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NOISE ANALYSIS OF COOLANT DYNAMICS IN TWO-PHASE HEAT TRANSFER SYSTEMS

by

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SUMMARY

The identification and characterisation of the dynamics of coolant flow in a boiling heat transfer loop is considered as a particular case of system identification using noise analysis techniques. The more applicable parts of general noise theory are summarised and it is shown analytically that if a system approximates to one having a single degree of freedom, then simplification of the noise analysis method is possible.

In the preliminary experiments a closed water-loop system was used. The pressure at the inlet to the tubular test section was 55 bar. The maximum heat flux in the test section was about 200 watts cm^{-2}. Measurements intended for detailed data reduction were limited to the inlet flow rate fluctuations and the dynamic component of the pressure drop across the test section. The results were processed using an on-line correlator and an on-line spectrum analyser.

The measured parameters approximated to a Gaussian amplitude distribution. In both measurements the expected low frequency hydrodynamic signals were severely influenced by higher frequency noise sources. The inlet flow rate fluctuations were measured with an orifice plate which generated significant turbulent noise and proved unsatisfactory for detecting the required signals. The test section pressure drop measurements yielded an inverse variance which was a strong function of the test section power, and it is postulated that these measurements can be used to determine the hydrodynamic stability characteristics of the test facility. The present tests do not conclusively confirm the method of analysis outlined in this paper, but they indicate the complexity of two-phase flow dynamics. The various noise sources are being given further detailed study in the current research programme.

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1. INTRODUCTION

It can be said of the majority of operating reactors that their installed safety systems have never been called upon to protect the reactor against an uncontrolled departure from intended conditions. Nevertheless, in the interests of ultimate safety, a great deal of effort is given to analysing the behaviour of reactor cores in accident conditions. A specific safety and operational problem arises from the fact that a typical reactor core consists of a multiplicity of parallel channels through which coolant is forced by a common pressure driving head. The number of channels in a power reactor is large and accordingly flow failure in one channel has a negligible effect on core pressure drop. Research at Winfrith [1], Halden [2] and at AEG [3] as well as other laboratories is directed towards establishing the upper bound to the hydrodynamic operation of liquid cooled boiling type reactors. The normal safety device used to protect the coolant channel is a series orifice or other form of pressure loss mechanism inserted at the entry of the channel. If this pressure loss is too small or the channel power too high then either unstable oscillations or excursive instabilities of the coolant flow could result. The former phenomenon is known as hydrodynamic instability and this paper outlines methods by which incipient hydrodynamic instability may be detected in-pile. These techniques were applied to a high pressure boiling water loop in which a bypass flow existed.

The method used to detect incipient hydrodynamic instability and estimate the margin to stability relies on noise analysis techniques, which assume that small fluctuations $x'(t)$ always occur around the nominally steady state value $\overline{x}$ of all measureable parameters. The inter-relation within and between these fluctuations contains valuable information about the dynamic state of the system. The method can be applied to any situation of parallel channel operation where hydrodynamic instability could be a limit to safe or efficient operation.

2. RELEVANT NOISE THEORY

It is not possible here to give a comprehensive treatment of noise theory, and interested readers are referred to texts such as Bendat and Piersol [4]. In general, a random variable $x(t)$ can be regarded as consisting of a time independent mean component $\overline{x}$ and a small fluctuating component $x'(t)$. A basic assumption of the noise theory presented here is that the process is "linear". This can be tested by comparing the computed probability density functions of the input and output quantities; for a linear system these should have the same shape.

The statistical quantities of interest in this work are the auto-correlation function $R_{xx}(\tau)$ and the power spectral density $S_{xx}(\omega)$, which are defined respectively as

$$R_{xx}(\tau) = \lim_{T\to\infty}\frac{1}{T}\int_0^T x(t)\,x(t+\tau)\,dt \qquad \ldots(1)$$

and

$$S_{xx}(\omega) = \lim_{T\to\infty}\frac{1}{T}\,X_T(\omega)\,X_T^*(\omega) \qquad \ldots(2)$$

where $X_T(\omega)$ is the Fourier transform of the sample $x(t)$ over the period T, and $X_T^*(\omega)$ is the complex conjugate of $X_T(\omega)$. Furthermore, the auto-correlation function and the power spectral density form the Fourier transform pair

$$R_{xx}(\tau) = \frac{1}{2\pi}\int_{-\infty}^{\infty} S_{xx}(\omega)\,e^{j\omega\tau}\,d\omega \qquad \ldots(3)$$

and

$$S_{xx}(\omega) = \int_{-\infty}^{\infty} R_{xx}(\tau)\,e^{-j\omega\tau}\,d\tau \quad . \qquad \ldots(4)$$

3. METHOD FOR ESTIMATING STABILITY MARGIN

3.1 Basic Assumptions

The method used to predict the channel power at which the system would exhibit hydrodynamic instability is based on the following assumptions:

(i) The system dynamics stability is fully accounted for by considering the mass flow and channel pressure drop as dominant terms. No doubt temperature and density fluctuations are of practical importance, but their effect is ignored in this simple approach.

(ii) The system is linear. This assumption limits the use of the method to high pressure systems.

(iii) The system approximates to one having a single degree of freedom and hence can be described by the simple differential equation

$$\ddot{y} + 2\,\xi\omega_n\,\dot{y} + \omega_n\,y = x \qquad \ldots(5)$$

where ω_n is the natural frequency and ξ is the damping ratio. This assumption is based on the fact that a numerical solution of the coupled mass, energy and momentum equations given by a computer code SLIP [5] bears a marked similarity to a damped cosine wave which is a solution to (5). Other possible forms such as a zero order Bessel function or a Sinc function also have similar features, but a second order system is the simplest to use.

(iv) The damping ratio ξ is a linear function of the channel power and further, the numerical value of ξ is a measure of the margin to instability of the channel. Indeed when $\xi = 0$ the channel exhibits self sustaining oscillations.

(v) The spectrum of the input fluctuations is "white". All that is practically necessary is that the input spectrum is reasonably flat in the region of the natural frequency ω_n.

3.2 Response to a Deterministic Disturbance

If a system described by equation (5) is given an initial displacement y_o at zero time, when $\dot{y}(o) = 0$ and the input is zero ($x = 0$), then the solution to (5) is

$$y(t) = y_o \exp(-\,\xi\,\omega_n t)\left\{\cos \omega_n \sqrt{1-\xi^2}\,t + \frac{\xi}{\sqrt{1-\xi^2}} \sin \omega_n \sqrt{1-\xi^2}\,t\right\} \quad . \qquad \ldots(6)$$

It is clear that as $\xi \rightarrow 0$, then $y(t)$ will oscillate indefinitely.

3.3 Response to a White Noise Input

If the system characteristic equation (5) is converted to the frequency domain by use of the complex Fourier integral transform, the result is

$$[(\omega_n^2 - \omega^2) + 2\,j\,\xi\,\omega_n\,\omega]\;Y(\omega) = X(\omega) \quad . \qquad \ldots(7)$$

Multiplying (7) by its complex conjugate and proceeding to the limit, the relationship between the input and output power spectral densities is

$$S_{xx}(\omega) = S_{yy}(\omega)\;[(\omega_n^2 - \omega^2)^2 + 4\,\xi^2\,\omega_n^2\,\omega^2]$$

and the output auto-correlation function can then be computed from

$$R_{yy}(\tau) = \frac{1}{2\pi}\int_{-\infty}^{\infty} \frac{S_{xx}(\omega)\;e^{j\omega\tau}\;d\omega}{(\omega_n^2 - \omega^2)^2 + 4\,\xi^2\,\omega_n^2\,\omega^2} \quad . \qquad \ldots(8)$$

Using assumption (v) that the input spectrum is white, that is $S_{xx}(\omega) = S_o$ (a constant), then the solution of (8) is

$$R_{yy}(\tau) = \frac{S_o}{2\,\pi\,\xi\,\omega_n^2}\;\exp(-\,\xi\,\omega_n\,\tau)\left\{\cos \omega_n \sqrt{1-\xi^2}\,t + \frac{\xi}{\sqrt{1-\xi^2}}\;\sin \omega_n \sqrt{1-\xi^2}\,t\right\} \quad . \qquad \ldots(9)$$

Note that the response function given by (6) and the auto-correlation to a white noise input, equation (9), have identical form.

3.4 Log Decrement and Inverse Variance

Using assumption (iv) that ξ is a measure of the margin to stability of the system and noting that

$$\text{Log dec(impulse response)} \equiv \text{log dec(auto-correlation)} = \frac{2\pi\xi}{\sqrt{1-\xi^2}},$$

then a simple linear extrapolation of the log dec versus power function yields an estimate of the instability threshold power when $\xi = 0$, that is the power at which the channel would exhibit hydrodynamic instabilities.

A similar estimate of the instability power can be made from the auto-correlation function by noting that its value at the origin is simply

$$R_{yy}(0) = \frac{S_o}{2\pi\omega_n^2\xi} = \frac{\text{constant}}{\xi} \quad . \qquad \ldots(10)$$

A linear plot of the inverse of $R_{yy}(0)$ as a function of channel power would again yield an estimate of the instability power when $\xi = 0$. This method, sometimes known as the Halden [2] method, is mathematically equivalent to the log dec method, since the function $R_{yy}(0) = \sigma_{\bar{y}}^2 + (\bar{y})^2$ where $\sigma_{\bar{y}}^2$ is the variance and $\bar{y}$ is the mean value, and measurements are usually referred to zero mean.

3.5 Non-Linear Limits

Assumption (ii), that the system is linear, clearly cannot hold when the variance becomes very large and in this case the oscillations would exhibit non-linear limiting characteristics. In this case extrapolations from the linear portion of either the log dec or inverse variance functions would yield conservative and hence safe estimates of the upper bounds of channel power for a specified flow.

4. EXPERIMENTAL RIG AND DATA PROCESSING EQUIPMENT

A simplified schematic diagram of the experimental rig and data processing equipment is shown in figure 1. The test facility is a closed loop high pressure system capable of operation over a range of pressures up to 70 bar absolute, test section powers up to 150 kW, and flow rates up to 30 ℓ/m. Demineralised water is circulated around the system by a canned turbine pump. The coolant flow rate through the test section is controlled by adjustment of the throttle and bypass valves. A 50 kW electrical preheater controls the coolant temperature at the test section inlet. The test section is 0.63 cm i.d. stainless steel tube and is heated by electrical power supplied from two 75 kW d.c. rectifiers connected in series. The steam produced in the test section is condensed in a steam separator/condenser system. The pump inlet temperature is controlled by the subcooler. All components, valves and pipework are made from stainless steel.

The mean flow rate through the test section is monitored by a turbine flow meter. Strain gauge differential pressure transducers were used to measure the flow rate fluctuations through an orifice plate and the pressure drop fluctuations across the test section. The millivolt signal outputs of these transducers were displayed on a cathode ray oscilloscope before being analysed by the spectrum analyser (Spectral Dynamics Model No. SD 301B) and the correlator (Hewlett Packard Model No. 3721A). The results from the spectrum analyser and correlator were plotted on graph paper using an X-Y plotter.

5. DISCUSSION OF RESULTS

The results obtained from the preliminary experiments for this paper are shown in figures 2 and 3. The amplitude-frequency spectrum of the orifice plate noise is shown in figure 2. A large resonance peak occurred at approximately 36 Hz and no significant peaks were detected in the hydrodynamic frequency region of 1 Hz. The variance results obtained from this unfiltered spectrum proved to be independent of the test section power. More detailed analysis in the low frequency region yielded no significant information which could be interpreted as hydrodynamic noise. It is postulated that the large 36 Hz resonance peak is due to turbulent noise generated by the orifice plate. This suggests that orifice plates are unsuitable for hydrodynamic noise measurements. The tests are being repeated using a standard venturi meter in place of the orifice plate.

In view of the non-significance of the orifice plate noise measurements, attention was transferred to the test section pressure drop measurements. A band-pass filter was used to reject signals outside a frequency range of 0.3 to 4 Hz. The inverse variance of these measurements as a function of power is shown in figure 3. It can be seen that the measurements are a strong function of power and show the expected trend. The experiments indicate an instability threshold power of 42 kW, whilst the predicted instability powers of a linearised frequency response computer code LOCO [6] and the SLIP code were 43.3 kW and 52.5 kW respectively. The large scatter between experiments 21 to 28 is considered to be due to instrumentation noise and so no definite conclusions can be made about the shape of the curve in the low power region.

The shape of the inverse variance versus power curve appeared to confirm the validity of the method outlined in this paper. However, detailed analysis of the auto-correlation functions of the test section pressure drop noise failed to give the results expected. The correlograms were similar in shape to either a highly damped signal or bandwidth limited white noise. These results indicate that there are at least two different noise sources present in the system which are a function of power. Two identifiable sources are the hydrodynamic oscillatory mode which is being investigated in this paper, and pressure fluctuations due to subcooled boiling [7]. Both these noise sources are known to increase with power. If the variances of these two sources are σ_1^2 and σ_2^2 respectively, and if it is assumed that these sources are uncorrelated, then the observed inverse variance is given by

$$\frac{1}{\sigma} = \frac{1}{\sigma_1^2 + \sigma_2^2} = \frac{1}{\sigma_1^2\,(1 + \sigma_2^2/\sigma_1^2)} .$$

If it is assumed that the ratio σ_2^2/σ_1^2 is a weak function of the test section power, then the observed inverse variance of the present measurements is proportional to the inverse variance of the hydrodynamic noise. The instability threshold power obtained from the present experiments would therefore be a valid result. However further tests are needed to confirm both this hypothesis and the method presented in section 3 of this paper.

6. CONCLUSIONS

From the present tests it can be concluded that

(1) the inverse variance measurements need to be confirmed by further tests in order to elucidate the relative strengths of the different noise sources present in the system.

(2) the hydrodynamic noise signals need to be enhanced before the present method of analysis can be confirmed conclusively.

(3) orifice plates are unsuitable devices for detecting hydrodynamic noise signals in two-phase heat transfer systems.

7. ACKNOWLEDGEMENTS

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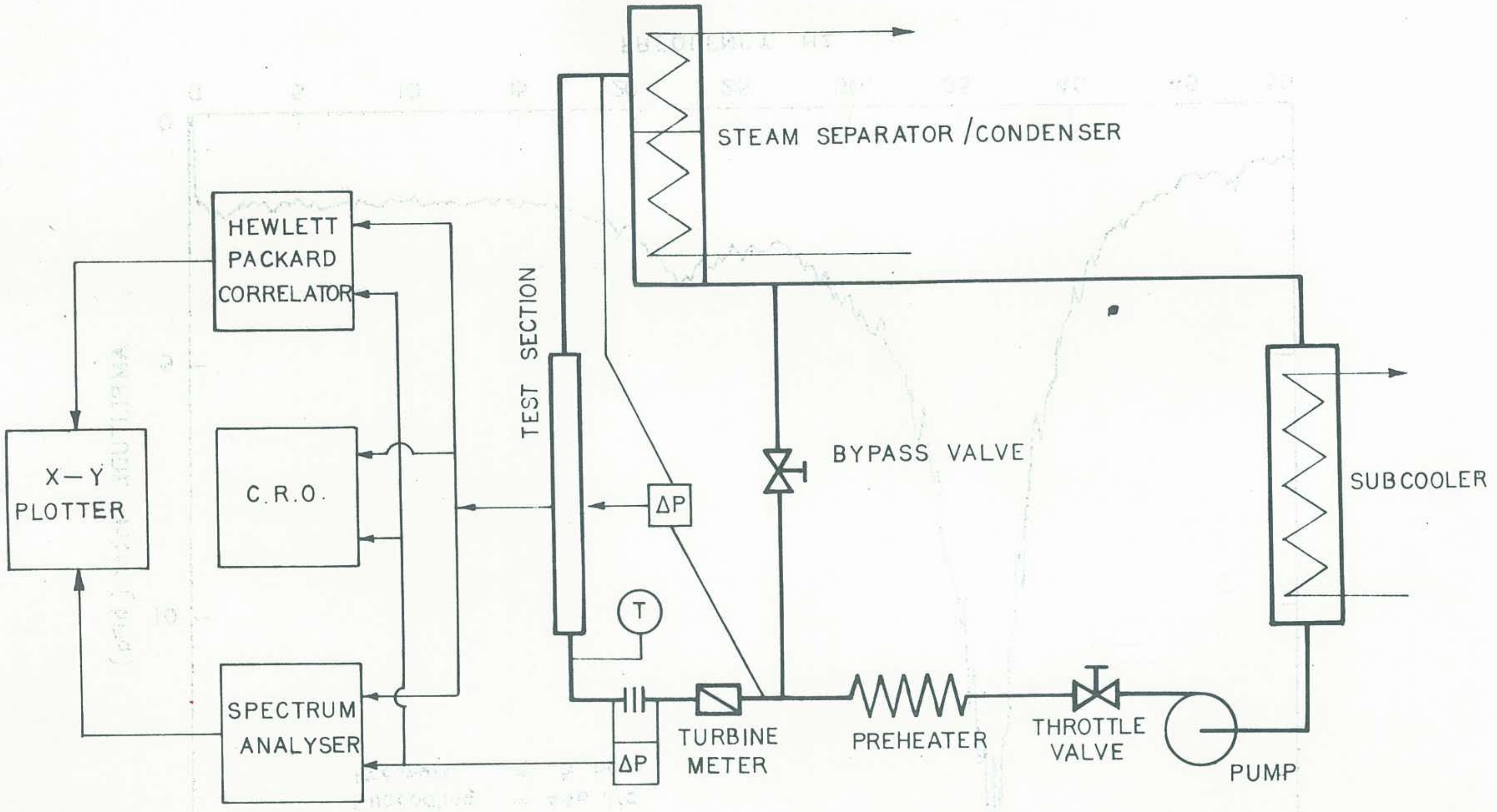

FIGURE I SIMPLIFIED SCHEMATIC DIAGRAM OF EXPERIMENTAL RIG AND DATA PROCESSING EQUIPMENT

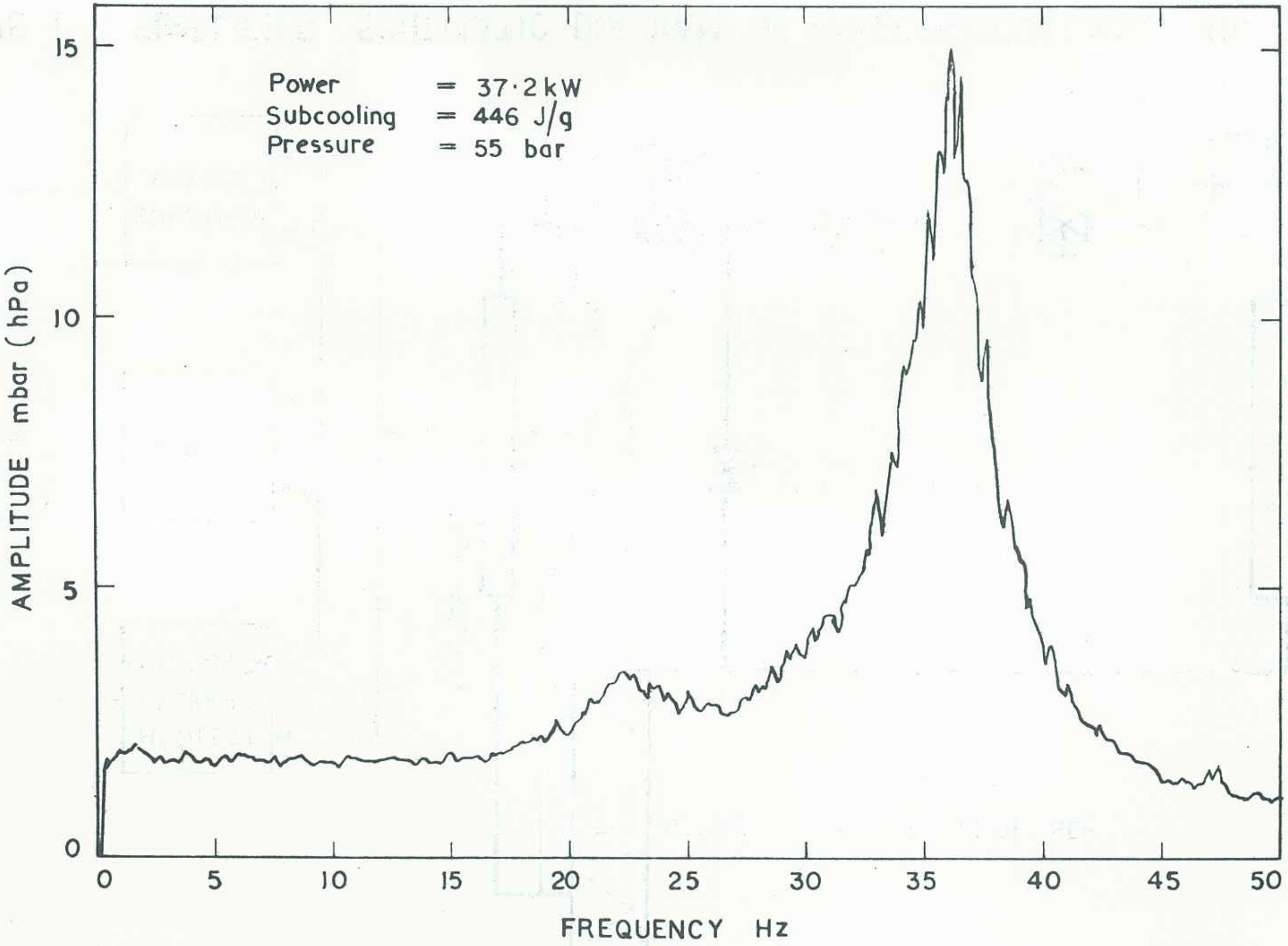

FIGURE 2 - AMPLITUDE - FREQUENCY SPECTRUM OF ORIFICE PLATE PRESSURE DROP FLUCTUATIONS

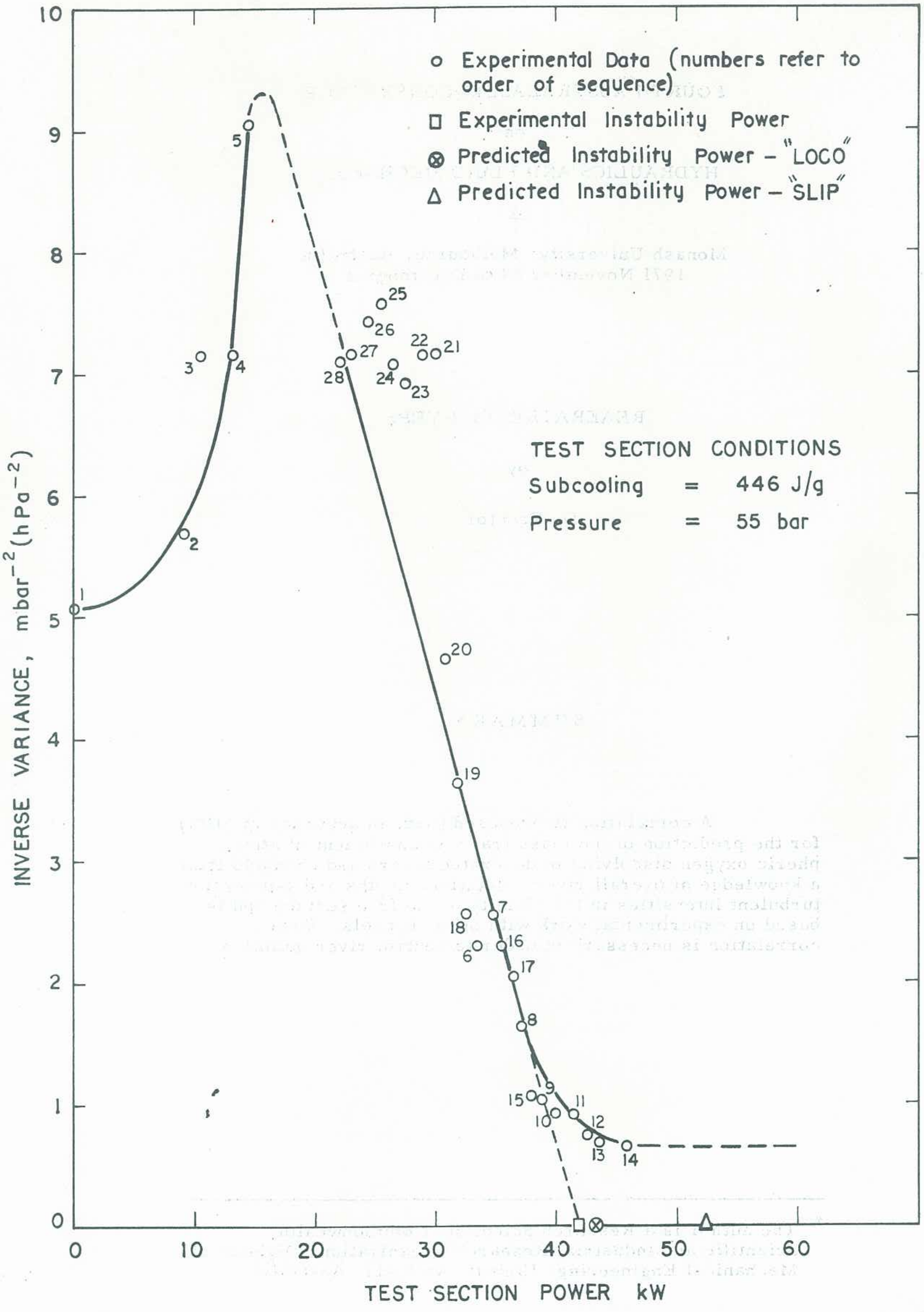

FIGURE 3 INVERSE VARIANCE OF TEST SECTION PRESSURE DROP FLUCTUATIONS AS A FUNCTION OF POWER