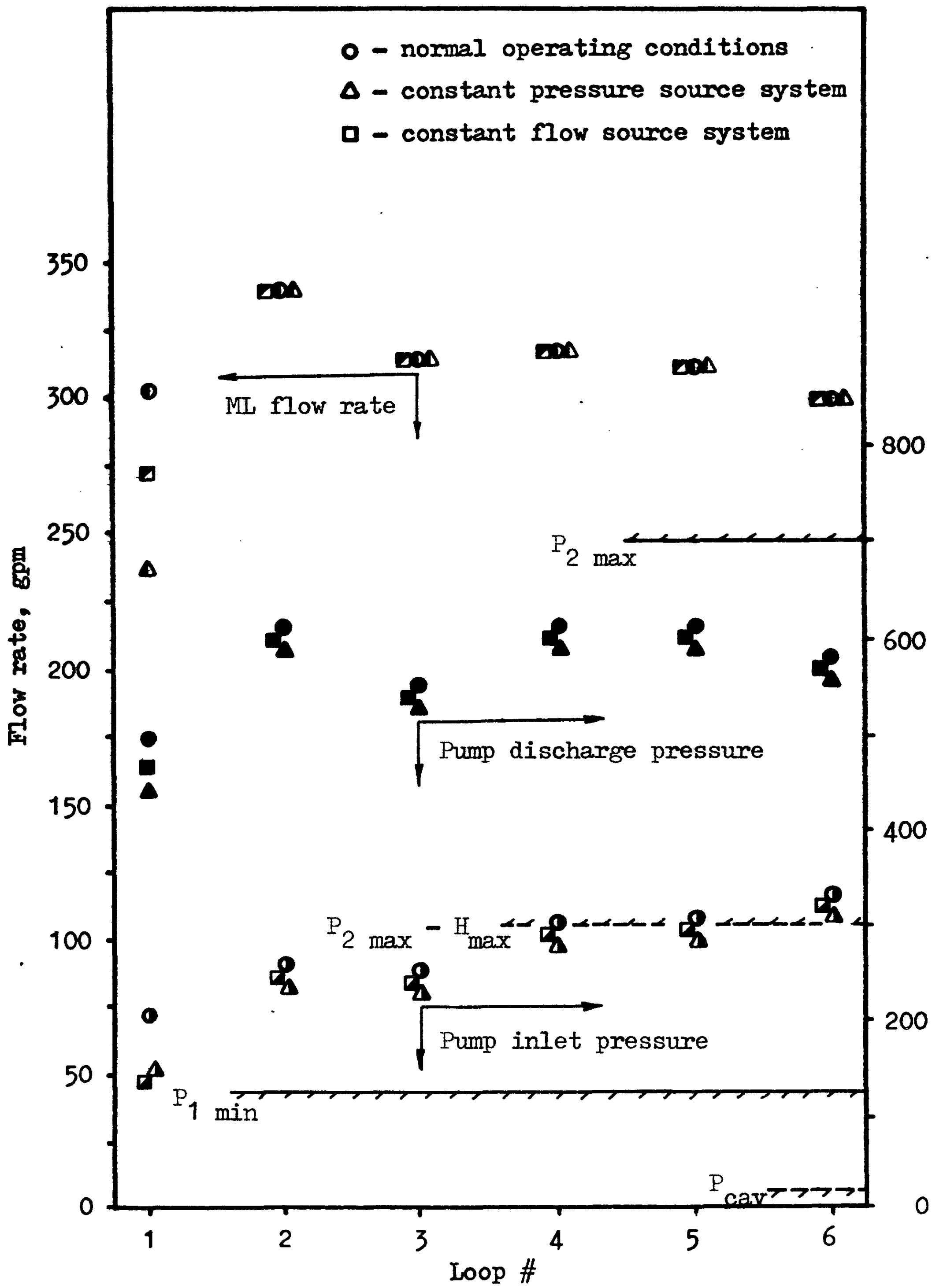


Fig. 20. Correction of the HE1 blockage (200% of normal HE1 flow resistance) of Fig. 19 by bypassing pump #1.

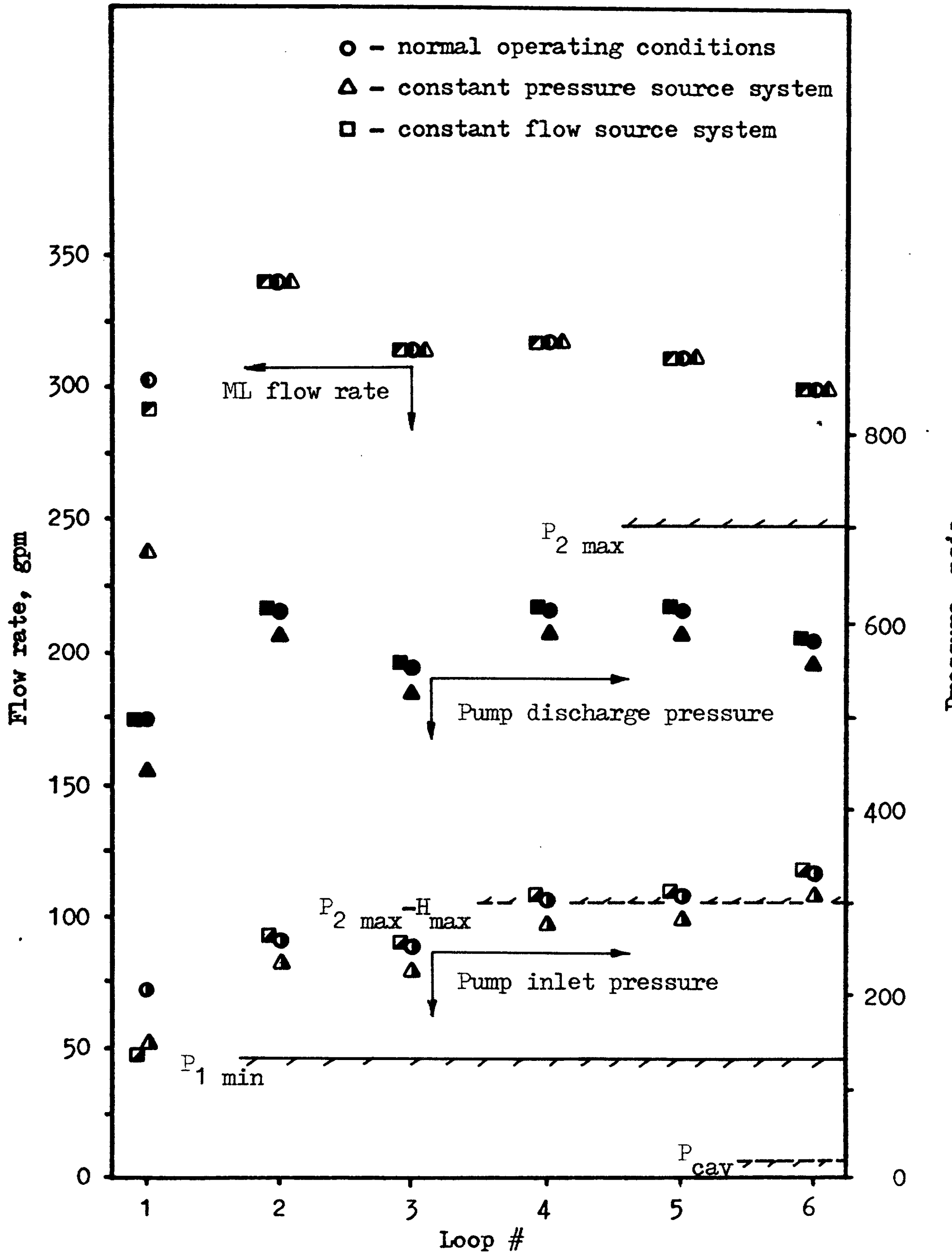


Fig. 21. Correction of the HE1 blockage (200% of normal HE1 flow resistance) of Fig. 19 by increasing the HT pressure (HT attached to loop #1)

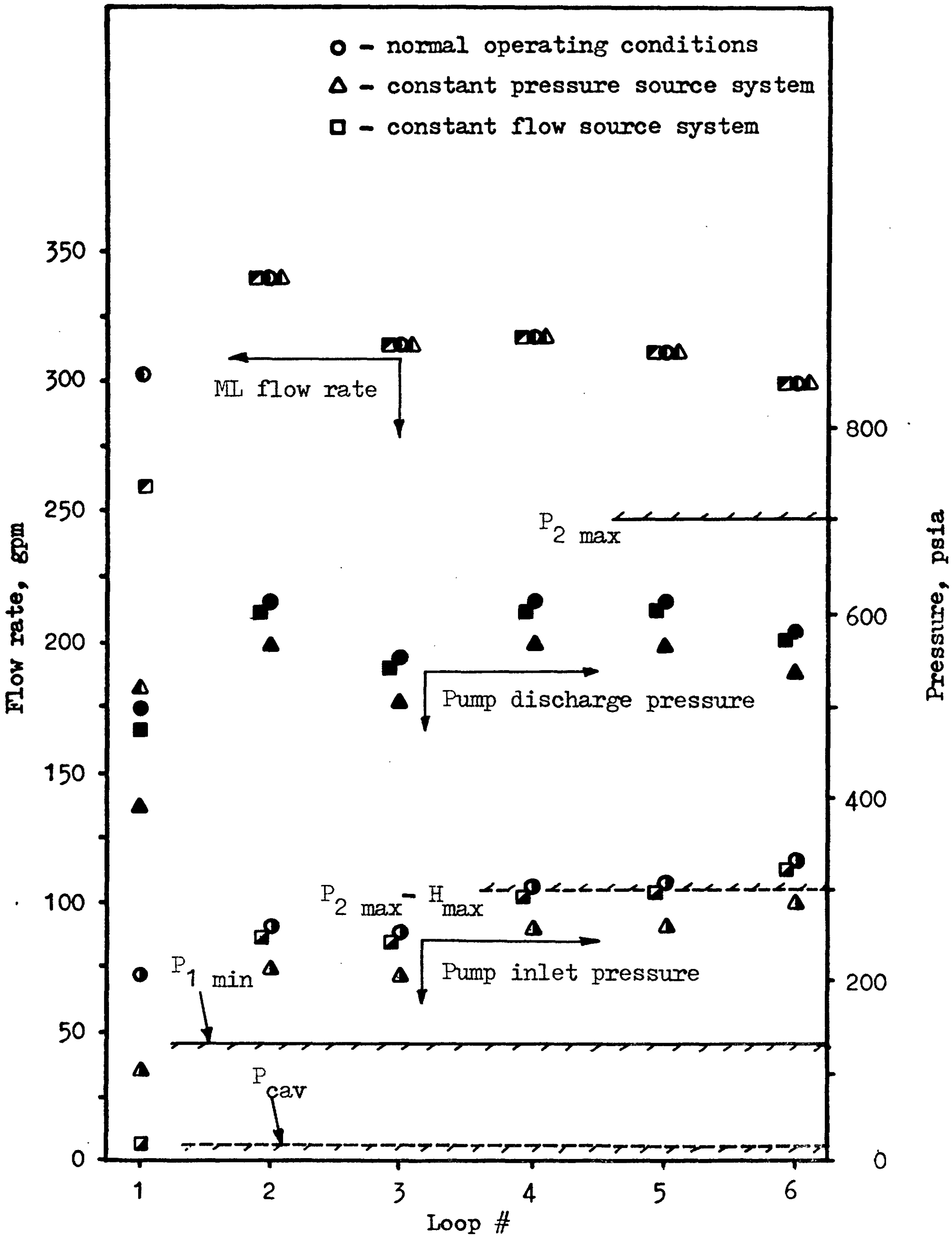


Fig.22. Effect of the HE1 flow resistance increase of 500% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

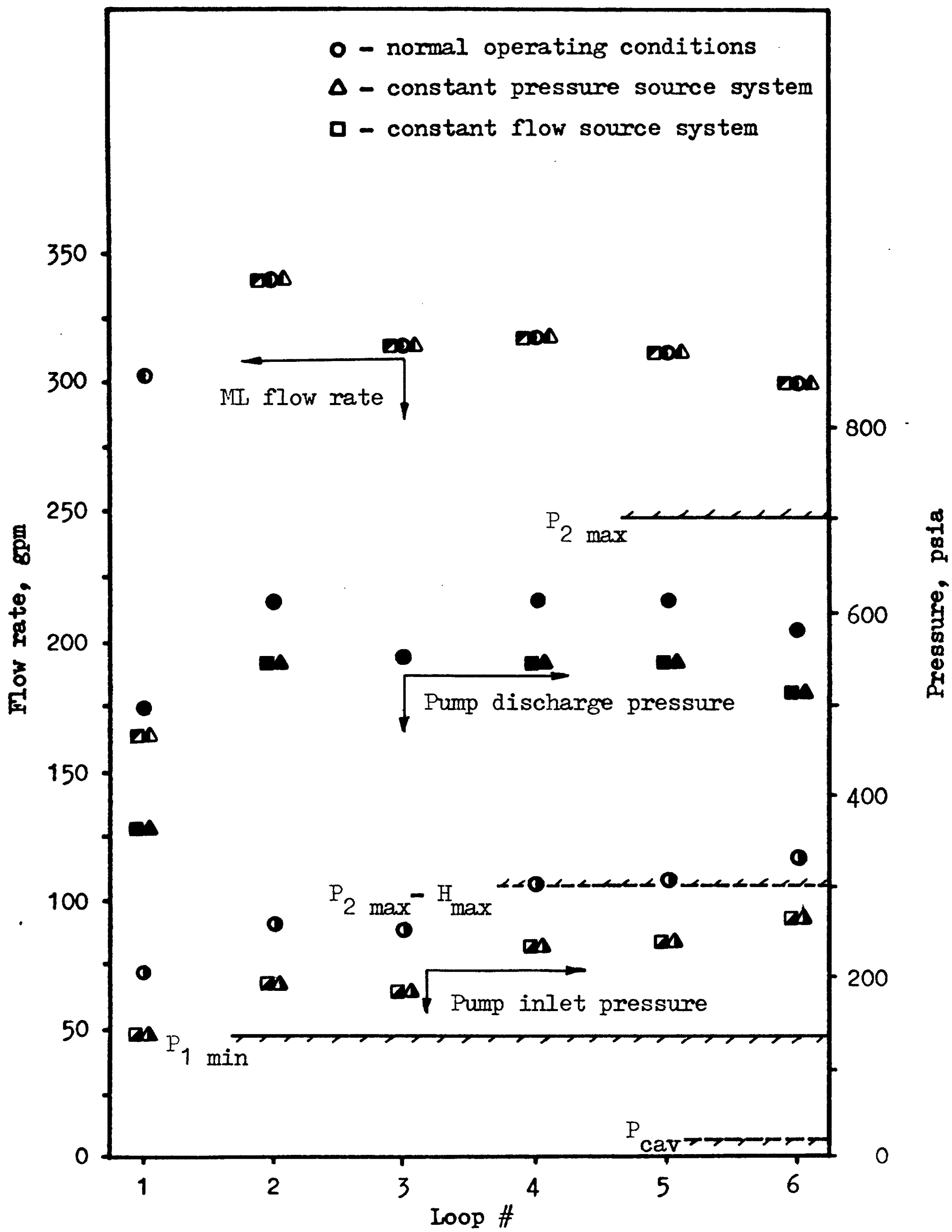


Fig. 23. Correction of the HE1 blockage (500% of normal HE1 flow resistance) of Fig. 22 by bypassing pump #1.

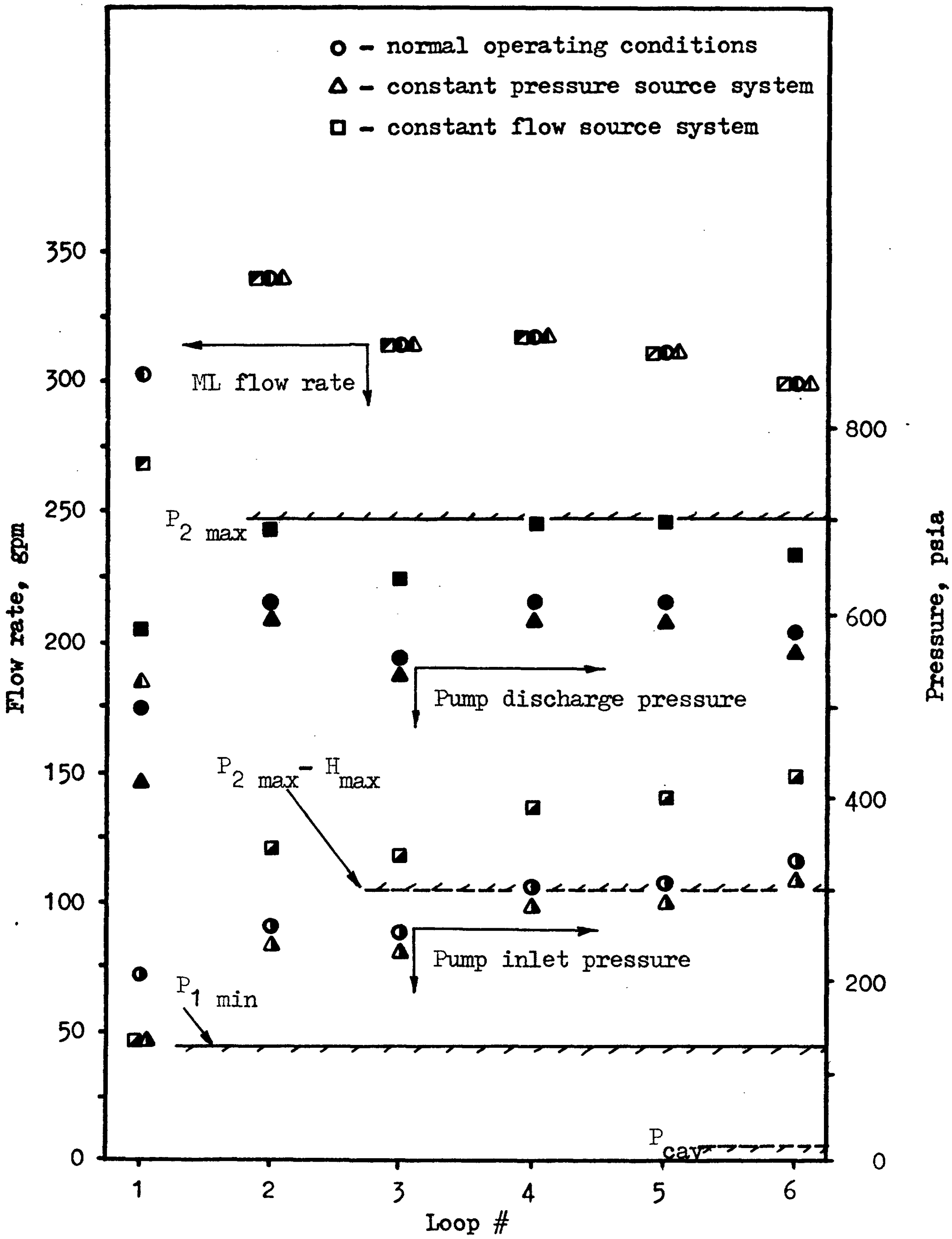


Fig. 24. Correction of the HE1 blockage (500% of normal HE1 flow resistance) of Fig. 22 by increasing the HT pressure (HT attached to loop #1).

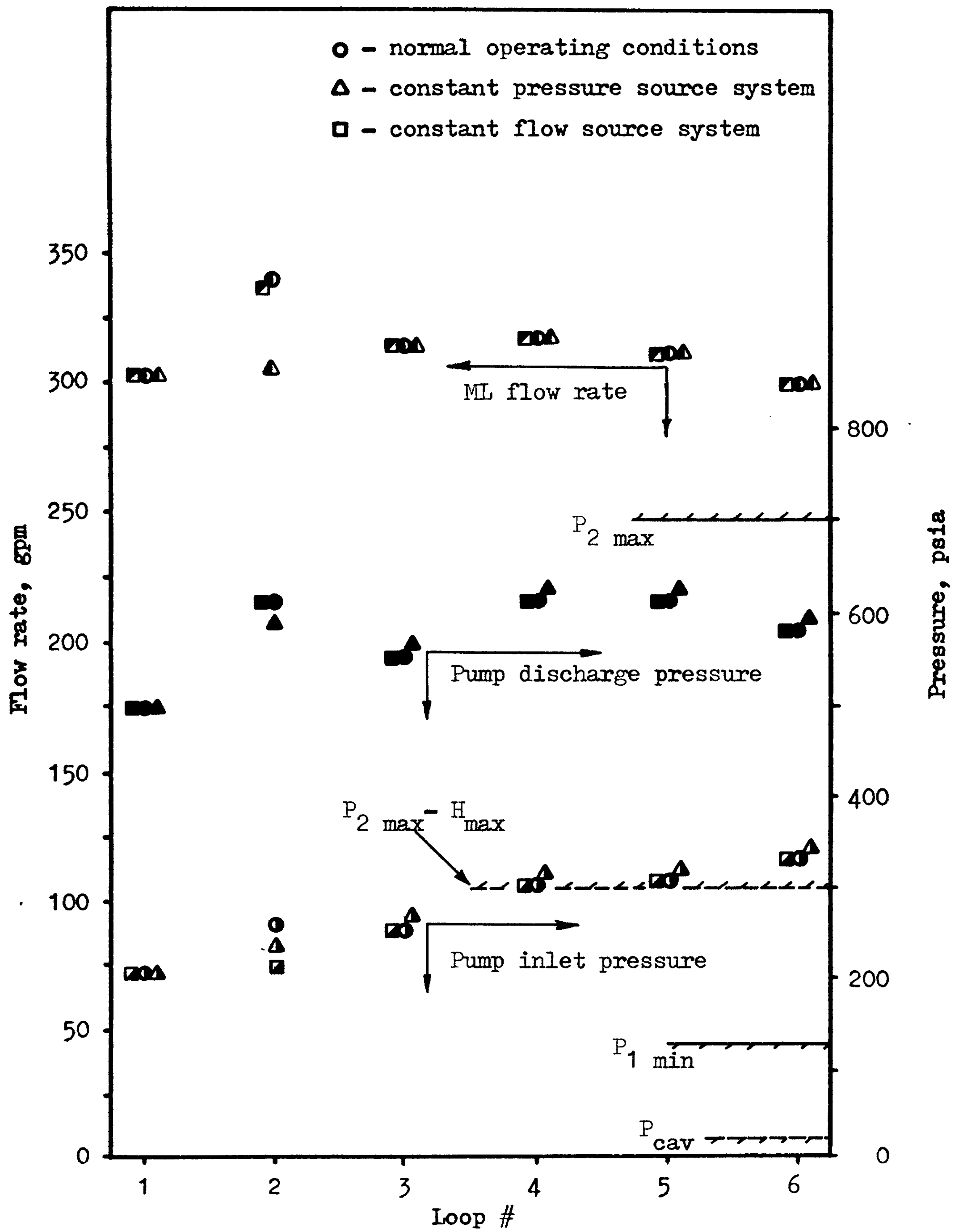


Fig.25. Effect of the HE2 flow resistance increase of 100% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

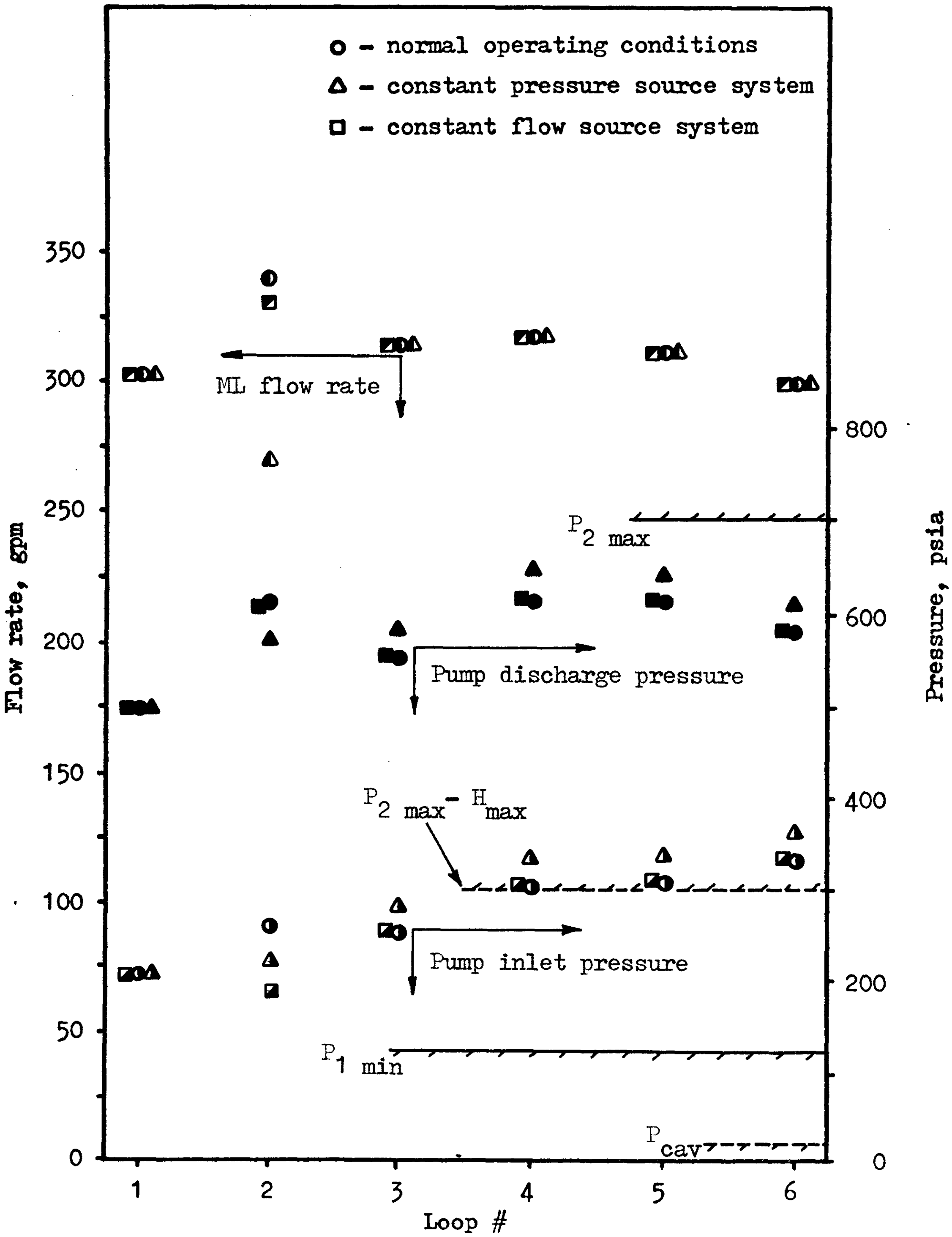


Fig. 26. Effect of the HE2 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

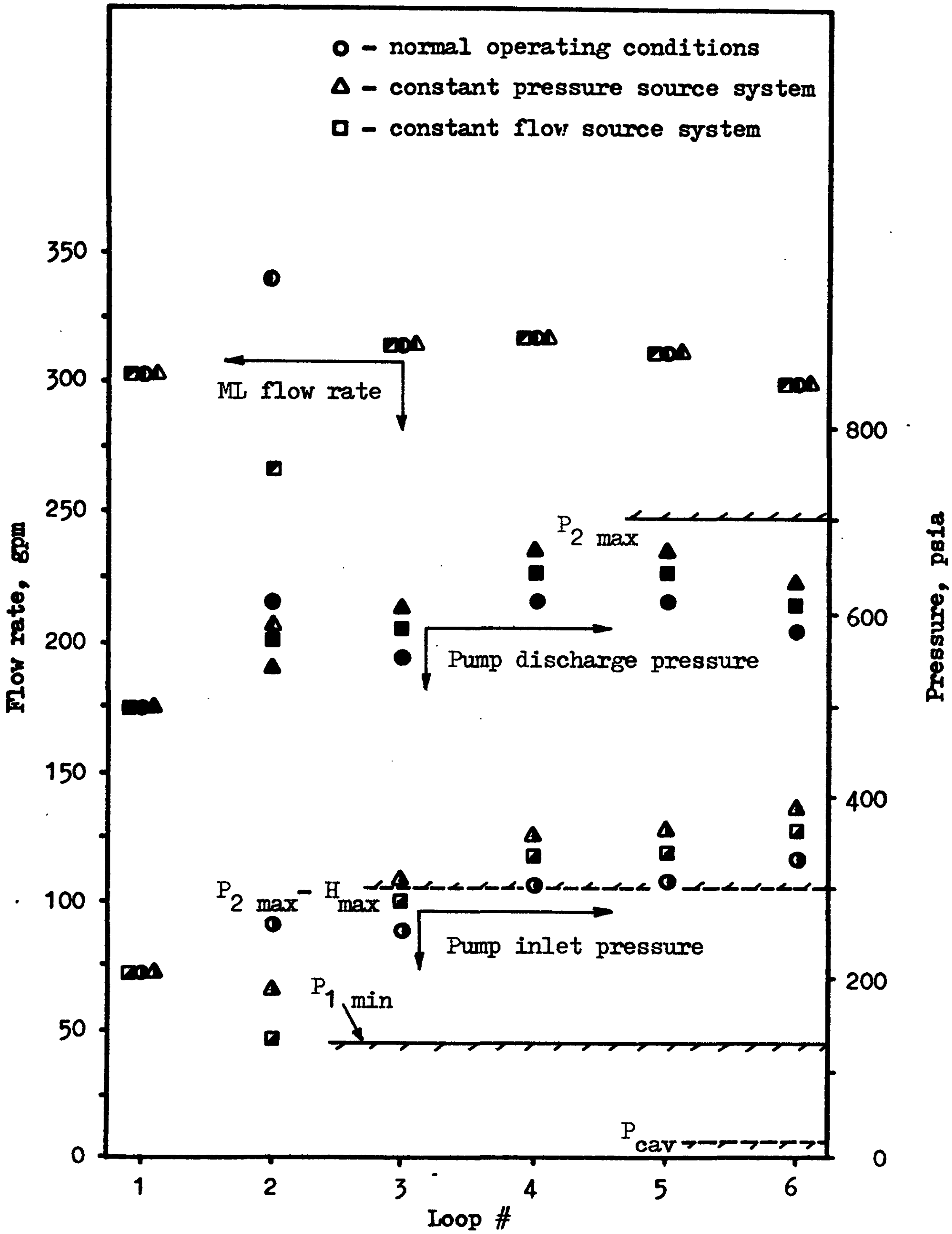


Fig.27. Effect of the HE2 flow resistance increase of 500% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

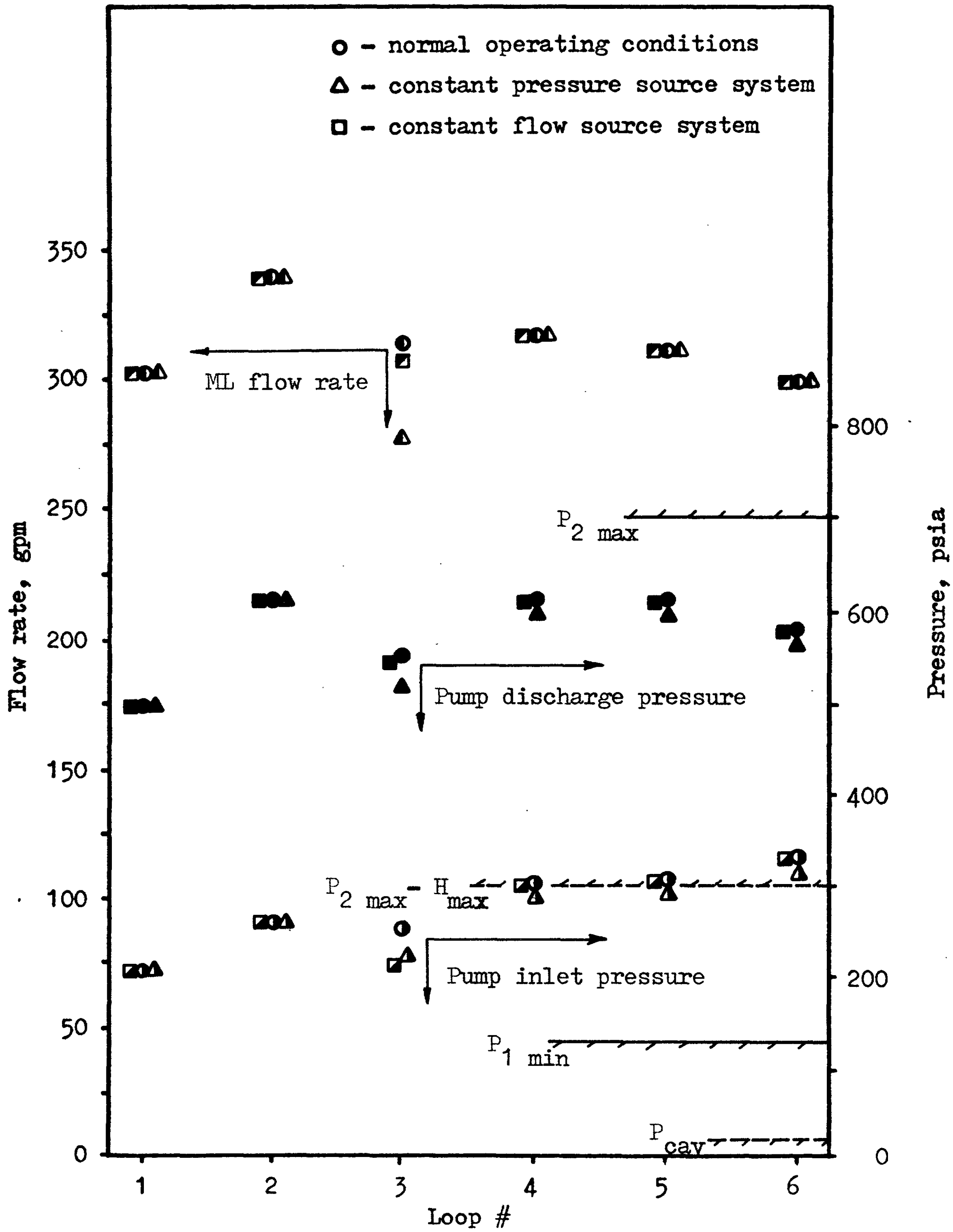


Fig. 28. **Effect of the** HE3 **flow resistance increase of** 100% **on the inlet and discharge pressures and the** ML **flow rate in Config. 1 with single pressure control at the line left end** (HT **attached to loop #1).**

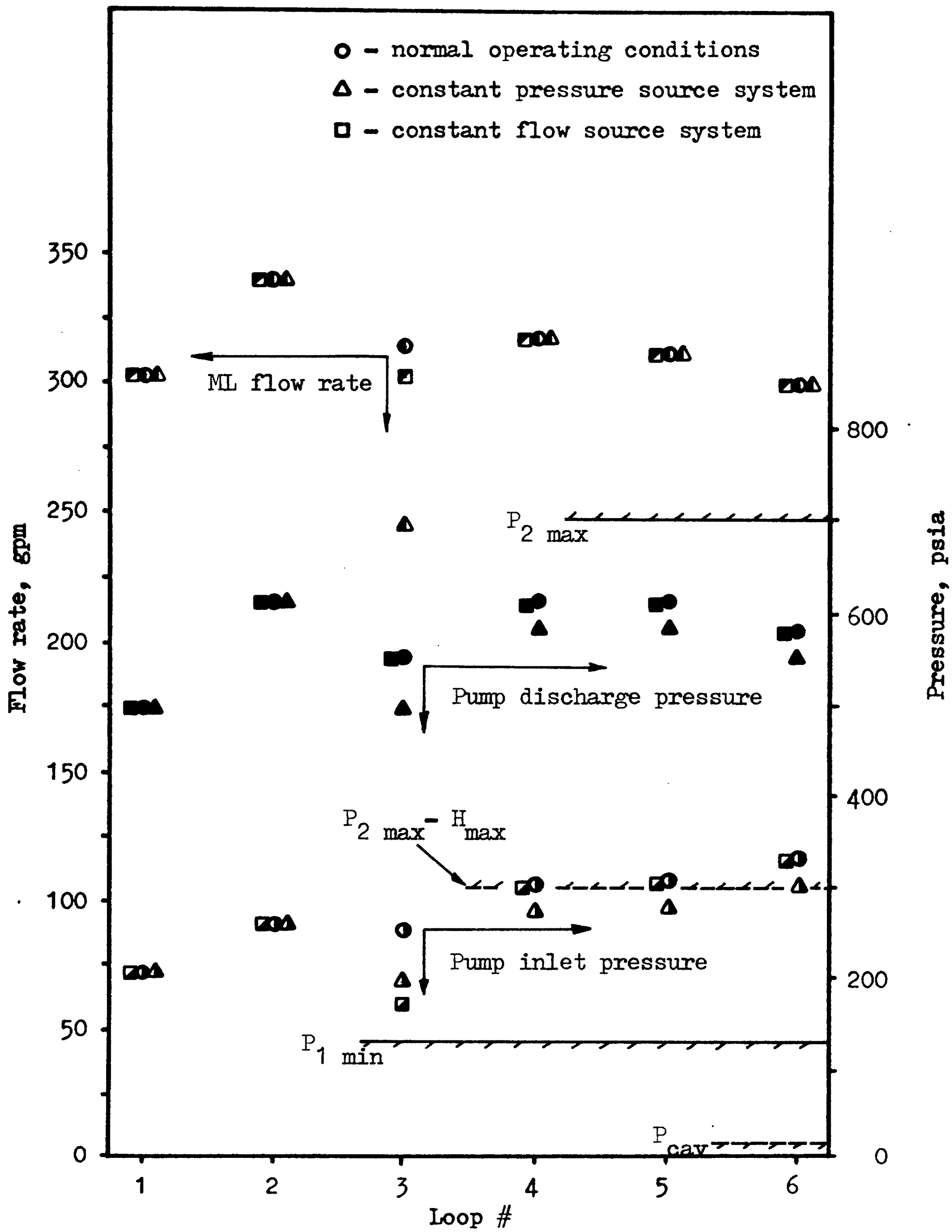


Fig. 29. Effect of the HE3 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

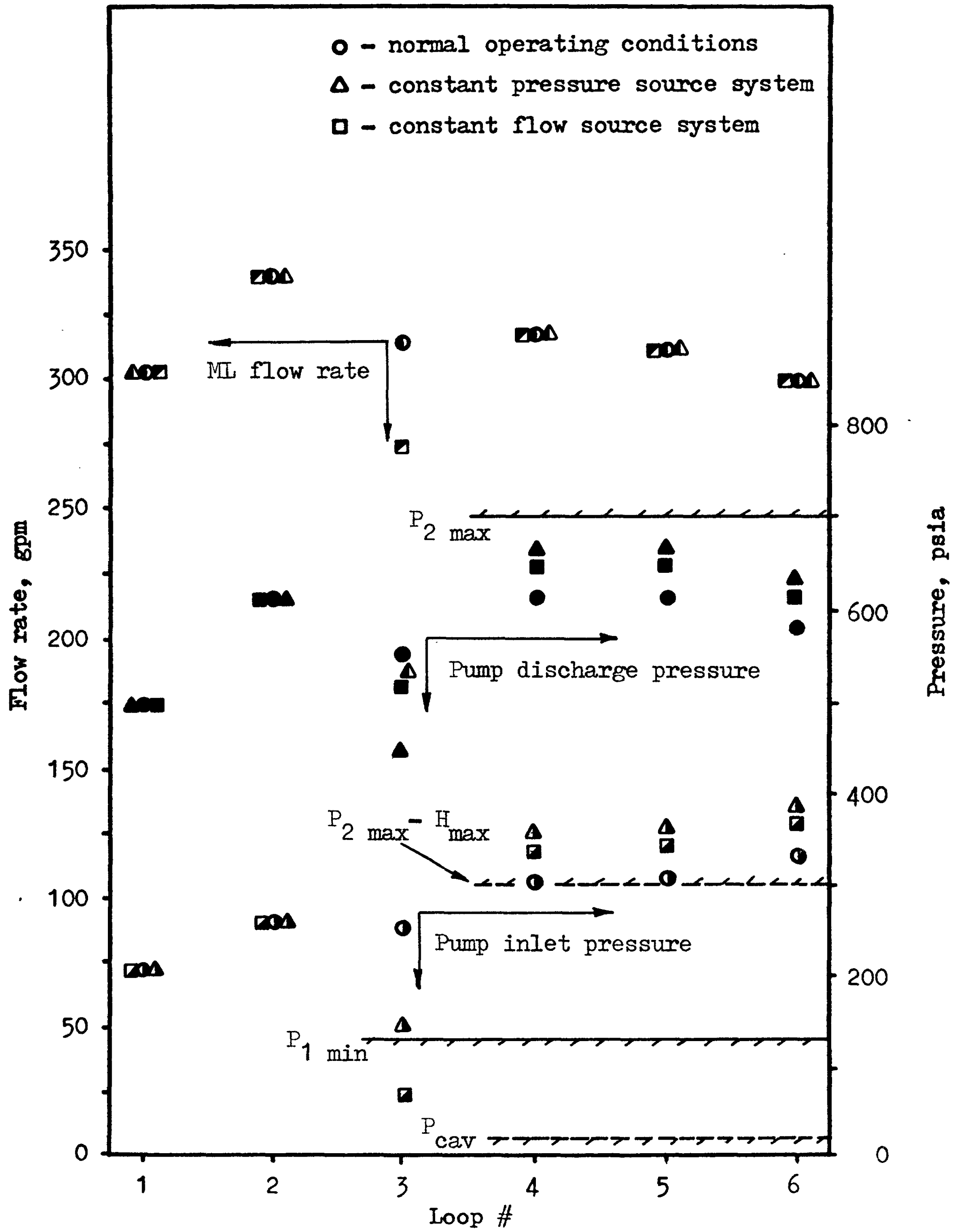


Fig. 30. Effect of the HE3 flow resistance increase of 500% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

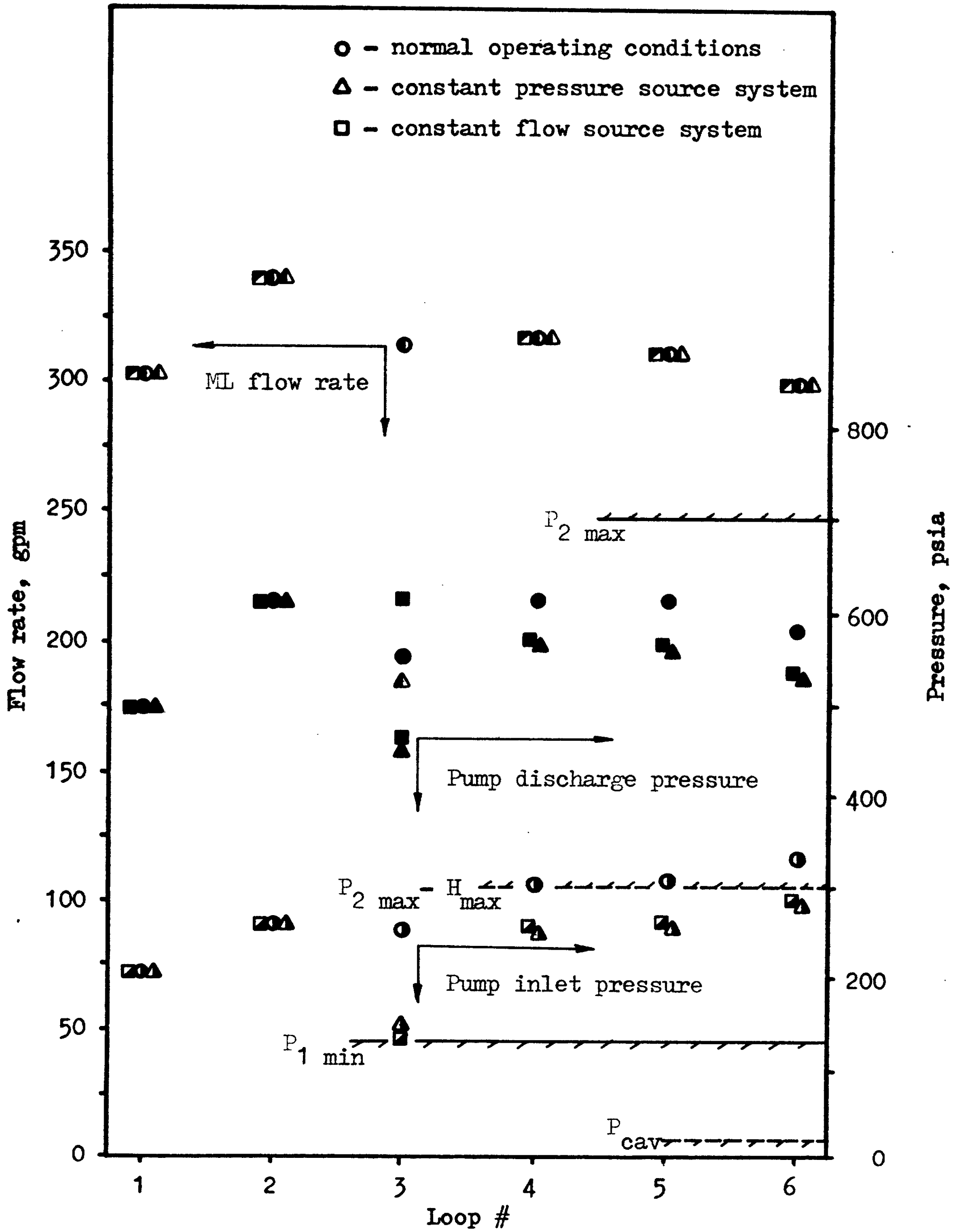


Fig. 31. Correction of the HE3 blockage (500% of normal HE3 flow resistance) by bypassing pump #3.

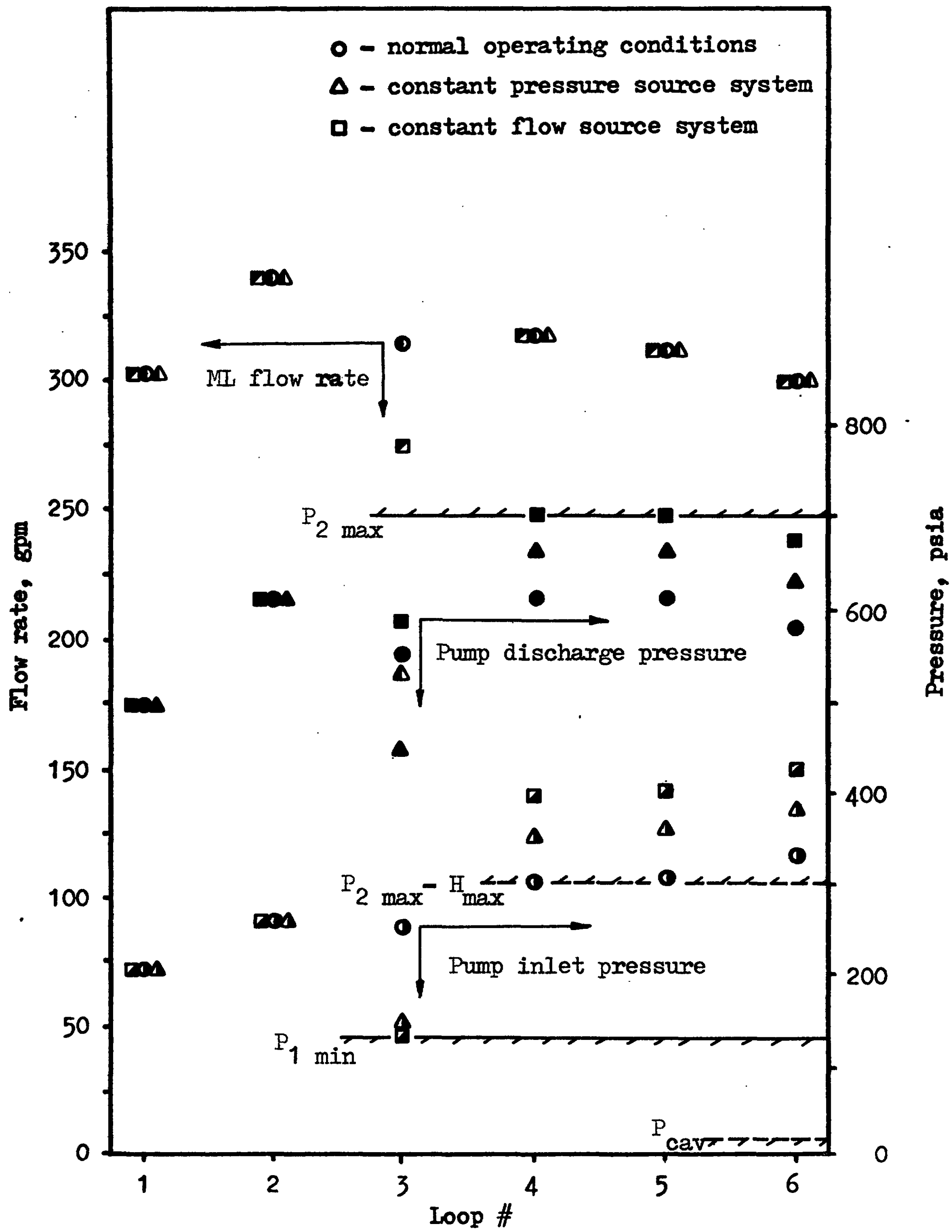


Fig. 32. Correction of the HE3 blockage (500% of normal HE3 flow resistance) of Fig. 30 by increasing the HT pressure (HT attached to loop #1).

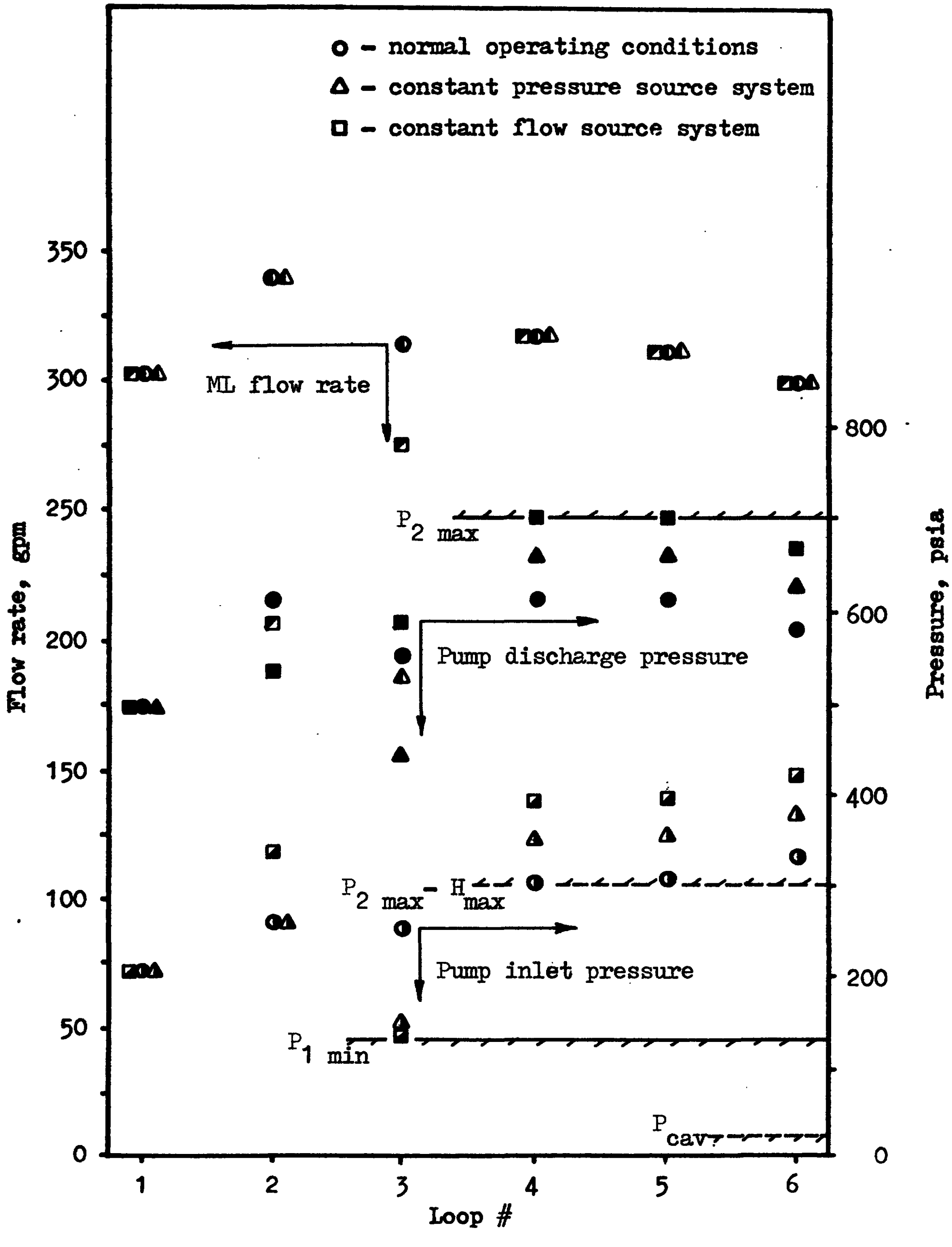


Fig. 33. Correction of the HE3 blockage (500% of normal flow resistance) of Fig. 30 by partially bypassing pump #2.

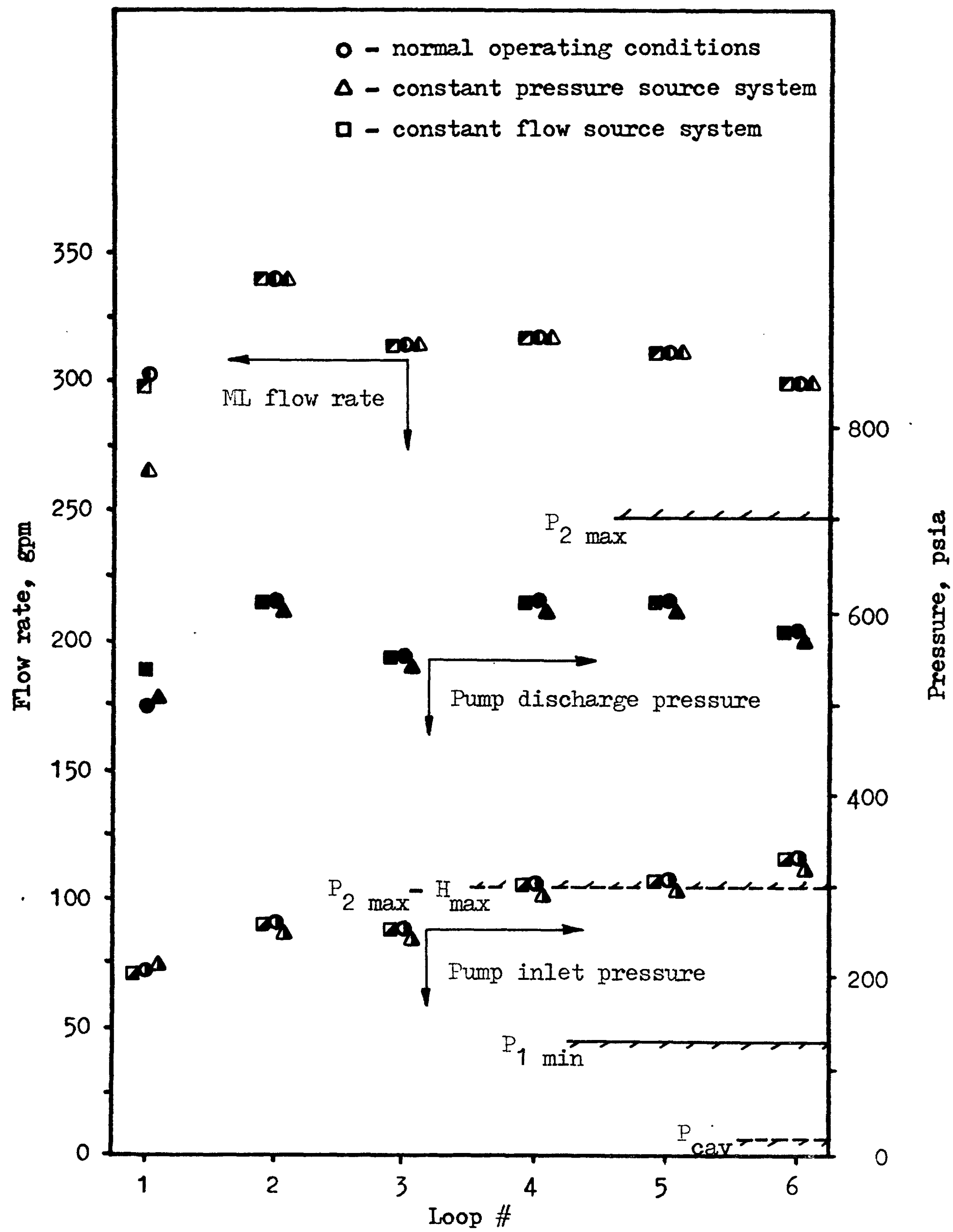


Fig. 34. Effect of the DL1 flow resistance increase of 30% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

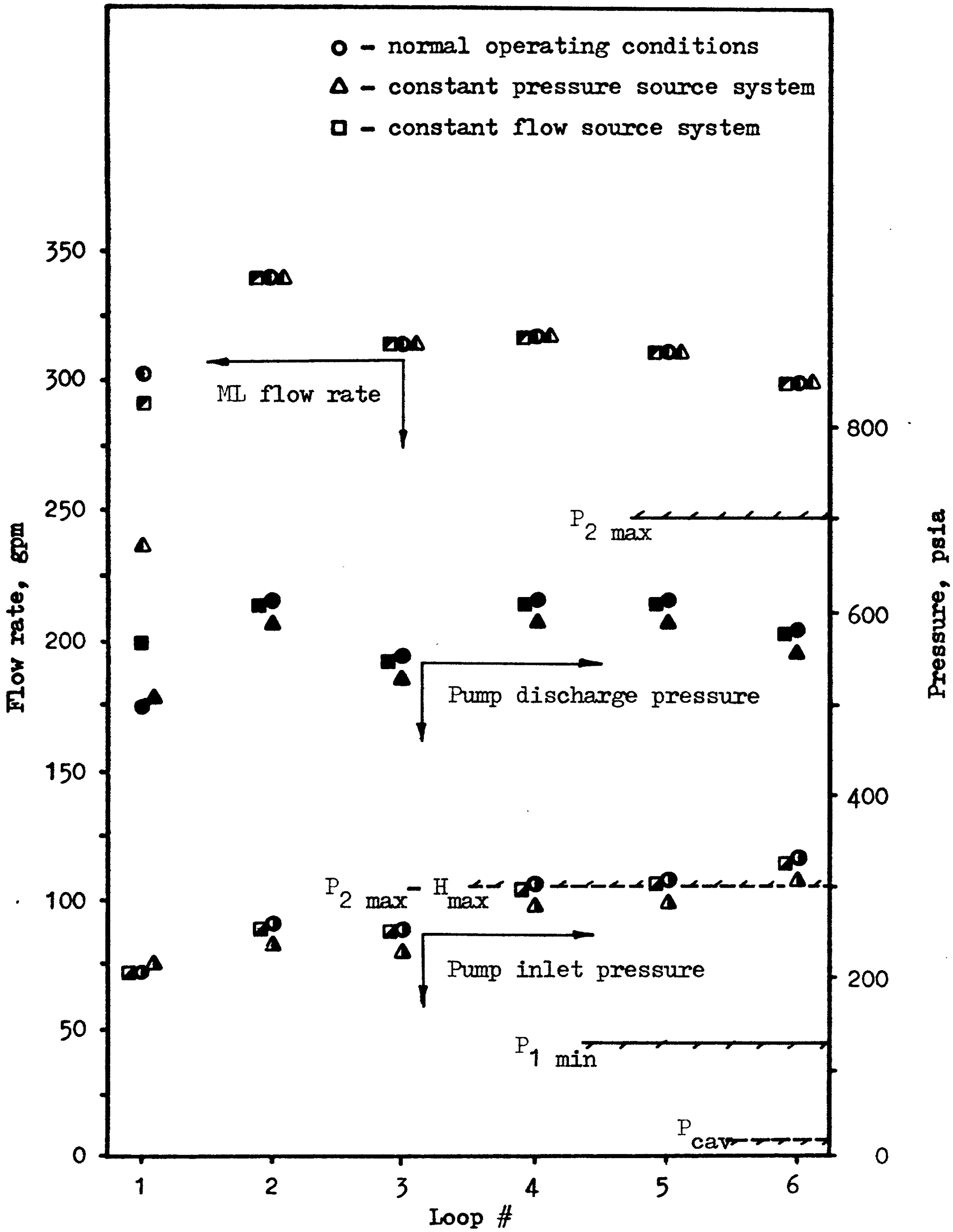


Fig.35. Effect of the DL1 flow resistance increase of 60 % on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

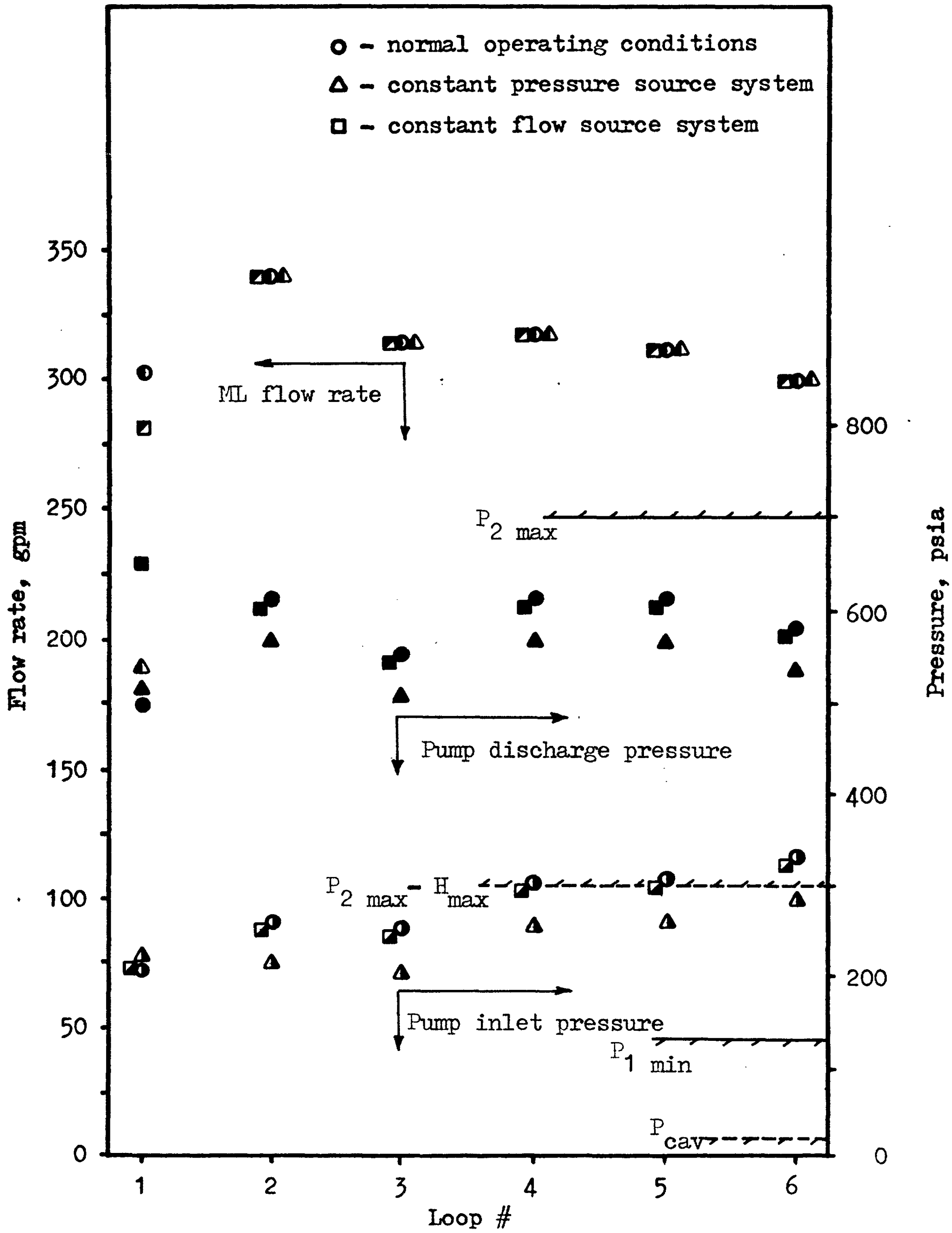


Fig. 36. Effect of the DL1 flow resistance increase of 150% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

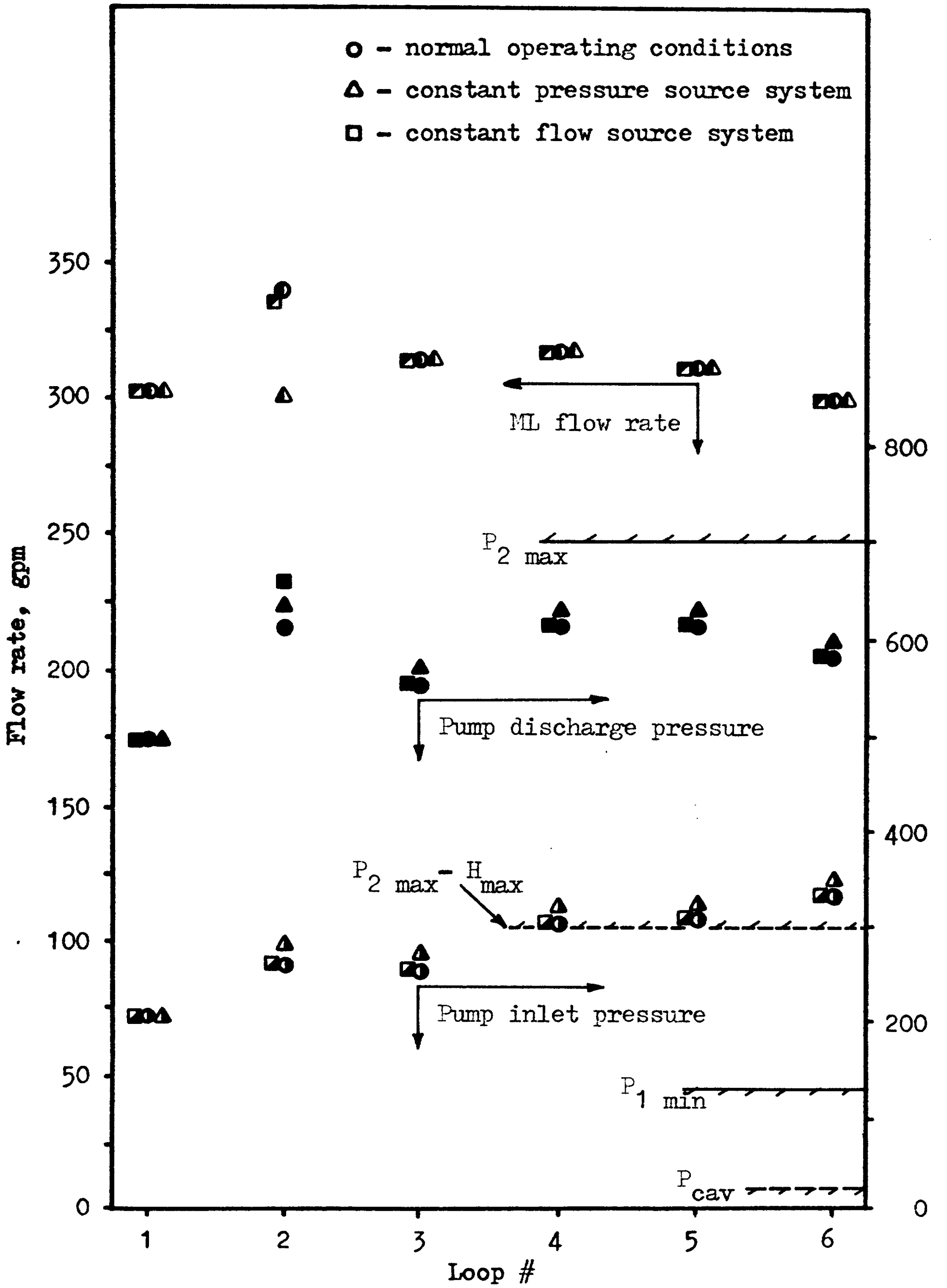


Fig.37. Effect of the DL2 flow resistance increase of 30 % on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

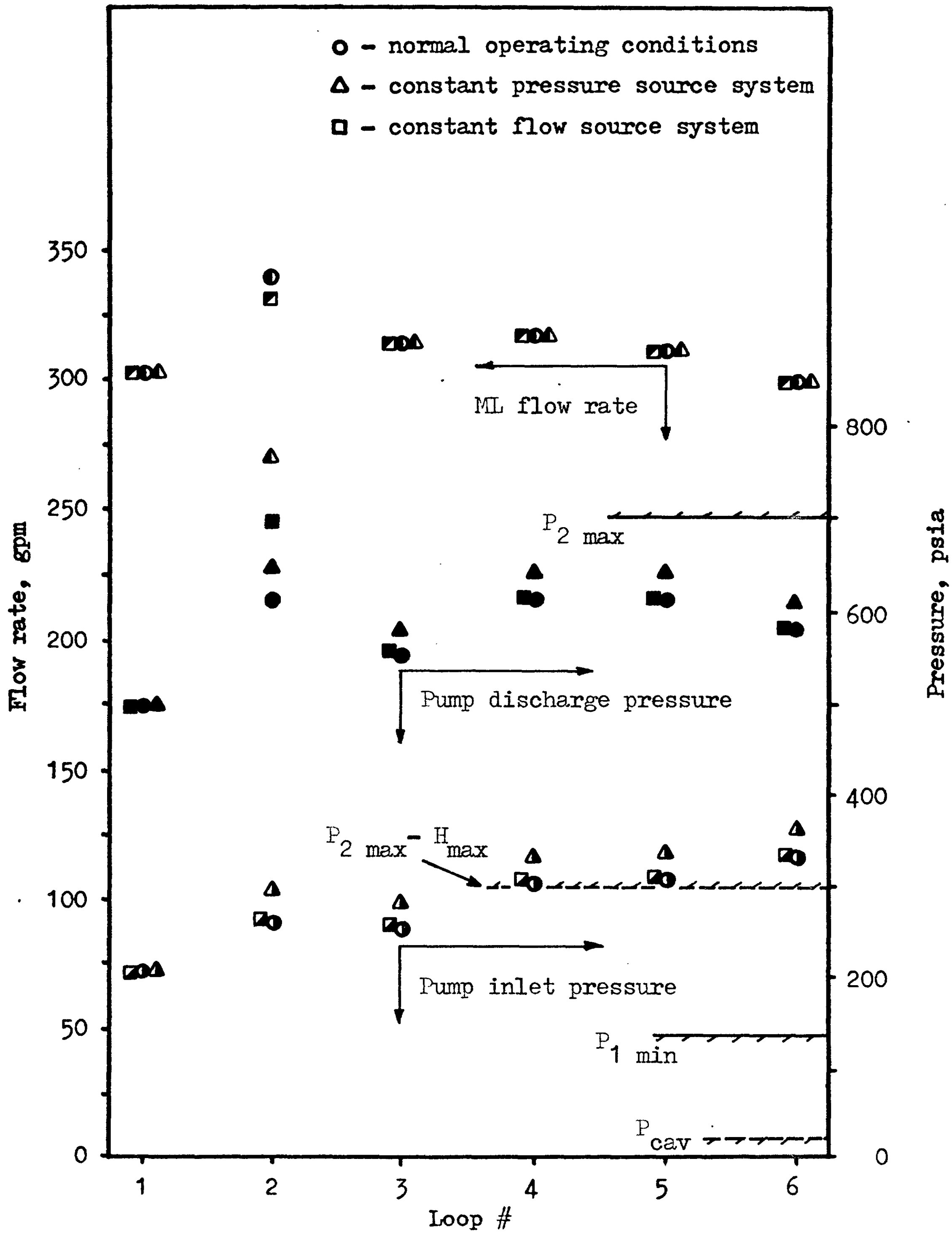


Fig.38. Effect of the DL2 flow resistance increase of 60 % on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

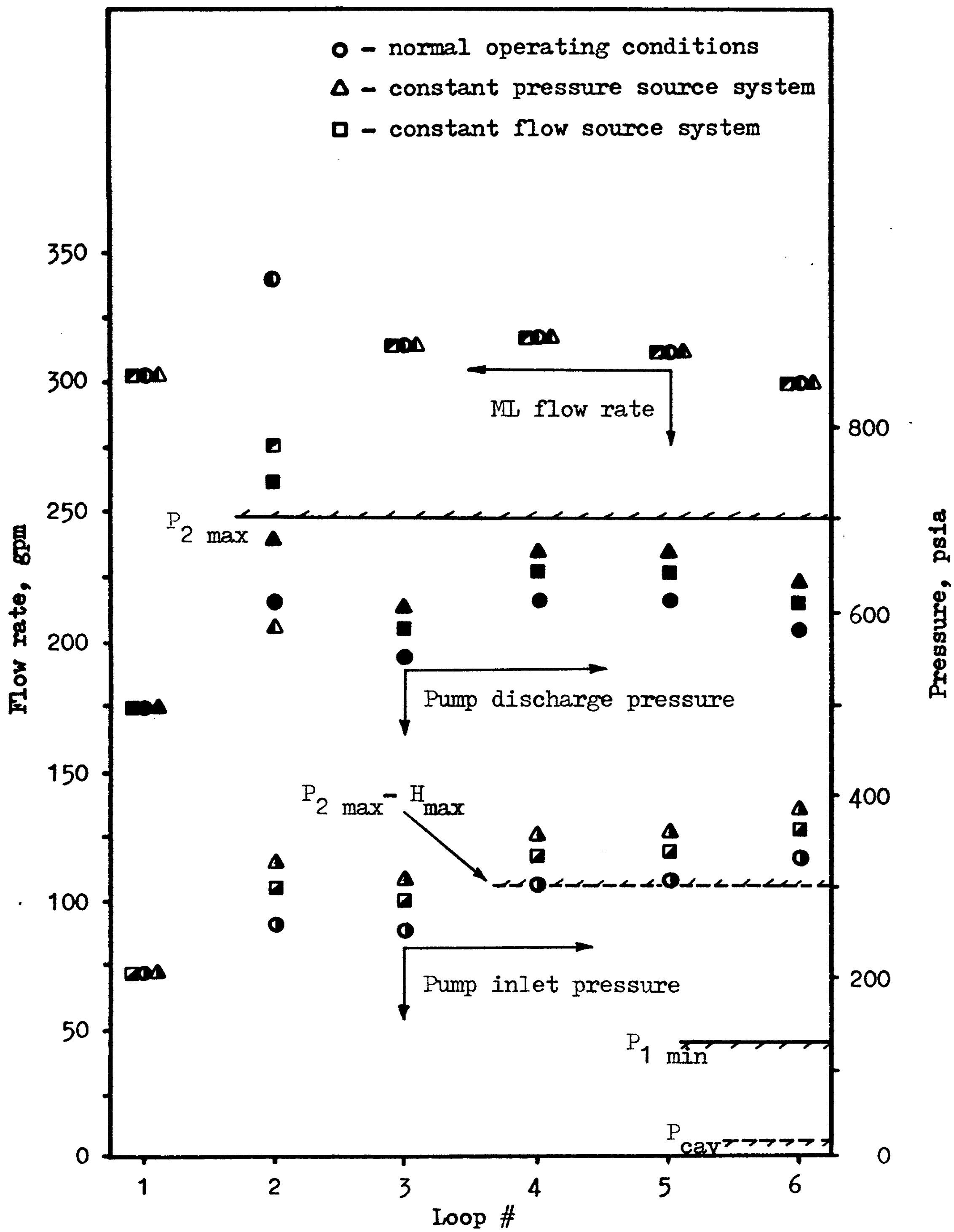


Fig.39. Effect of the DL2 flow resistance increase of 150% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

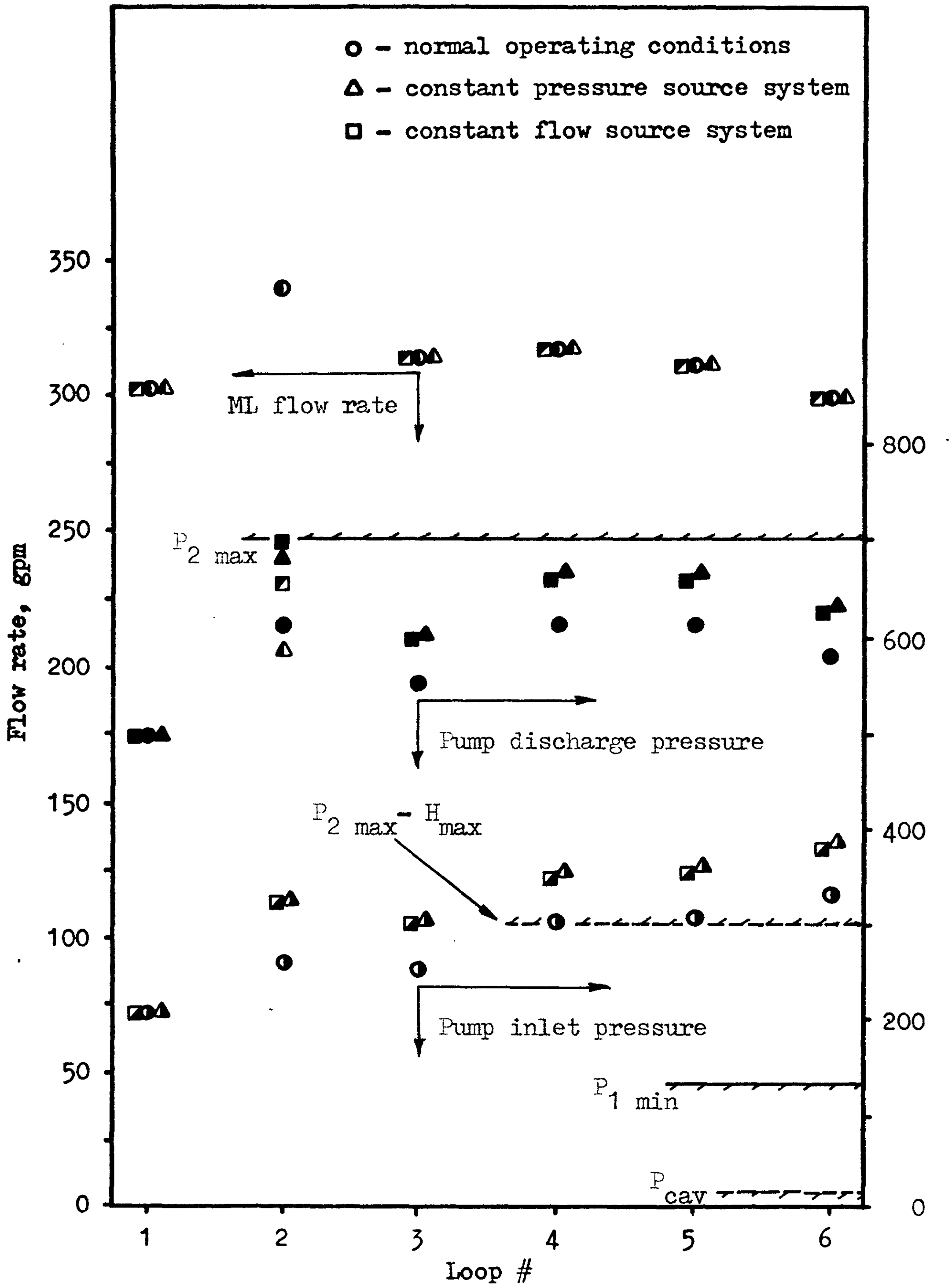


Fig. 40. Correction of the DL2 blockage (150% of normal DL2 flow resistance) of Fig. 39 by partially bypassing pump #2).

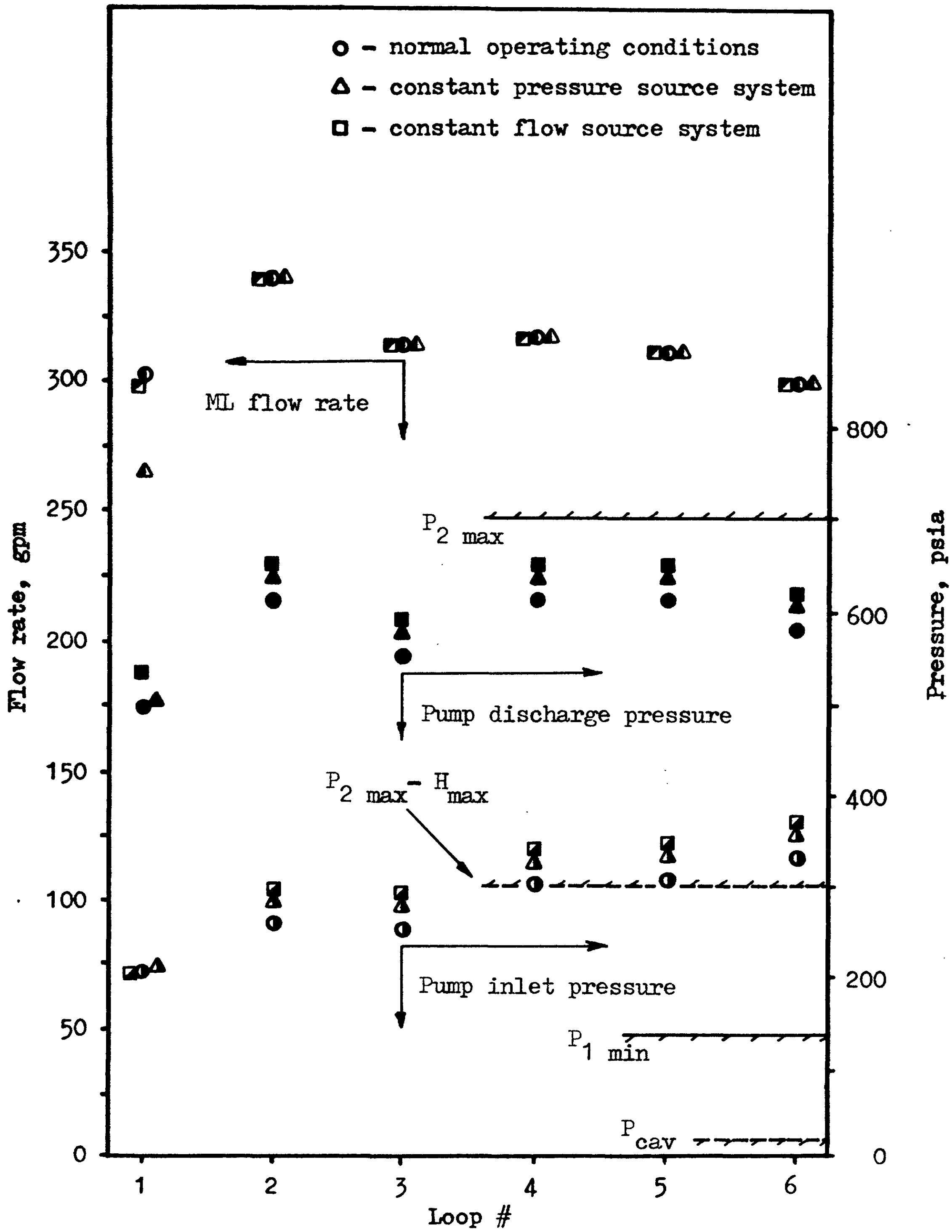


Fig.41. Effect of the ML1 flow resistance increase of 40% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

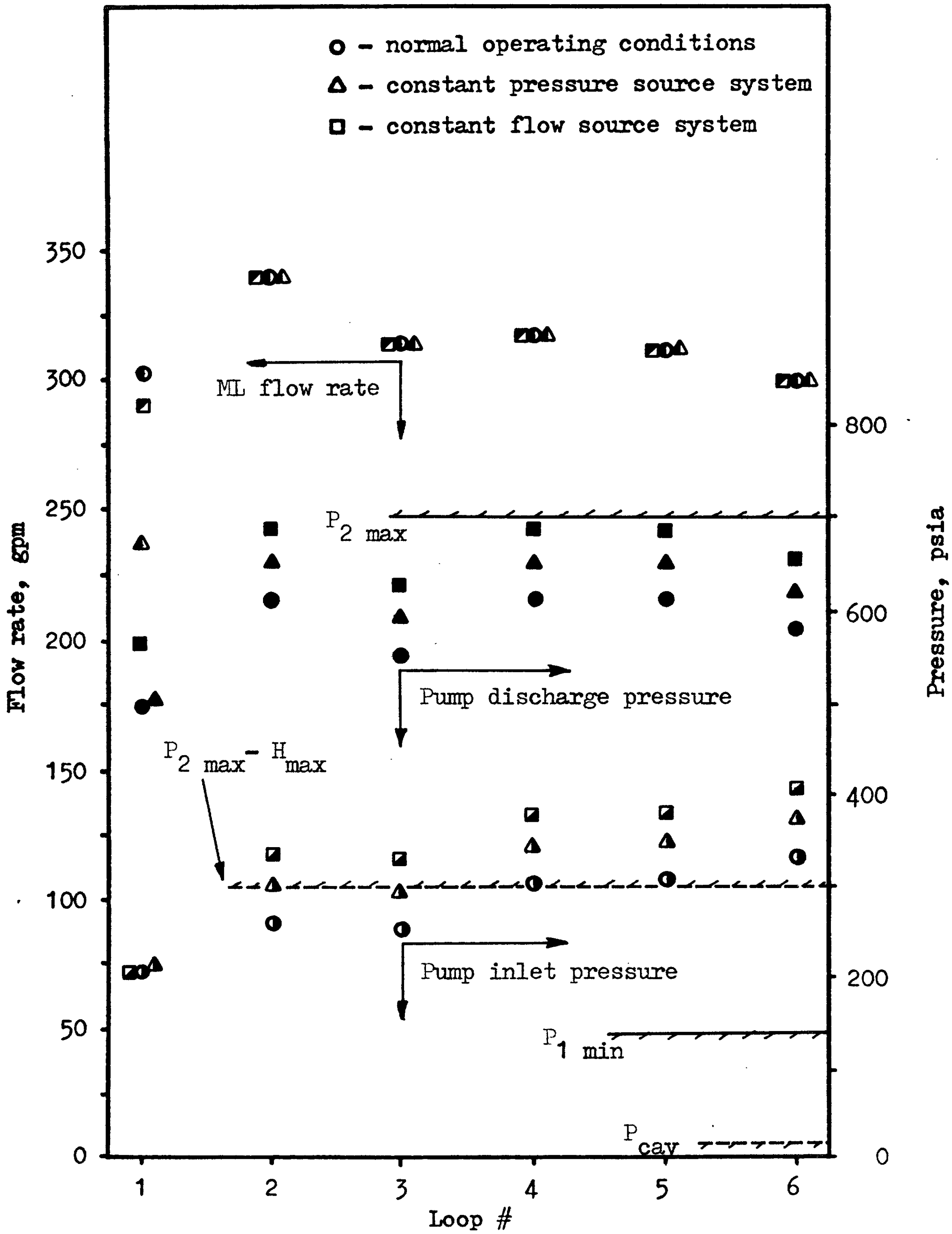


Fig. 42. Effect of the ML1 flow resistance increase of 80 % on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

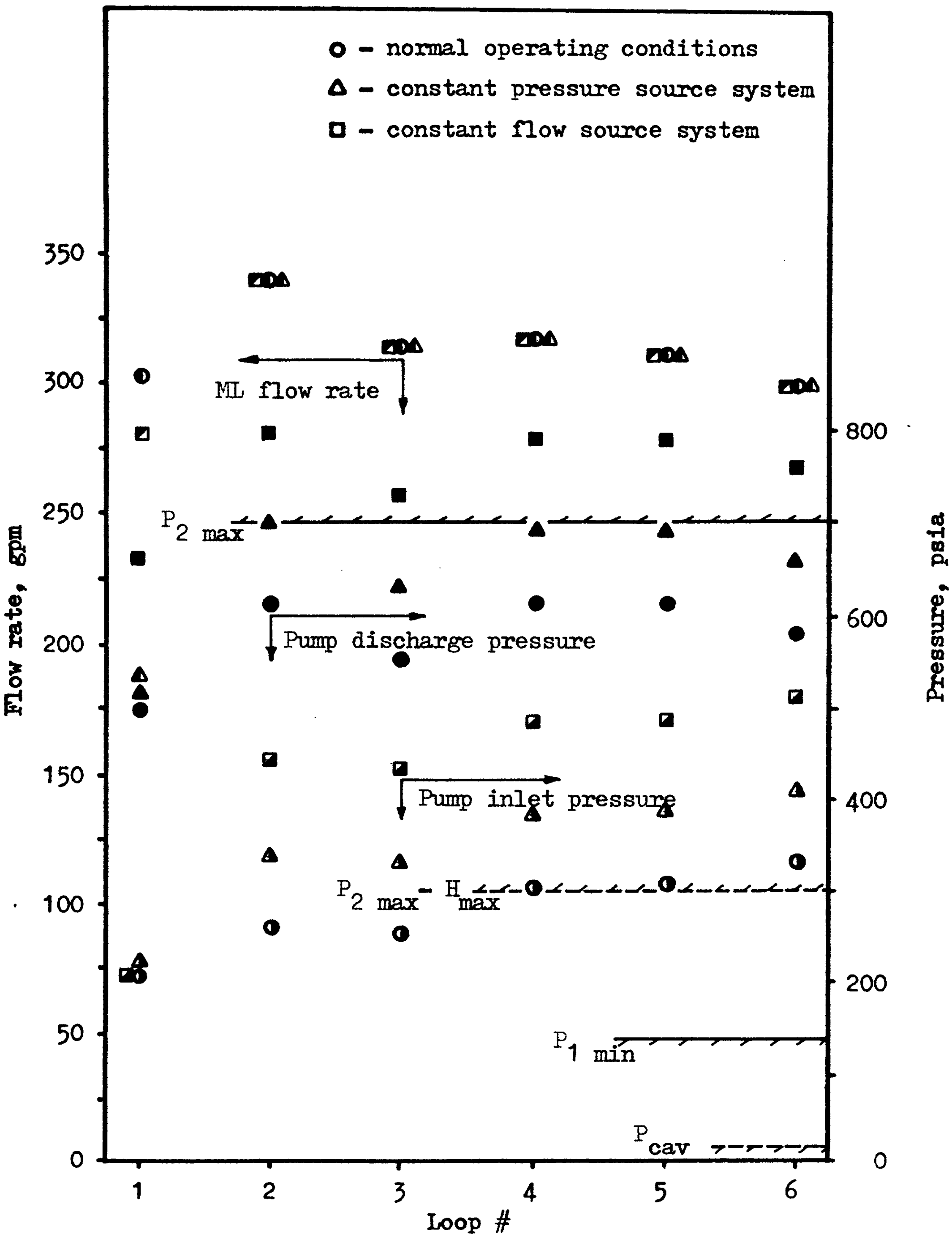


Fig.43. Effect of the ML1 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

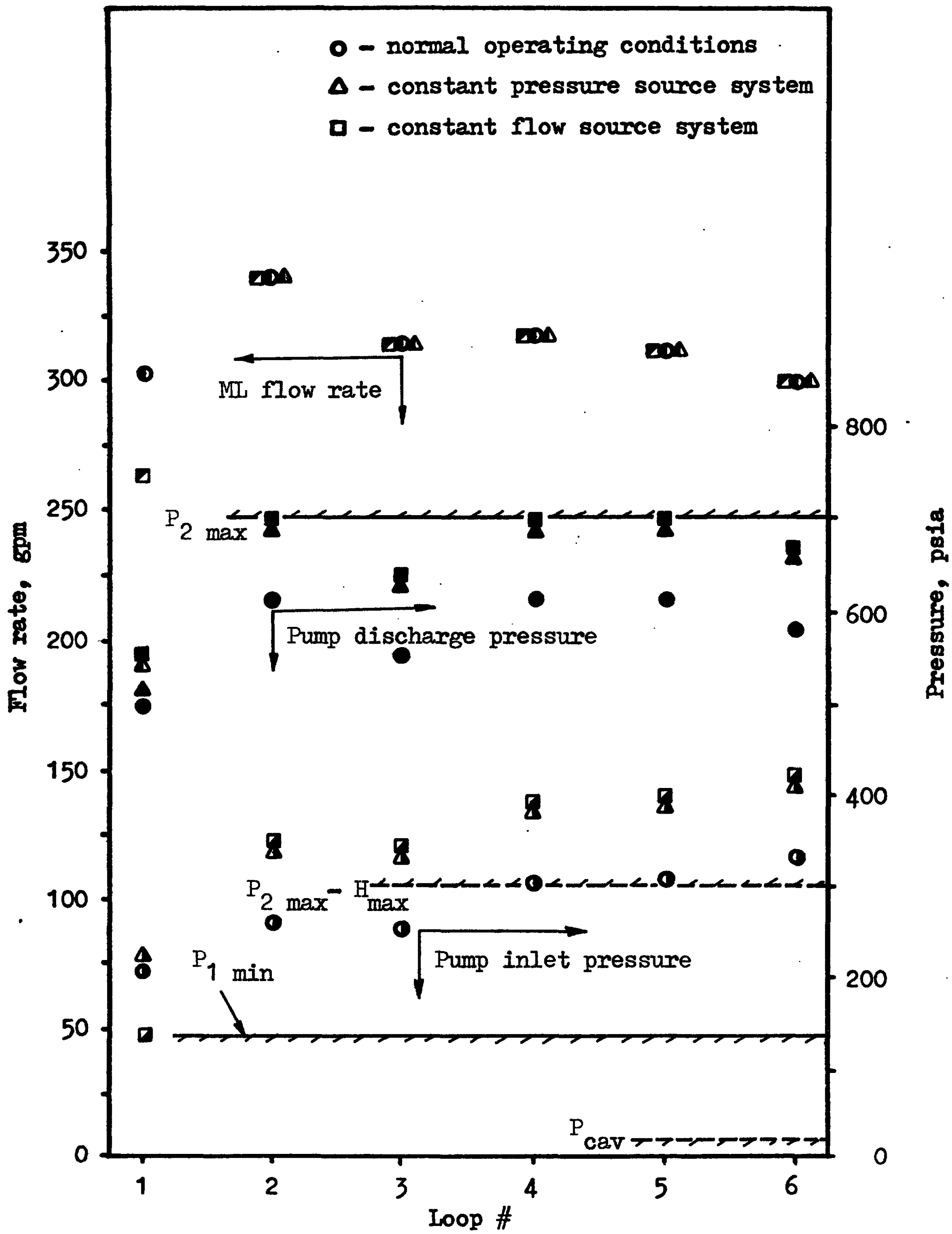


Fig. 44. Correction of the ML1 blockage (200% of normal ML1 flow resistance) of Fig. 43 by reducing the HT pressure (HT attached to loop #1).

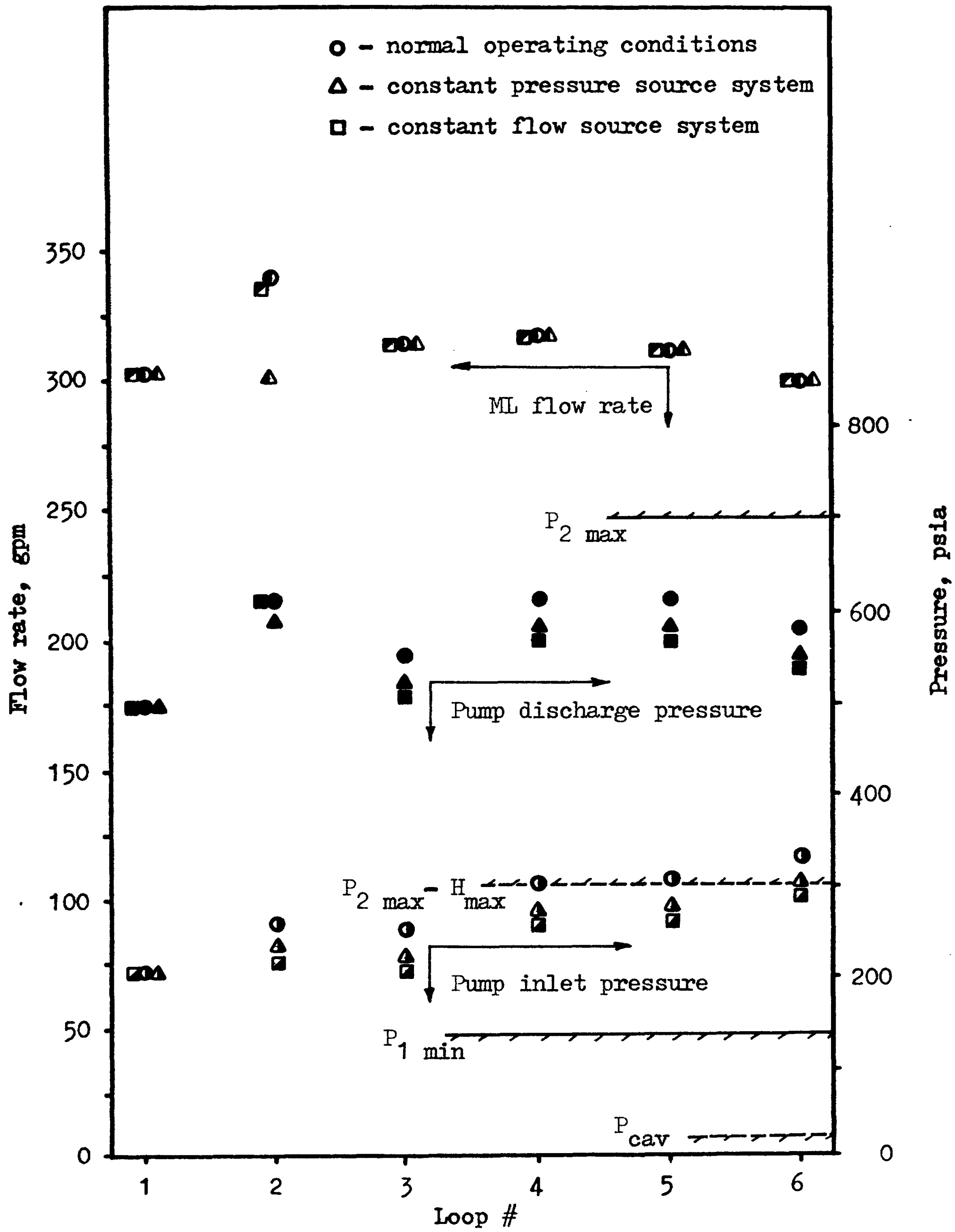


Fig.45. Effect of the ML2 flow resistance increase of 40% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

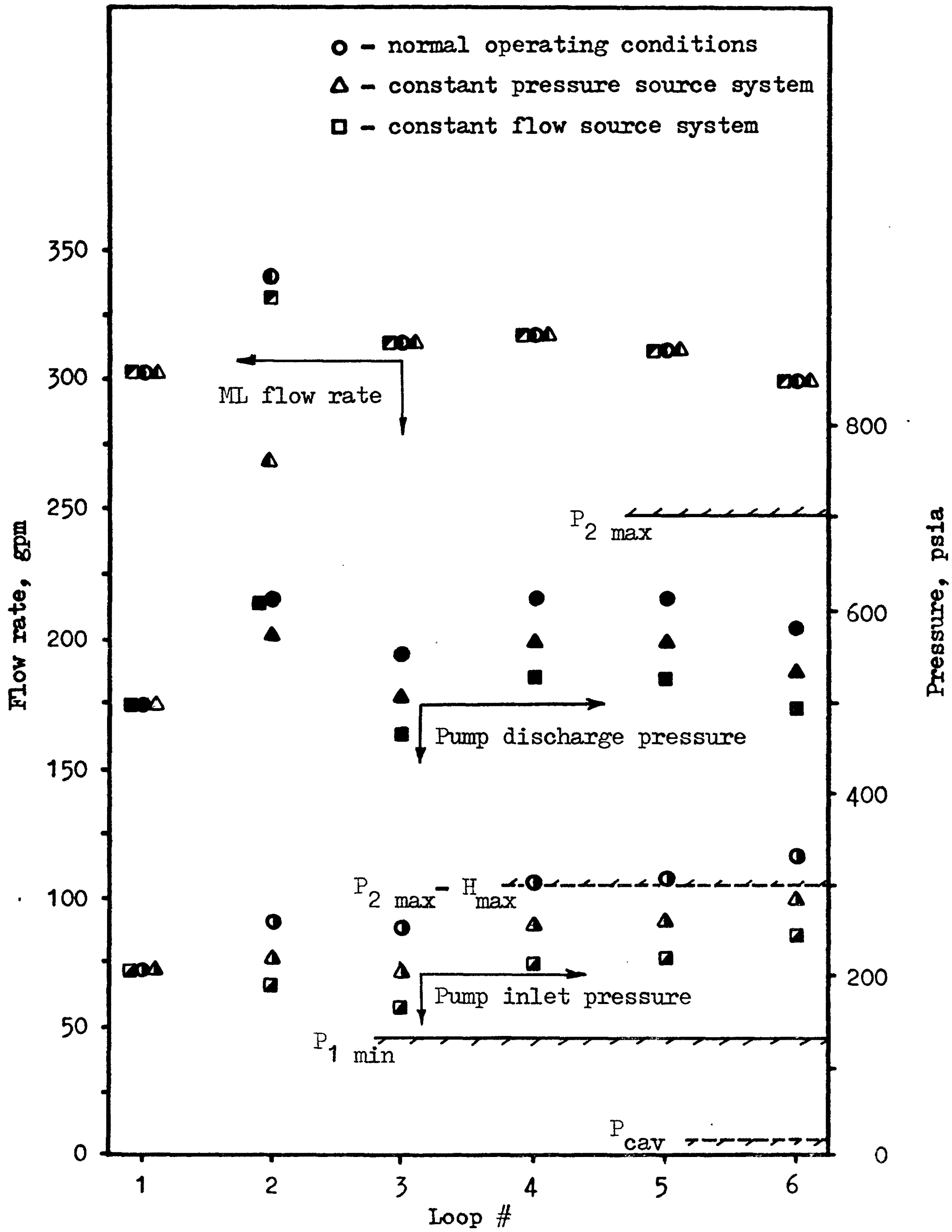


Fig. 46. Effect of the ML2 flow resistance increase of 80% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

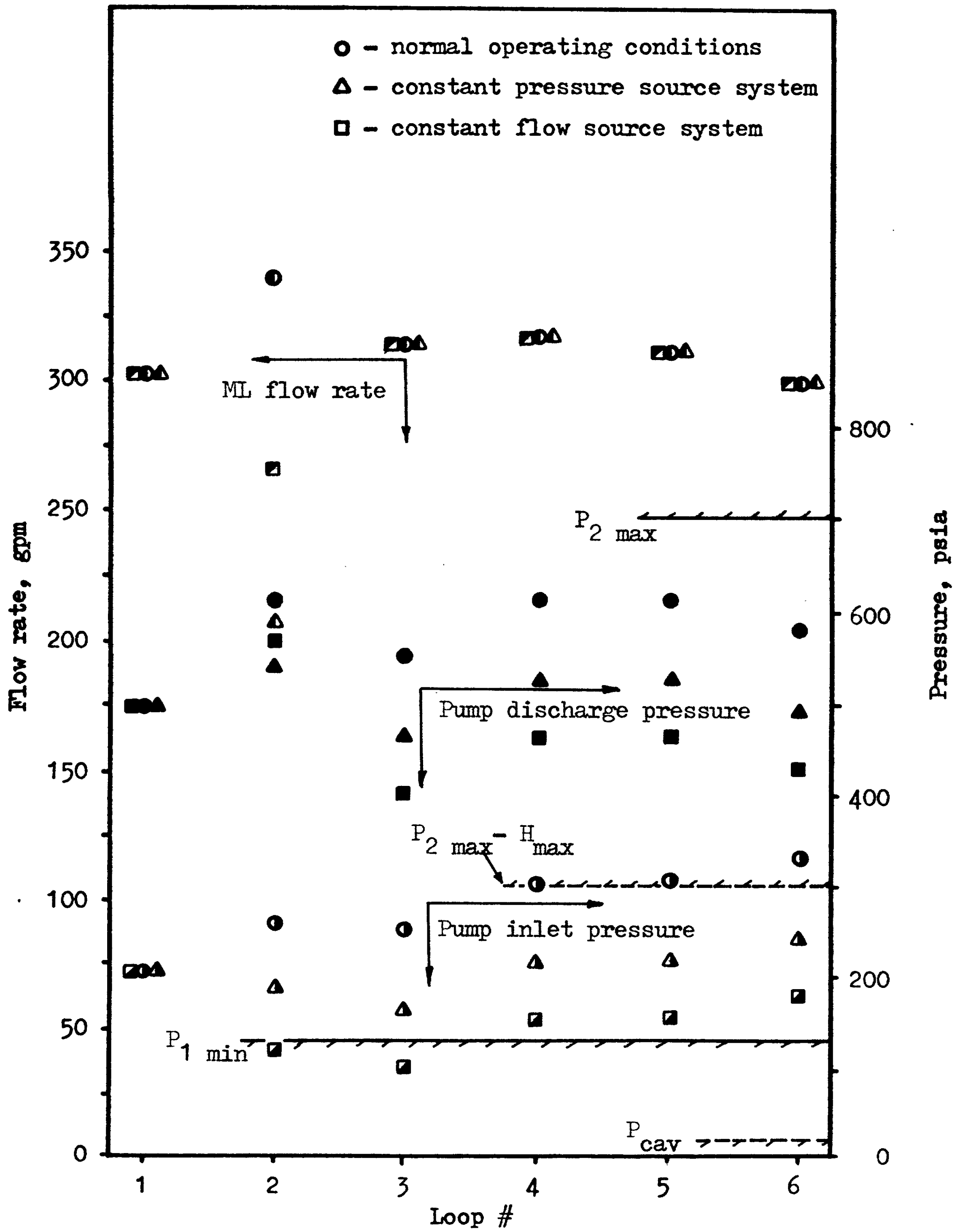


Fig. 47. Effect of the ML2 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

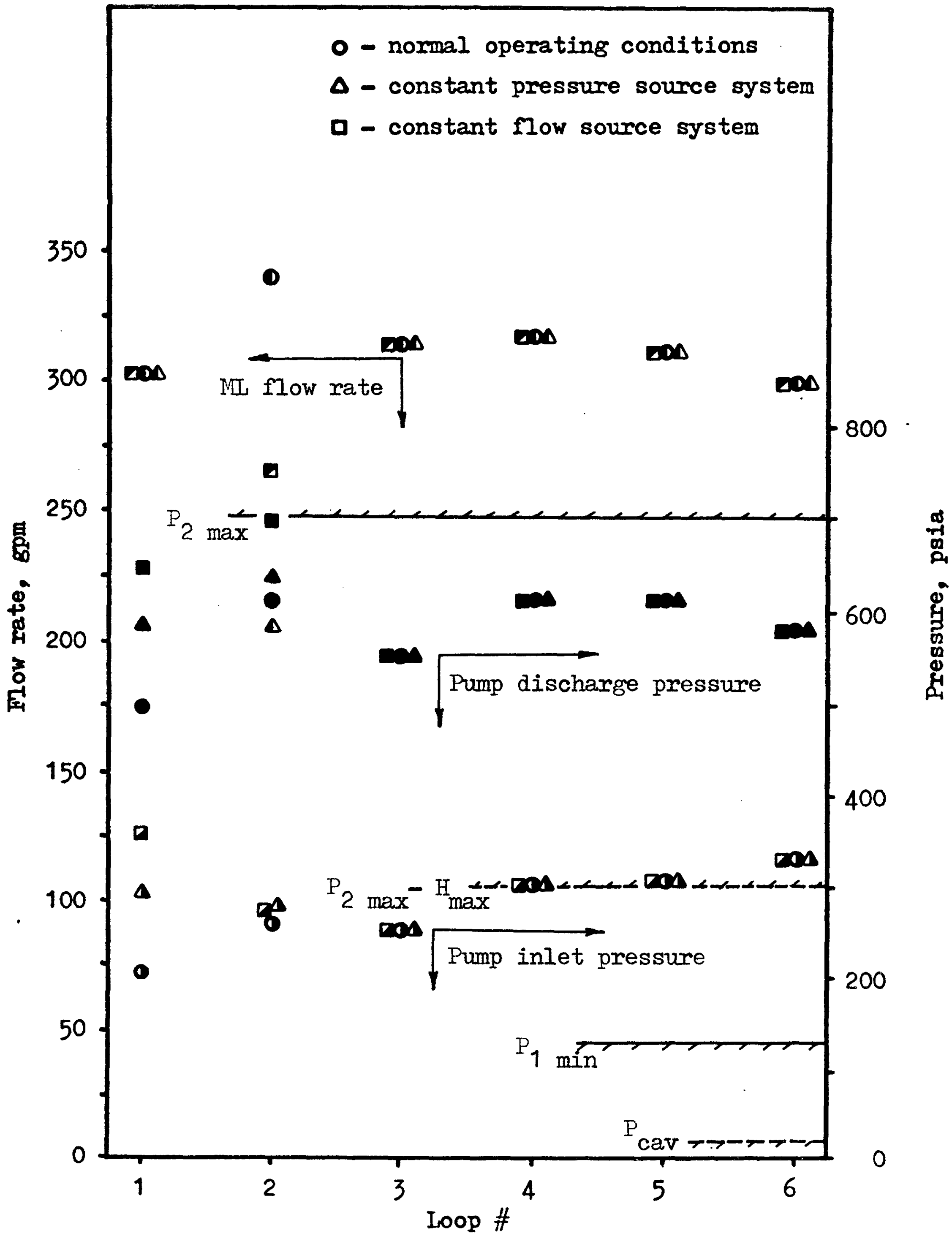


Fig. 48. Correction of the ML2 blockage (200% of ML2 normal flow resistance) of Fig. 47 by switching the pressure control from the left HT to the right one (HT attached to loop #6).

6. GENERALIZATION OF THE IMBALANCE EFFECTS ON CONFIGURATION 1 AND 2

Imbalance effects on the configurations which consist of separate loops, connected by only a single point, can easily be generalized for the use on systems with different number of circuits. Recall that imbalance is defined as a pipe flow blockage or resistance increase (DL, ML, HE), partial pump flow bypass (BPS), or a pump shutdown (DWN) which alters the pressure profile or the ML flow rate in the system.

Without a loss of generality, it can be assumed that the HT in SPC systems is attached to the circuit #1 and keeping this mind, imbalances can be classified by two criteria — their effects inside the circuit and their effects outside the circuit of the imbalance origin:

Type 1 (2) imbalance lowers (raises) the pump inlet and discharge pressures in the circuit containing the imbalance.

Type A (B) imbalance lowers (raises) the pressure level to the right of the circuit with the imbalance in SPC systems; or lowers (raises) the pressure level to the right and increases (decreases) it to the left accompanied by negative (positive) transverse flow in the dual pressure control systems.

Positive transverse flow is defined as the flow from the left HT (loop #1) to the right one (loop #N)

For the Configuration 1 with N number of loops (N even), individual imbalances are classified as follows:

TYPE 1 IMBALANCES	TYPE 2 IMBALANCES
Flow resistance increase	
HE 1,2,3,...,N-1,N	DL 1,2,3,...,N-1,N
ML 2,4,6,...,N-2,N	ML 1,3,5,...,N-3,N-1

TYPE A IMBALANCES	TYPE B IMBALANCES
Flow resistance increase	
HE 1,3,5,...,N-3,N-1	HE 2,4,6,...,N-2,N
DL 1,3,5,...,N-3,N-1	DL 2,4,6,...,N-2,N
ML 2,4,6,...,N-2,N	ML 1,3,5,...,N-3,N-1
Pump flow bypass	
BPS 1,3,5,...,N-3,N-1	BPS 2,4,6,...,N-2,N
Pump shutdown	
DWN 1,3,5,...,N-3,N-1	DWN 2,4,6,...,N-2,N

The pump flow bypass cannot be classified into Type 1 or 2, since it simultaneously lowers the discharge and increases the inlet pressure. The flow bypass thus can be used to correct both, a Type 1 or a Type 2 imbalance within a loop of the imbalance origin at the expense of reduced flow in the same loop. A pump shutdown is neither a Type 1 nor a Type 2 imbalance since it can not be corrected within the same loop. If desired, similar classification can be performed on the imbalances in Configuration 2.

Attaching the HT to the N^{th} circuit in SPC does not change the definition of Type 1 (2) imbalance; it does, however, change the Type A (B) definition in the sense, that now a Type A (B) imbalance would raise (lower) the pressure level to the right of the circuit with the imbalance in SPC systems.

7. TRANSIENT EFFECTS DURING A PUMP START-UP

The oil in a loop cannot be accelerated instantaneously. For this reason, when a pump is turned on, the oil is compressed until the pressure across the pump reaches a value set by the pressure relief valve. Ordinarily, the time needed for the relief valve to open is negligibly small (here probably less than a second). During this time the pump can cavitate, since the inlet flow, which is virtually zero, does not match the discharge flow, but because the time period is indeed so small, no significant damage to the pump can result[13]. As soon as the relief valve opens, all of the oil starts to flow through it, the discharge flow matches the inlet flow, and the momentary cavitation is relieved. During a pump start-up, a pump relief valve thus helps to relieve the low inlet pressure and protects the pump from cavitation damage. The oil is accelerated only slowly through the HE and therefore the pressure drop due to the accelerated flor through this passage is negligible. As the cooling oil is accelerated through the pipes the relief valve gradually closes. By the time the relief valve reaches its NOC position, the circuit flow rate is not far from the steady state. Thus a pump - relief valve system behaves as a constant pressure source during a pump start-up even in configurations with constant flow sources. Only the applied pump head differs.

The system behavior just described should be proven experimentally and for this purpose the following analysis was performed.

"A fluid transmission system may be characterized by means of lumped models whenever the significant wave lengths of all variables are large compared with the physical dimensions of the system. Otherwise, the actual distributed nature of the system may produce appreciable effects not present in the lumped model" [1]. For shorter lines the fluid capacitance, inertance, and resistance can be assumed to be concentrated at single locations and their interaction can be assumed to be negligible.

To determine whether lumped or distributed model should be used for a pump start-up simulation, the time constant τ_a associated with the acceleration of the oil and the time constant τ_b associated eith the sonic velocity in the oil are compared. The distributed nature of the system can be neglected if $\tau_b \ll \tau_a$.

From the force balance on an incompressible, inviscid, uniform pipe flow:

$$\rho \frac{\partial u}{\partial t} = - \frac{\partial p}{\partial x} \tag{7-1}$$

Each loop is composed of three segments — HE, DL, ML, but only DL, ML need be considered in the unsteady flow since the dimensions of the HE are negligibly small compared with the other two segments. Since only an estimate is needed here, assume that the ratio of pressure drop across each segment and the pump head remain constant for all time:

$$\frac{\Delta P_{DL}}{H} = \text{const.} = r_1$$

$$\frac{\Delta P_{ML}}{H} = \text{const.} = r_2$$

Thus from (7-1):

$$\rho_{DL}\frac{\partial u}{\partial t} = \frac{r_1 H}{L_{DL}}$$

$$\rho_{ML}\frac{\partial u}{\partial t} = \frac{r_2 H}{L_{ML}} \tag{7-2}$$

Integrating (7-2) and noticing that $u_{t=0} = 0$:

$$u = \frac{rH}{\rho L}\,t \tag{7-3}$$

From (4-7) at steady state the flow velocity is:

$$v = \left(\frac{2rHd}{\rho L f}\right)^{\frac{1}{2}} \tag{7-4}$$

The steady state is reached when $u = v$ and the time τ_a to reach steady state is, from (7-3) and (7-4):

$$\tau_a = \frac{\rho L}{rH} \left(\frac{2rHd}{\rho L f} \right)^{\frac{1}{2}}$$

Or

$$\tau_a = \left(\frac{2\rho dL}{rHf} \right)^{\frac{1}{2}} \qquad (7\text{-}5)$$

Using the average oil properties and system dimensions listed in Tab. 1 following results are obtained:

Constant flow source system, $H = H_{max} = 450$ psi : $\tau_{a\ DL} = 8.58$ sec

$\tau_{a\ ML} = 8.15$ sec

Constant pressure source system, $H = 311$ psi : $\tau_{a\ DL} = 10.50$ sec

$\tau_{a\ ML} = 9.98$ sec

The time constant τ_b associated with the speed of sound c in oil is equal to:

$$\tau_b = \frac{L}{c} \qquad (7\text{-}6)$$

where $$c = \left(\frac{\beta}{\rho} \right)^{\frac{1}{2}}$$

Again, using the average properties listed in Tab. 1 and $\beta_{oil} = 3.6 \times 10^7\ lb_f$ / sq. ft. , and considering the loop as a whole from (7-6) :

$$\tau_b = 5.9 \text{ sec}$$

Thus, since the order of magnitude of both time constants is equal, the pipeline network must be analyzed as a distributed parameter system. For a long transmission line the simplification of the continuity and momentum equations for an unsteady, inviscid, compressible flow in a uniform elastic pipe leads to a pair of simultaneous partial defferential equations, known as the wave equations:

$$- \frac{\partial q}{\partial x} = \frac{A}{\beta} \frac{\partial p}{\partial t}$$

$$- \frac{\partial p}{\partial x} = \frac{\rho}{A} \frac{\partial q}{\partial t} \qquad (7\text{-}7)$$

The frictional effects may be lumped at either end, or both ends of the line[1].

It is not possible therefore to simulate a pump start-up on an electric analogy model, such as the one used in this work, by adding inductors into the circuits. Rather, the equations (7-7) should be solved numerically on a digital computer, taking into account the fact that each loop of Configuration 1 consists of three segments with varying oil properties and dimensions.

Since the numerical analysis of the pump start-up was not performed it is recommended,on the basis of the results of Chapter 5, that each pump be started individually starting with the one farthest from the operational head tank. A similar procedure should be used for a systematic pump shutdown.

8. CONCLUSIONS AND RECOMMENDATIONS — OPTIMAL DESIGN

Since only Configuration 1 does not require flow blocking diaphragms, which are presently not available, for its normal operation, it remains the best solution.

The installment of head tanks is really possible only at the feeder ends; a head tank at one end should be built. For greater safety, a second head tank may be installed at the other end of the cable route. To build a return line is impractical, however, and therefore, to prevent transverse flow and its adverse effects on the system ML flow rates, only one of the head tanks should be operated (SPC) at a time, while the other head tank would be used as a stand-by.

Since it was found that the pump - relief valve arrangement operating as a constant flow source is, for practical imbalance sizes, superior to the pump - relief valve arrangement operating as a constant pressure source, each pump should have a relief valve initially adjusted for the required main line flow rate, and its further adjustment would be delayed until the pressure across it reaches the maximum pressure rating of the pump (here taken as 450 psi). Thus the pump head is allowed to increase and the main line flow rate stays approximately constant (FSS).

For existing installations, such as the Dunwoodie - Rainey system, the pump bypass is the simplest and effective additional line pressure profile control. For its application each pump must be provided with an extra bypass pipe and a valve. The valve position should be simultaneously controlled by the pump discharge and inlet pressures and whenever the discharge pressure exceeds a maximum specified value (here taken as 700 psia), or the inlet pressure drops below a set minimum (here 130 psia), the valve further opens and adjusts the out-of-line pressure. For the system of Fig. 2 and Tab. 1 (Dunwoodie - Rainey) it was found that the pump bypass line pressure profile control was sufficient for all practical imbalance sizes, but for greater security, a second independent control, measuring the discharge pressure and set for somewhat higher value than the pump bypass, could be used to shut down the pump in case of a malfunction of the pump bypass. A pump should also be shut down when the ML flow rate drops below a minimum value (here assumed 200 gpm) and when, at the same time, the pump discharge pressure is at $P_{2\ max} = 700$ psia or higher.

For new designs, the best and most effective line pressure profile control was found to be the head tank pressure adjustment. For this purpose all pump inlet and discharge pressures must be monitored and whenever a pressure exceeds the maximum specified discharge pressure or falls bellow the minimum specified inlet pressure, the head tank pressure is automatically adjusted to a new value. As in the case of the pump bypass, for the system of Fig. 2 and Tab. 1 it was found, that the head tank pressure adjustment was sufficient for all practical imbalance sizes. Again, as a safety measure, pump shutdown could be used in case of malfunctioning primary control, or when the flow rate drops below the minimum specified value. Using the partial pump bypass as a complementary line pressure profile control to the HT pressure adjustment would improve the controlability of the system pressure profile disturbed by large imbalances,similar to the one illustrated in Fig. 44,or by a number of imbalances.

The advantage of the head tank adjustment method over the pump bypass lies in the fact that no additional flow reduction occurs after an imbalance occurs.

If the choice of pump station locations is available, to permit a wider range for the application of the head tank pressure adjustment method or the partial pump bypass method of the line pressure profile control, the length of individual loops should be selected in such a way so as to offset the pressure increase due to elevation. An improvement to the various line pressure control methods would result from monitoring the ML pressures and using them as an additional information for determining the instant of control application. This would remove the need for a pump shutdown in cases when the ML flow rate drops below 200 gpm.

To prevent chattering, the pump start-up should be manually controlled, and in order to minimize imbalance accumulation and propagation, the pumps should be started individually, beginning with the one farthest from the head tank in use; the flow in each loop should be allowed to reach steady state before another pump is started.

In case of a major main line blockage, both head tanks should be operated simultaneously. In this case the system would really be separated into two parts, each being a single pressure control system.

For further work it is recommended that the effects of the pump start-up on the line pressure profile be investigated.

It is possible that still other configurations permitting more efficient and less extensive pressure control exist. For their analysis, it might be helpful to build a model having nonlinear relationship between flow rate and pressure drop, corresponsing to that of steady state fully developed turbulent pipe flow.

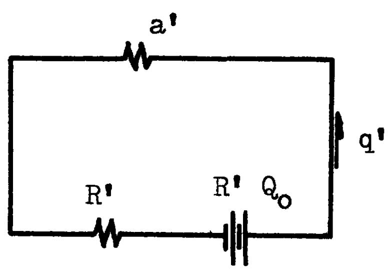
APPENDIX A

Error Analysis

Several types of errors are encountered when one wants to compare the model performance with the prototype's:

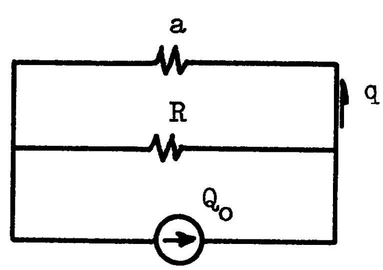
(1) Flow error due to the linearization

For simplicity consider only a single loop. Such error analysis will be completely valid only in cases where the transverse flow is maintained at zero (SPC), since then all loops are independent of each other, except for the absolute pressure level which cannot influence the pressure drop vs. flow rate relationship. It is probably safe to assume that these results can be used for approximating the errors in cases where the transverse flow is small.



Linear model

$$a' = a\,Q$$
$$A' = A\,Q$$
$$R' = R\,(Q_o - Q)$$



Non-linear prototype

Under normal operating conditions:

$$a = A \qquad q = Q$$
$$a' = A' \qquad q' = Q$$

where $A = H/Q^2$ = normal nonlinear loop resistance

A' = normal linearized loop flow resistance

$R = H/(Q_o - Q)^2$ = nonlinear pump leak resistance

R' = linearized pump leak resistance

a = variable nonlinear loop flow resistance

a' = variable linearized loop flow resistance

Q = normal loop flow rate

Q_o = ideal pump flow rate

q = nonlinear prototype flow rate

q' = linear model flow rate

Define

$$k \equiv \frac{a'}{R'} \qquad K \equiv \frac{A'}{R'} = \frac{Q_o - Q}{Q}$$

Since $$q' = \frac{R' Q_o}{a' + R'} \implies k = \frac{Q_o - q'}{q'}$$

$$a q^2 = R(Q_o - q)^2 \implies q = \frac{Q_o}{1 + (Kk)^{\frac{1}{2}}}$$

Or

$$q = \frac{Q_o}{1 + K^{\frac{1}{2}}\left(\frac{Q_o}{q'} - 1\right)^{\frac{1}{2}}} \tag{A1}$$

But here

$$K^{\frac{1}{2}} = 0.442$$

$$Q_o = 373 \text{ gpm}$$

Thus

$$q = \frac{373}{1 + 0.442\left(\frac{373}{q'} - 1\right)^{\frac{1}{2}}} \tag{A1'}$$

Or, in terms of the flow error due to the linearization, ϵ_f :

$$\epsilon_f = q' - q = q' - \frac{Q_o}{1 + K^{\frac{1}{2}}\left(\frac{Q_o}{q'} - 1\right)^{\frac{1}{2}}} \tag{A2}$$

The error ϵ_f is plotted as a function of loop flow rate reduction $Q - q'$ in Fig. 49.

(2) Errors due to inaccurate model performance:

(a) Resistor accuracy

The resistors used were accurate within 5% and so the contribution to the uncertainty of answers $\epsilon_r = 0.05$.

(b) Current lost to ground

Since all resistors grounding the various points

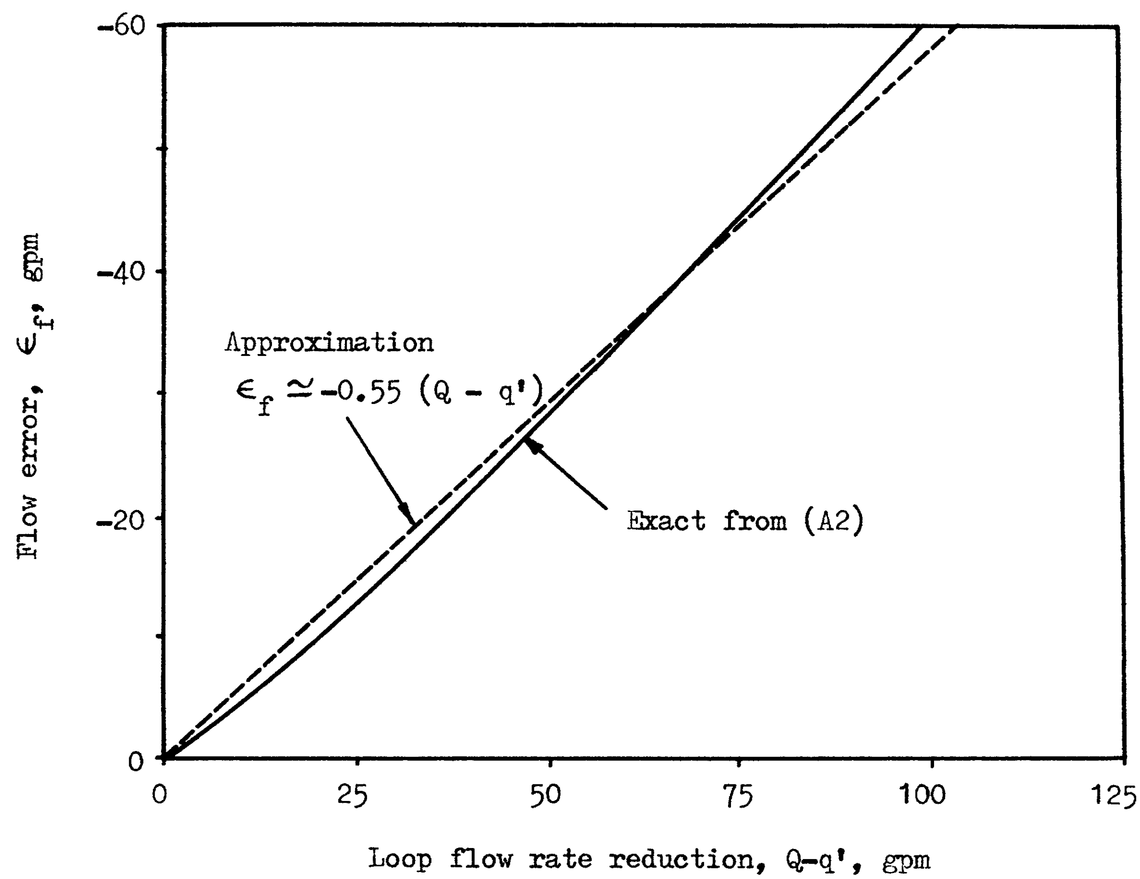


Fig. 49. Flow rate error due to the linearization as a function of the loop flow reduction.

were extremely large when compared with the circuit resistors, the amount of this error is negligible.

(3) Errors due to current lost through voltmeters:
There can be no significant error due to this effect since operational amplifiers were used wherever there was a danger of loosing some current through a measuring device.

(4) Errors due to accuracy of measuring devices:
All voltmeters and microammeters were accurate within 2% and thus the contribution to the uncertainty of answers $\epsilon_m = 0.02$.

(5) Errors due to presence of microammeters within the circuits:
The resistance of the microammeters was included in the overall resistance of each line segment, and therefore there is no error due to this effect.

(6) Pressure drop error due to linearization:
Considering again only a single loop, the validity of the following is limited to non-transverse flow systems. The nonlinear prototype loop pressure drop Δp, and the linear loop pressure drop $\Delta p'$ are

$$\Delta p' = a' q'$$
$$\Delta p = a q^2 = a' \frac{q^2}{Q}$$

where

$$a' = kR' = \frac{Q_o - q'}{q'} \frac{H}{Q_o - Q}$$

From Fig. 49: $q \simeq 0.45\, q' + 0.55\, Q$

Define the pressure drop error due to linearization, ϵ_p :

$$\epsilon_p = \Delta p' - \Delta p = (Q_o - q') \frac{H}{Q_o - Q} \left[1 - \frac{1}{q'Q} \left(0.45 q' + 0.55 Q \right)^2 \right] \tag{A3}$$

Here $Q_o = 373$ gpm

$Q = 312$ gpm

$H = 311$ psi

Thus

$$\epsilon_p \simeq (373 - q')\, 5.1 \left[1 - \frac{1}{312 q'} \left(0.45 q' + 171.6 \right)^2 \right] \qquad (A3)'$$

Evaluating (A3)' it is found that ϵ_p is negligible for $q' > 200$ gpm and is of the order of 5 psi at $q' = 200$ gpm. There is therefore no significant error in the pressure drop measurements due to the linearity of the model.

The measured flow rates will always be slightly lower than in the real case.

The overall uncertainty in the pressure drop and flow rate measurements ϵ_T is:

$$1 - \epsilon_T = (1 - \epsilon_r)(1 - \epsilon_m)$$

Or

$$\epsilon_T = 1 - (1 - \epsilon_r)(1 - \epsilon_m) \qquad (A4)$$

But since $\epsilon_r = 0.05$ and $\epsilon_m = 0.02$

$$\epsilon_T = 0.069$$

APPENDIX B

Effect of line blockages on Configuration 1

The Figures 49 - 59 represent the effects of very large unrealistic blockages on the constant pressure source system and the constant flow source system in Configuration 1, and compare these effects on the line pressures and the ML flow rates with NOC.

$P_{1\ min}$ represents the minimum inlet pressure, $P_{2\ max}$ is the maximum discharge pressure, P_{cav} is the pump cavitation pressure,and $P_{2\ max} - H_{max}$ represents the maximum inlet pressure when the constant flow source control scheme is employed together with the inlet pressure observation for the discharge pressure control. No imbalance correction was applied in either of these figures.

More information about the information presented here can be found in Sec. 5.3.

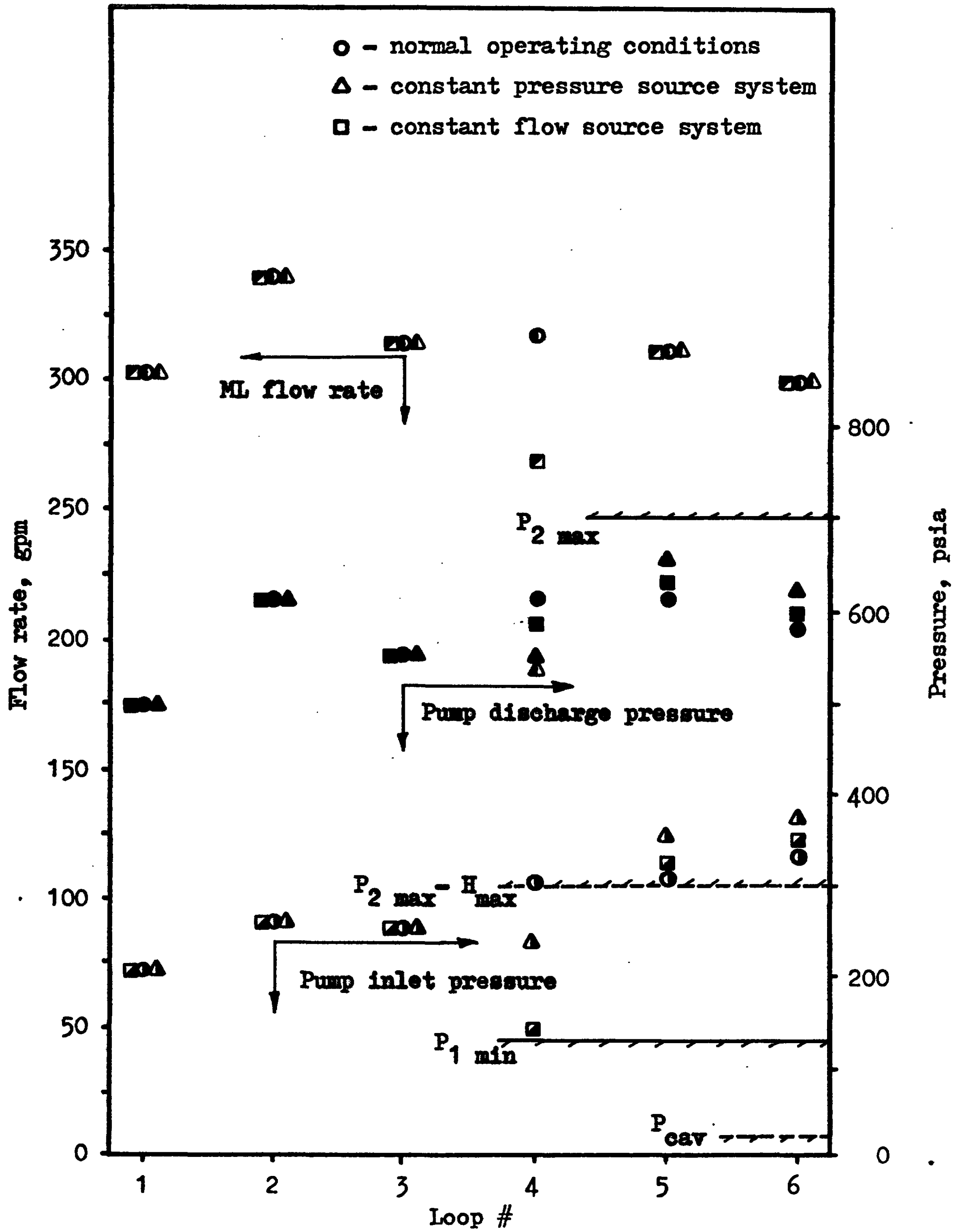


Fig. 50. Effect of the HE4 flow resistance increase of 500% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

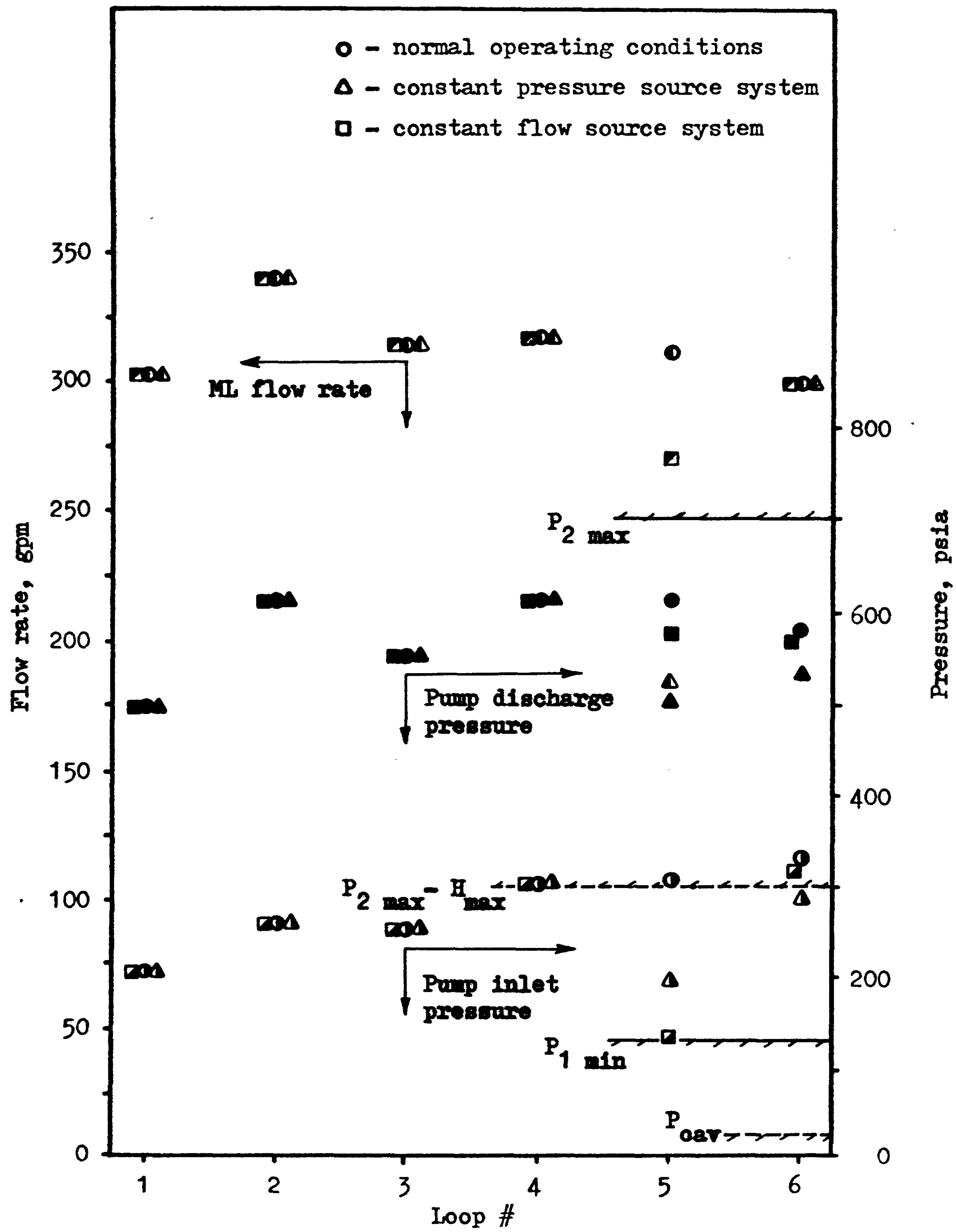


Fig.51. Effect of the HE5 flow resistance increase of 500% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

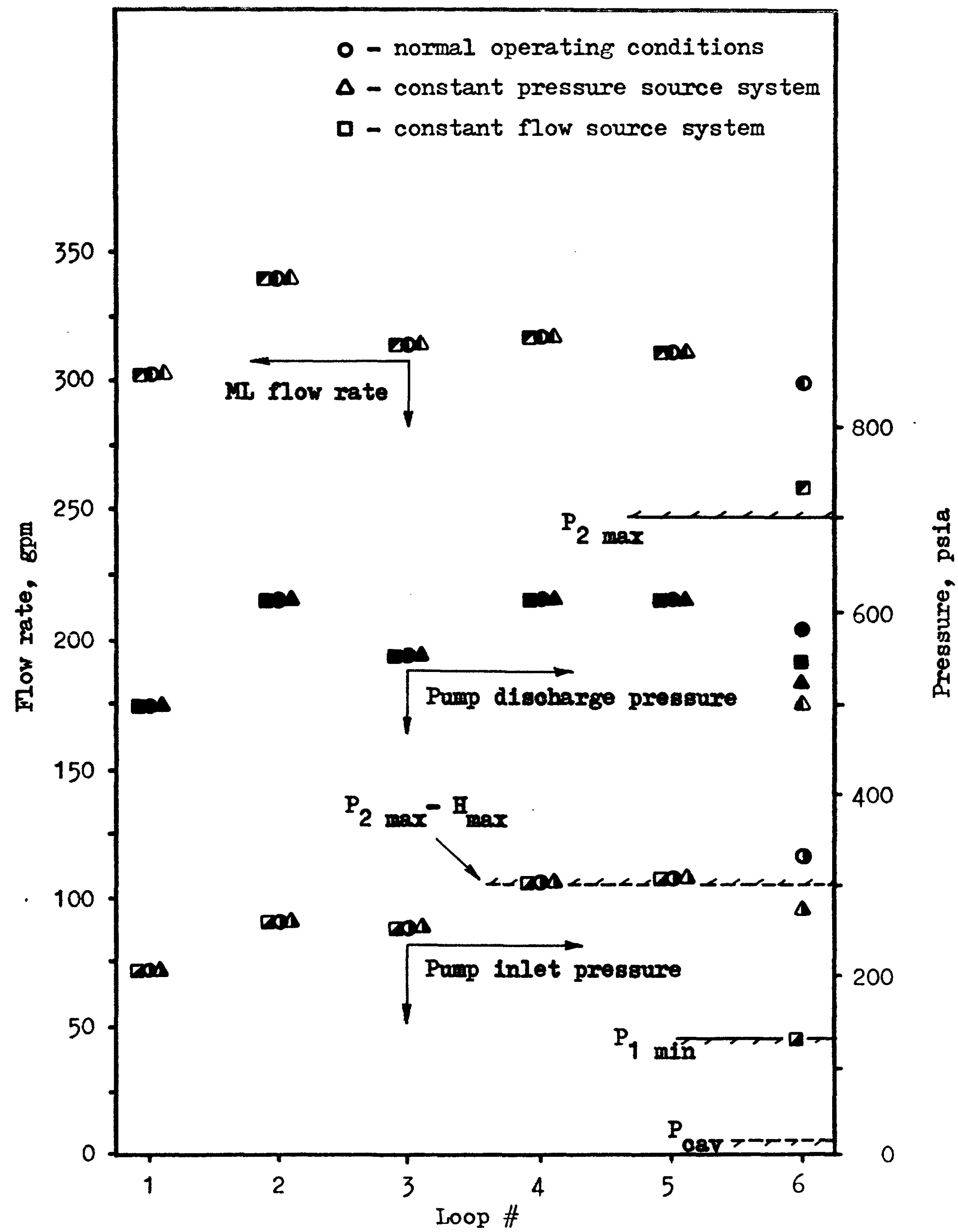


Fig.52. Effect of the HE6 flow resistance increase of 500% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

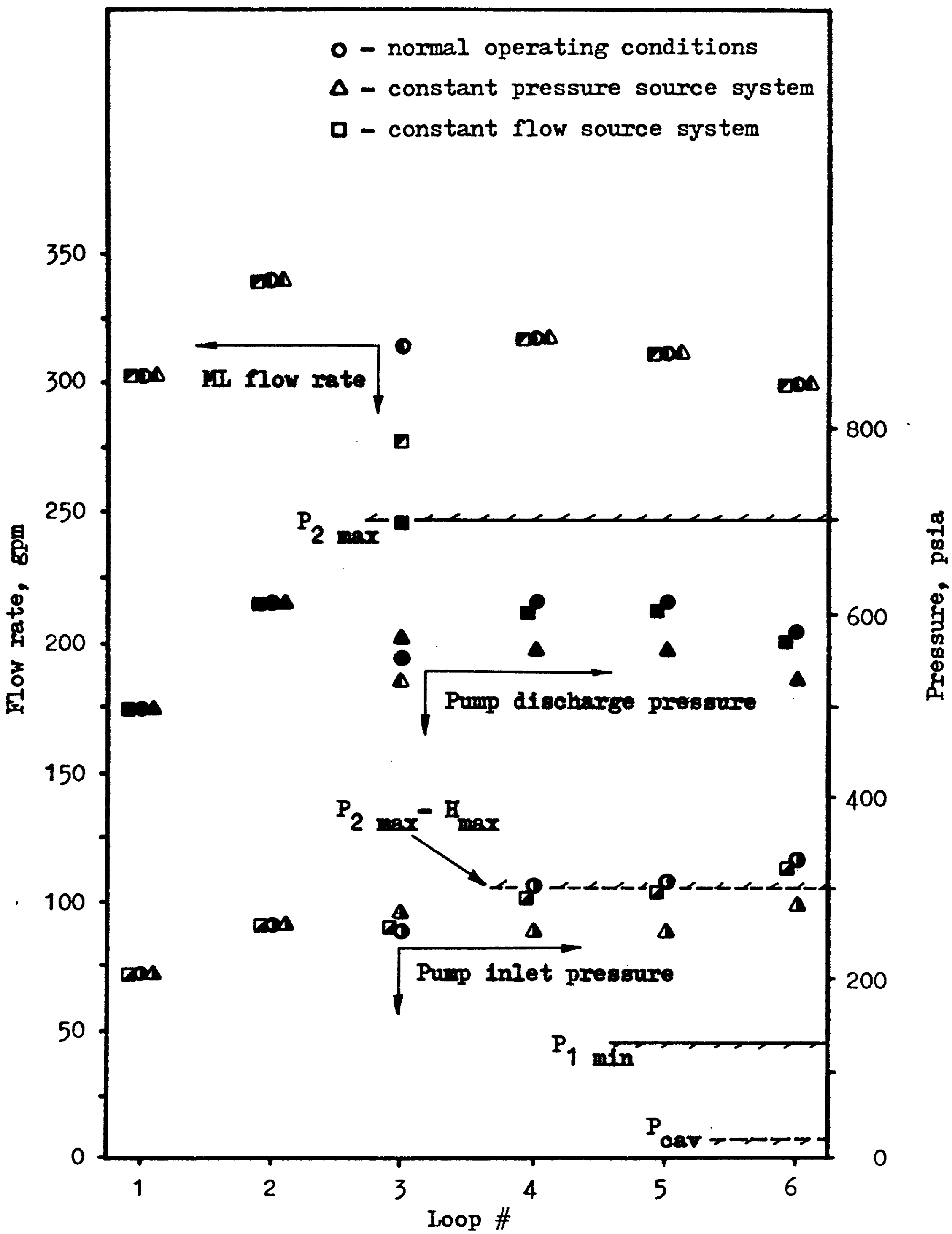


Fig.53. Effect of the DL3 flow resistance increase of 150% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

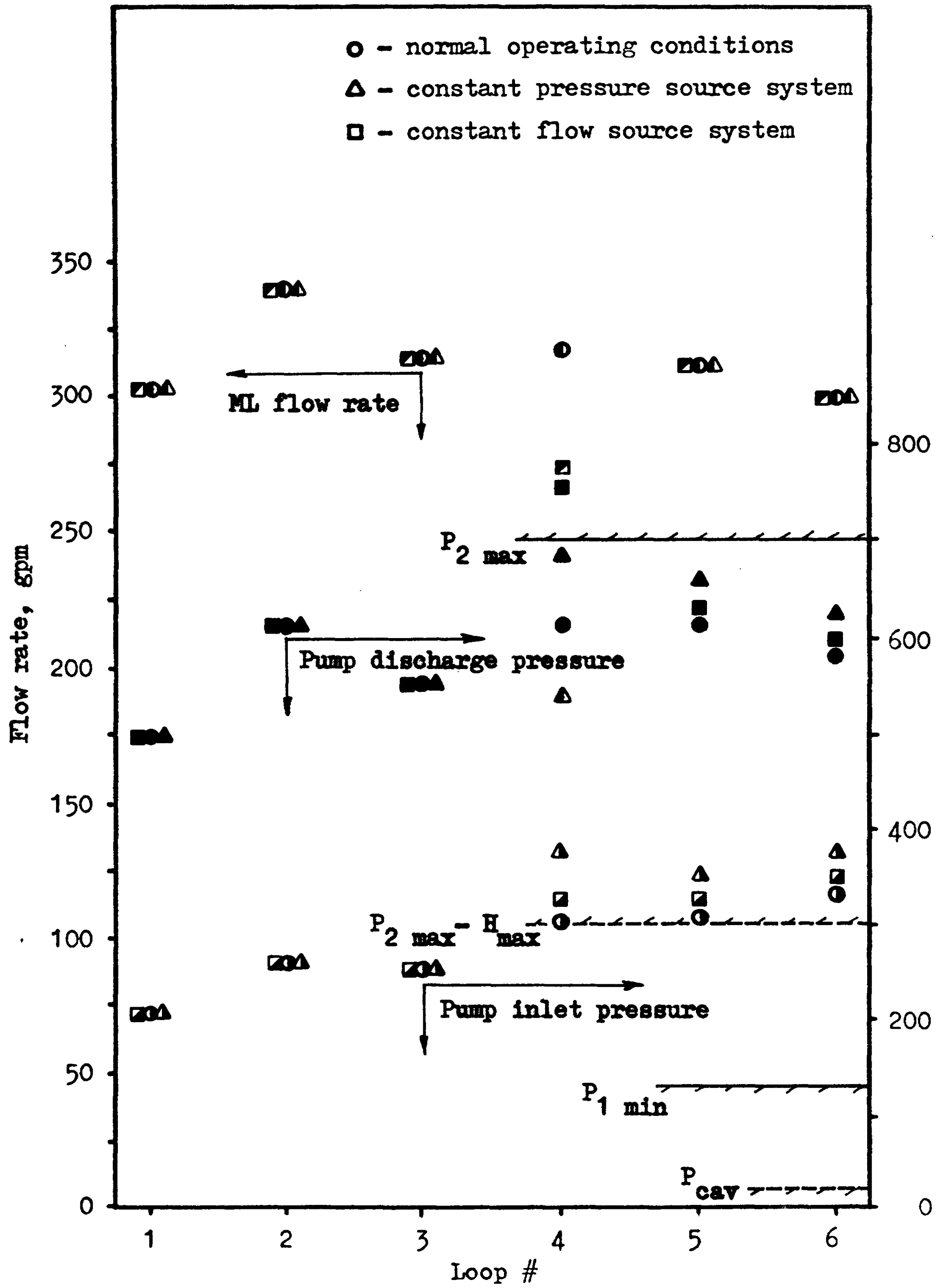


Fig.54. Effect of the DL4 flow resistance increase of 150% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

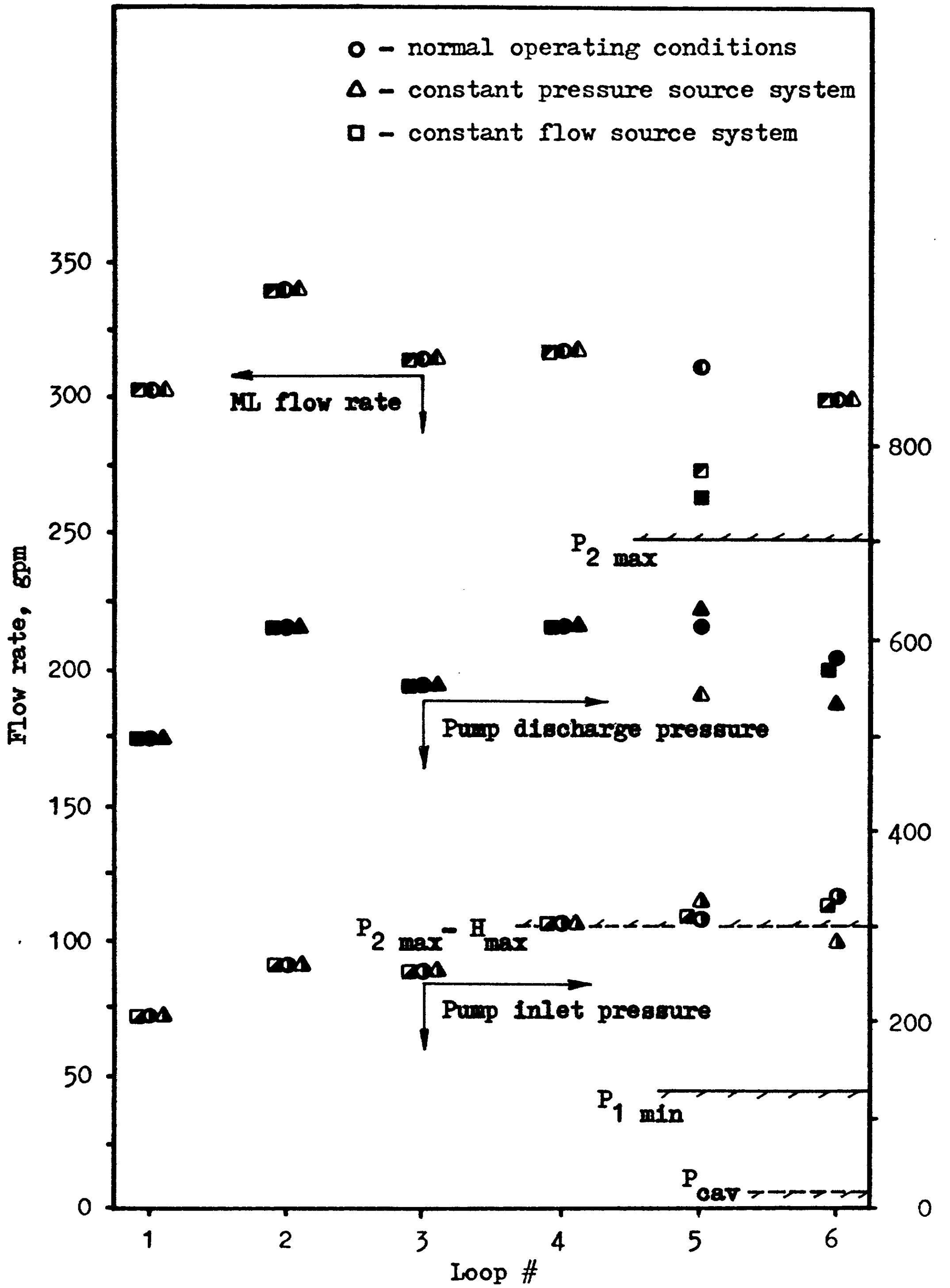


Fig.55. Effect of the DL5 flow resistance increase of 150% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

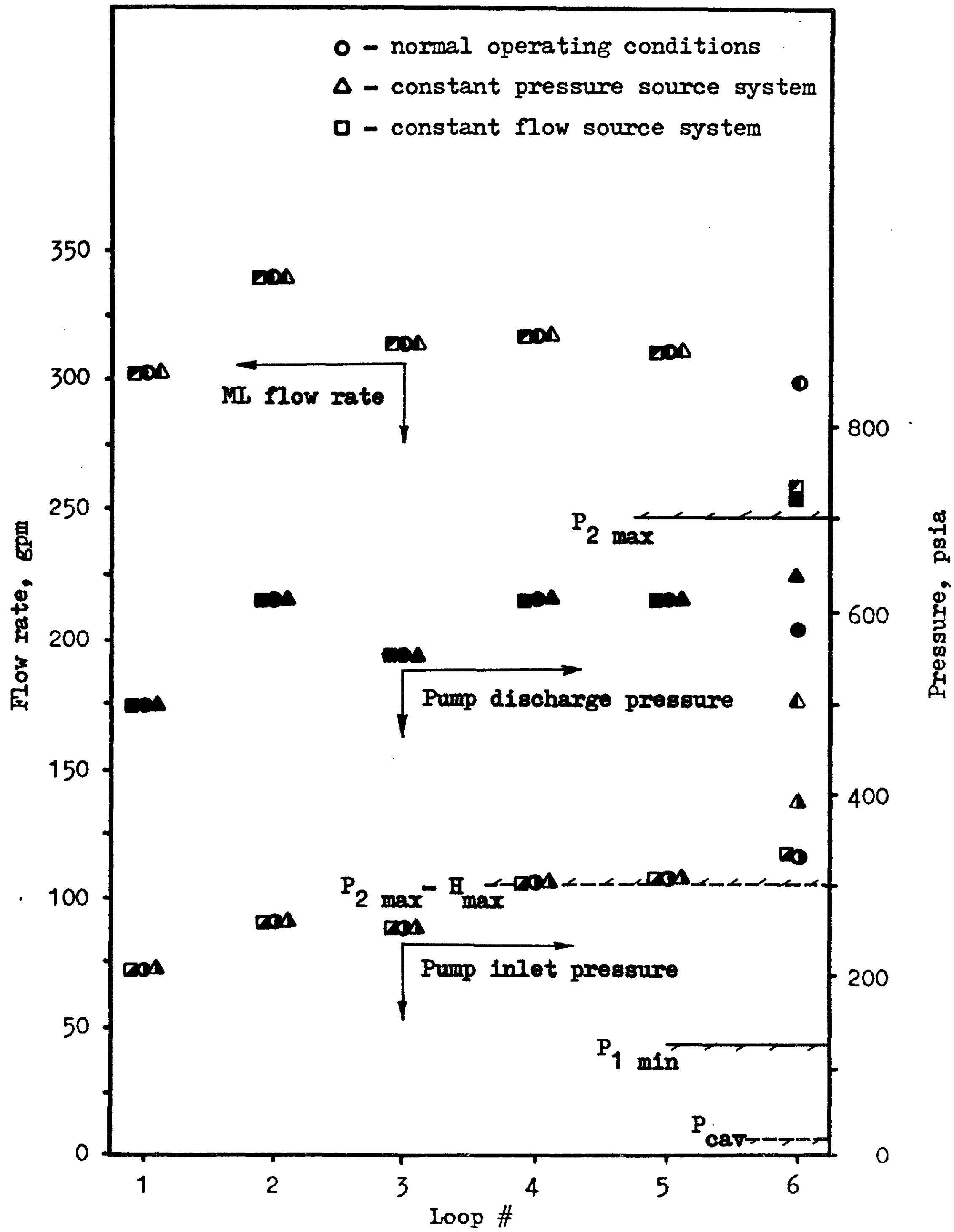


Fig.56. Effect of the DL6 flow resistance increase of 150% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

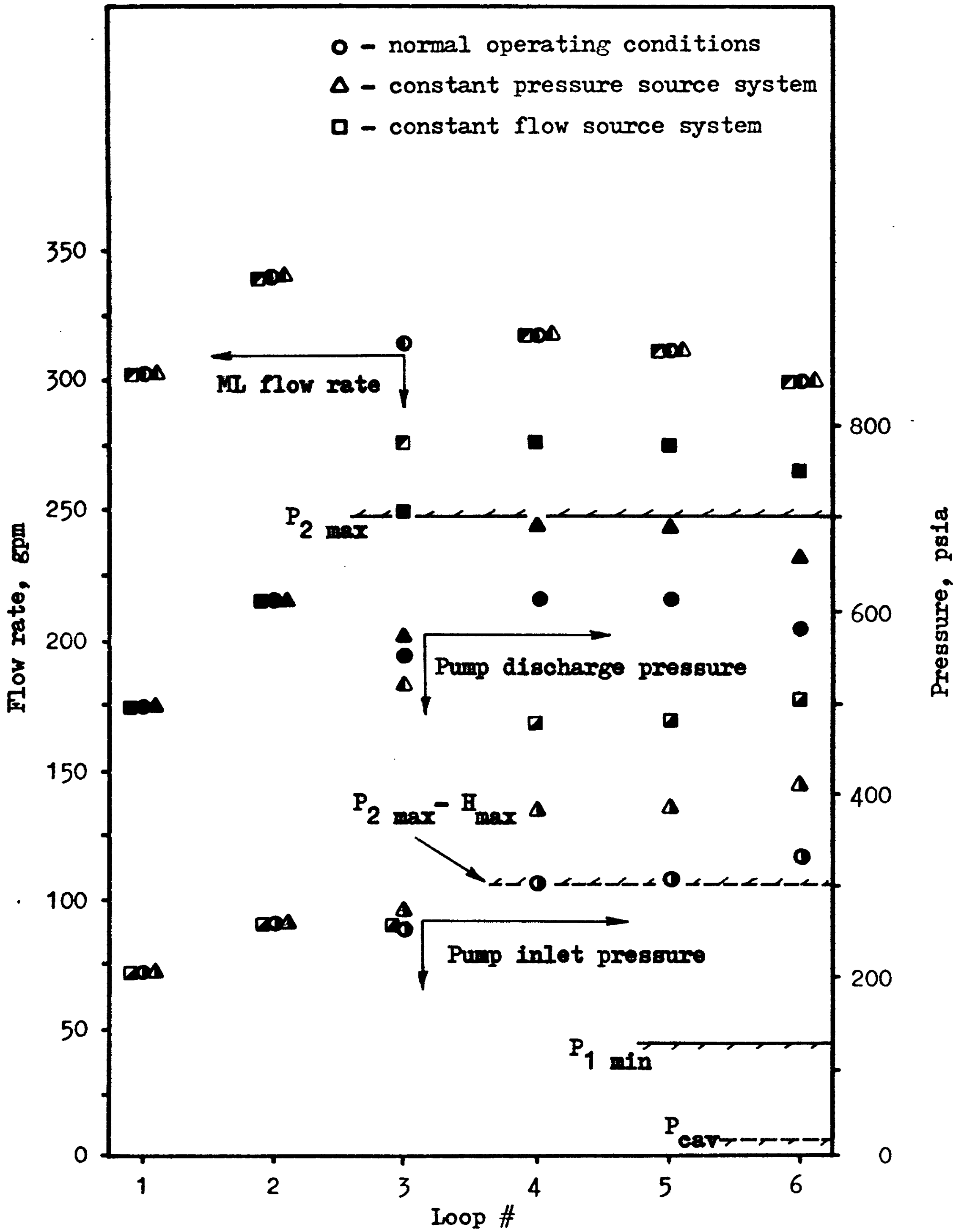


Fig.57. Effect of theML3 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

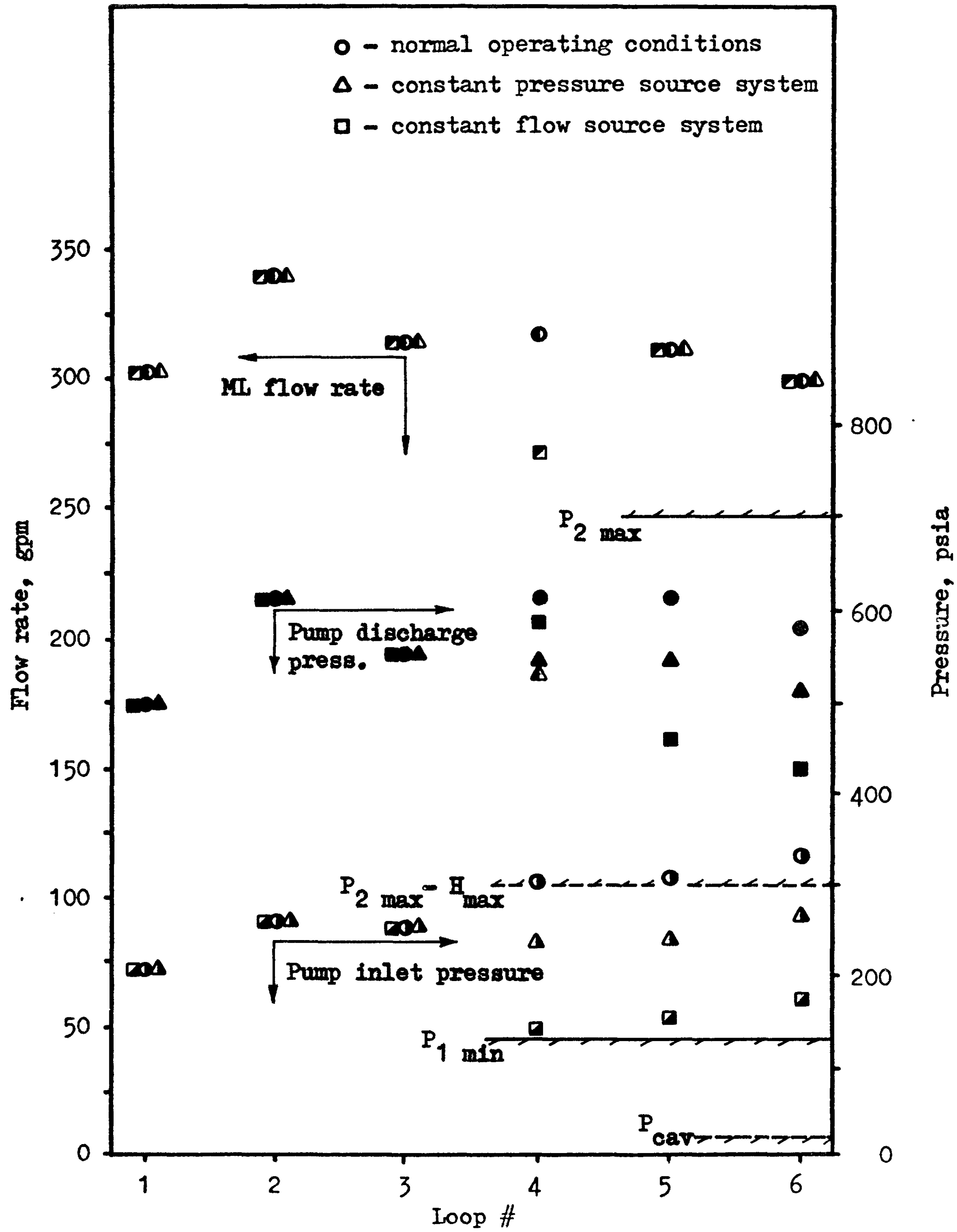


Fig.58. Effect of the ML4 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

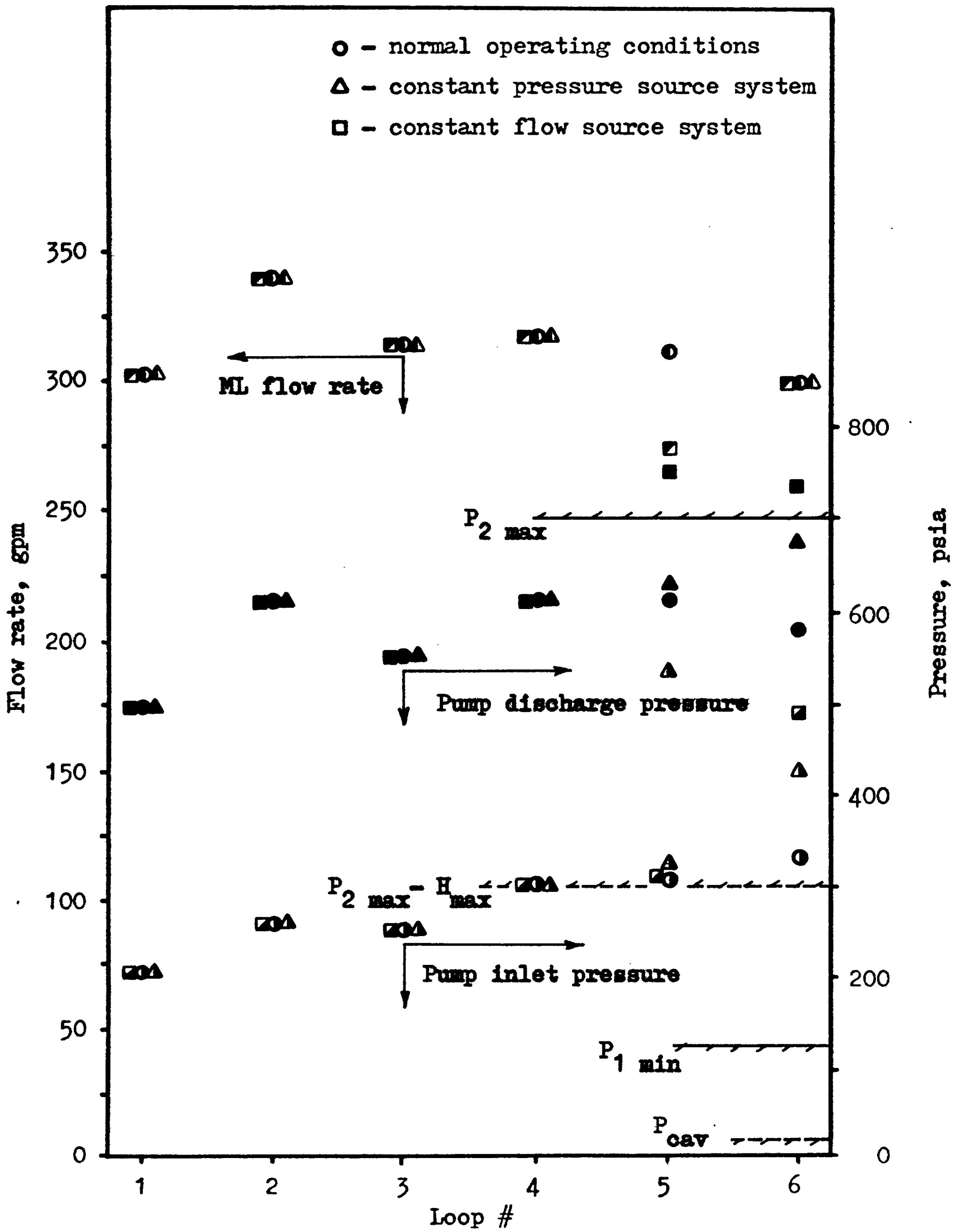


Fig. 59. Effect of the ML5 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

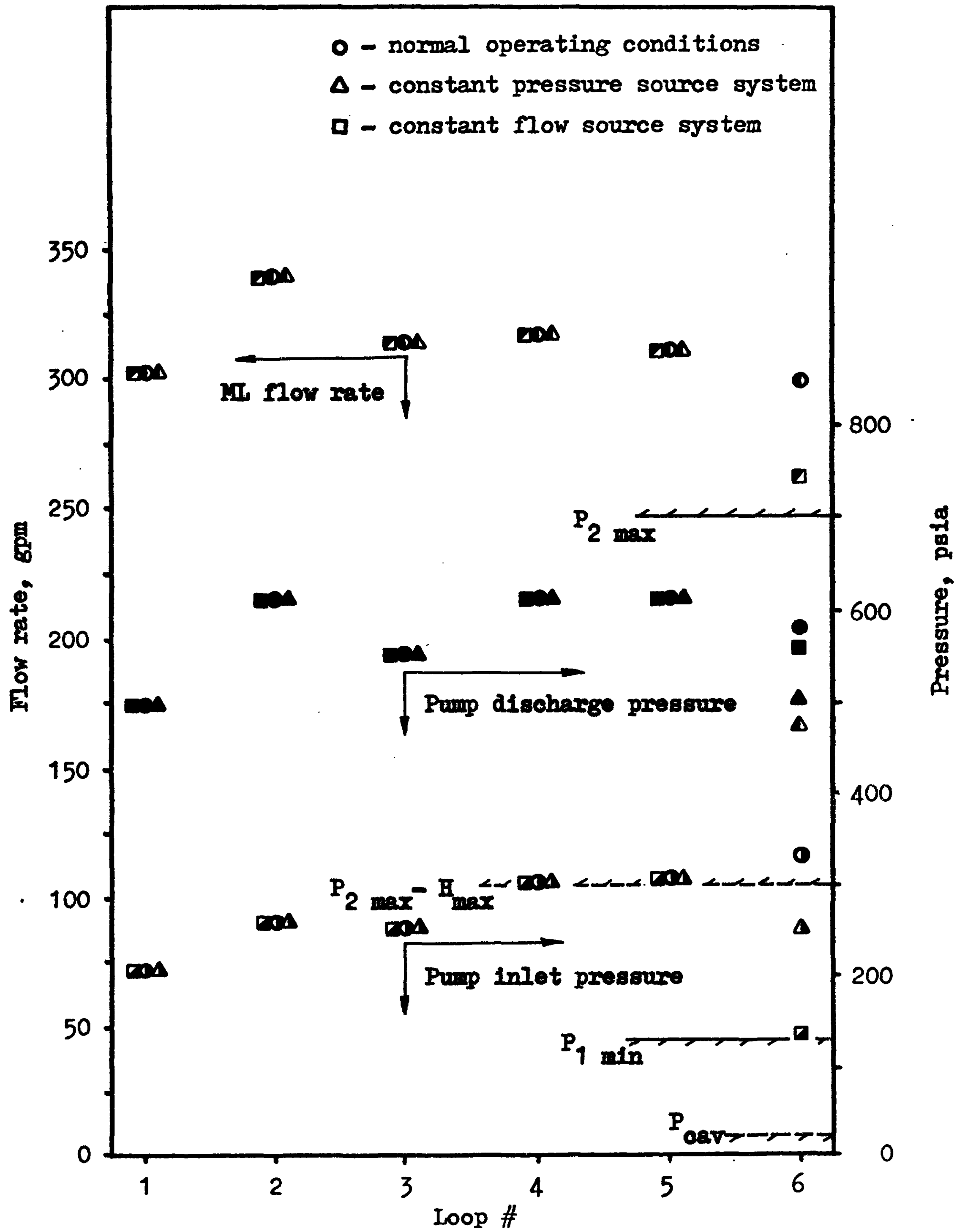


Fig. 60. Effect of the ML6 flow resistance increase of 200% on the inlet and discharge pressures and the ML flow rate in Config. 1 with single pressure control at the line left end (HT attached to loop #1).

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